KTH Industrial Engineering
and Management

Waste heat recovery from SSAB's Steel plant in Oxelösund using a Heat Pump

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Master of Science Thesis

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KTH Industrial Engineering and Management	**Master of Science Thesis EGI_2016-082 MSC** Waste heat recovery from SSAB's Steel plant in Oxelösund using a heat pump Amir Abbas Sohani	
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Sammanfattning

Detta projekt är inriktat på spillvärmepotentialer inom järn och stålindustrin. Högtemperaturvärme-pumpar för medelvarma temperaturkällor har modellerats. SSABs stålverk i Oxelusund har använts som exempel. Järn- och stålindustrin i Sverige är storkonsument av energi, tillsammans med pappers och massaindustrin. Det finns också en stor potential för spillvärmeåtervinning i stålindustrin. Det görs redan i Luleå t ex [1].

Järn och stålindustrins produktionsmetoder och spillvärmeåtervinning, speciellt i USA och Sverige har studerats genom en litteraturstudie. Dagens metoder och potentialer för spillvärmeåtervinning inom järn och stålindustrin i Sverige studerades speciellt. SSABs anläggning i Oxelösund, har i decennier planerat inte bara att värma Oxelösunds stad som idag, utan också expandera till näraliggande Nyköping bara 12 km bort [2].

Typiskt är den maximala framledningstemperaturen till Nyköpings fjärrvärmenät 110 °C den kallaste dagen. En spillvärme-värmepump når normalt inte upp till så höga temperaturer. Dock räcker 80 °C maximal framledningstemperatur från värmepumpen för att nyttiggöra spillvärmekällan kontinuerligt. Även en lägre temperatur som 75 °C skulle sannolikt räcka. Bara några få fjärrvärme-värmeväxlare i några hus skulle behöva bytas för att denna lägre temperatur skulle räcka till. De överskjutande graderna mellan 80 °C (75 °C) och 110 °C kan tas med värme från t ex existerande biobränslepannor lokalt i Nyköping.

Att använda värmepumpar i detta sammanhang är inte självskrivet. Generellt är värmeflödena från ett stålverk så högtempererade att ingen värmpump behövs. Om man försöker komma åt dessa högtemperaturflöden i en gammal anläggning kan det bli väldigt dyrt och störa produktionen. Därför

koncentrerades studien på medeltemperaturkällor (30 °C till 40 °C) och användande av högtemperaturvärmepumpar. Sådan värme dumpas nu med kyltorn. På så sätt kan 50 % av Nyköpings värmebehov tillgodoses med lätt tillgänglig spillvärme. Om man antar en värmefaktor på cirka 5, och lägger till värmepumpens förbrukade elektricitet blir det 62 % av Nyköpings fjärrvärmebehov.

Oxelösundanläggningen är bara ett exempel och studien fokuseras på högtemperaturs-industriella värmepumpar HITIHP för sådana här och liknande användningar. Lämpliga komponenter och köldmedia har undersökts och generella konstruktionsprinciper av HITIHP föreslås. En litteraturstudie för att finna de bästa HITIHP-köldmedierna har gjorts.

En tvåstegs högtemperaturvärmepump, som använder den tillgängliga värmekällans kapacitet och temperaturer tillsammans med fjärrvärmenätets krav, har modellerats och simulerats. Simuleringen har huvudsakligen gjorts med programmet EES. R245fa har t ex visat sig vara lämpligt som köldmedium i det andra steget av en högtemperaturvärmepump. Med R245fa kan till och med högre temperaturer än 90 °C uppnås till fjärrvärmesystemet. Tidigare skulle R134a ha använts i en sådan här applikation, men R245fa har t e lägre GWP (Global Warming Potential omkring 1000 istället för omkring 1300)[3]. Många olika köldmedia har simulerats i lågtemperatursteget av värmepumpen som initialt antogs vara en skruvkompressor-kaskad-värmepump. En större värmpump med två turbokompressorsteg och flashtank har också simulerats. Den gav också tillfredställande resultat. I det senare fallet studerades både R1234ZE(z) och R245fa som gav goda resultat men R1234ZE(z) ger mycket lägre GWP.

Alla värmefaktorer (COP, energibehov, kondensortryck och tryckförhållanden (hög-/lågtryck) jämfördes. R245fa-R245fa och R600a-R245fa studerades noga i tvåstegs-kaskad-systemet med skruvkompressor. Dessa kombinationer gav bäst resultat. R717-R245fa var också bra men hade andra begränsningar. I tvåstegssystem med turbokompressorer och flashtank visade sig visade sig R1234ZE(z) ge gen bästa värmefaktorn. Man hade naturligtvis inte heller något temperaturfall i någon värmeväxlare mellan de två stegen. Om SSABs spillvärme av någon anledning inte skulle vara tillgängligt kan en sådan värmpump istället använda havsvatten som värmekälla.

Begränsningen av koldioxidutsläppen är mycket svåra att beräkna. Detta kommer att bero mer på politisk övertygelse än på lättbevisade fakta. En mycket grov beräkning av kostnaden har också gjorts. Uppskattningsvis kommer projektet att kosta mellan 420 och 450 MSEK. Kostnadsuppskattningen inkluderar värmepumpen och en 12 km lång förbindelse till Nyköping. Kostnaden för värme levererad till Nyköping, kommer att variera mellan 0,2 kr/kWh och 0,65 kr/kWh när elpriset varieras mellan 0,5 och 2 SEK/kWh. Den högre värmkostnaden 0,65 kr/kWh beror också på att östersjövatten – inte spillvärme används som värmekälla.

Värme från ett kyltorn kan återvinnas med en högtemperaturvärmepump. Den kan levereras från Oxelösund till Nyköping. De ekonomiska detaljerna har bar studerats översiktligt. Faktorer som om renovering den gamla pannan i Nyköping eller SSABs kyltorn kunde senareläggas, skulle kunna förbättra intresset för projektet. Ett spillvärmerör mellan Oxelösund och Nyköping har studerats sedan mitten av 70-talet av t ex Lars-Åke Cronholm [4]. Kan det vara dags nu?

Abstract

This project was focused on waste heat potentials in the iron and steel industry. High temperature industrial heat pumps (HTIHP) for medium temperature, waste heat recovery were modelled. The SSAB iron and steel plant in Oxelösund was used as an example. The iron and steel industry in Sweden is a large energy consumer, together with the pulp and paper industry. There is also a large potential for waste heat recovery in the steel industry. This is already done in for instance Luleå [1].

Iron and steel production methods and waste heat recovery in the world, especially in the US and Sweden, have been reviewed in a literature study. Current methods and potentials of waste heat recovery in the iron and steel industry of Sweden were especially reviewed. The SSAB iron and steel plant in Oxelösund has been planning for decades, not only to heat the city of Oxelösund as today, but also to expand to the nearby city of Nyköping 12 km away [2].

Typically the maximum temperature entering the district heating network of Nyköping would be 110 °C on the coldest day. The heat pump output from a waste heat recovery plant generally does not have to reach such a high temperature. However, 80 °C maximum forward temperature would surely be enough to use recovered heat all the time. Even a lower temperature like 75 °C would probably be sufficient – as only a few heat exchangers in individual houses then would have to be changed, to accept that lower temperature. The extra degrees between 80 °C (75 °C) and 110 °C can be taken with heat from e.g. existing biofuel furnaces locally in Nyköping.

Using heat pumps in this context is not self-evident. Generally the heat flows from a steel plant are available at such high temperatures that no heat pump ideally is needed. However collecting the heat at those high temperatures, in an old plant, can get very expensive and interfere with the processes. Therefore the study is focusing on medium temperature (30 – 40 °C) waste heat potentials implementing **H**igh **T**emperature **I**ndustrial **H**eat **P**umps (HTIHP). The heat is now being rejected by a cooling tower. That way, easily available waste heat, can cover 50% of the total need from Nyköping. Assuming a COP of around 5 and adding the electricity needed to run the heat pump, the total will result in totally 62% of the energy need for Nyköping.

The Oxelösund Plant is just an example and the study is really focusing on HITIHP for this and similar purposes. Appropriate components and refrigerants have been evaluated and the general layouts of proper HITIHP types are suggested. A literature study on the best choice of refrigerant in the high temperature heat pump has been done.

A two stage high temperature heat pump has been modeled and simulated using the available heat sink capacity and temperature, together with the demanded temperatures in the district heating network. The simulation has mainly been performed using the EES software. R245fa is e.g. a good candidate as refrigerant in a second stage (high temperature stage) of a two stage cascade heat pump. With R245fa even higher temperatures than 90°C to the district heating can be achieved. Earlier, R134a would be used in this application but R245fa has e.g. a lower GWP (around 1000 instead of around 1300) [3]. Many different refrigerants have been simulated in the first of two stages of a smaller screw compressor driven cascade heat pump. Also a two stage turbo compressor throttling heat pump, using a flash tank, has been simulated, showing a good performance. In the latter case both, refrigerants R1234ze(z) and R245fa have good characteristics but R1234ze(z) has a much lower GWP.

All COPs, compressor energy consumptions, condenser pressures, pressure ratios were compared. R245fa-R245fa and R600-R245fa were studied in the two stage cascade systems. They came out with the best results. R717-R245fa also showed a very good performance, but had other limitations. In two stage flash tank systems, R1234ze(z) had the best performance (COP) and no temperature loss between the two stages (like in the cascade systems). If SSAB Oxelösund's blast furnace and cooling tower water would not be available, the turbo heat pump can produce the demanded heat, using sea water as heat source instead.

The CO_2 emission reduction is very hard to calculate. That will be more of a political conviction problem. A very rough cost estimation of the projects investment cost is also done. It will cost between 420 and 450 MSEK. This cost estimation includes a heat pump and 12 km pipe to Nyköping. The cost of heat delivered in Nyköping will vary between 0,2 and 0,65 SEK/kWh when the cost of electricity is varied between 0,5 and 2 SEK/kWh (include taxes). In that price the capital costs for the heat pump and pipe is included. The high cost level 0, 65 SEK/kWh assumes that sea water is used as heat source.

A cooling towers waste heat can be recovered, using a high temperature heat pump. This heat can thus be delivered from Oxelösund to Nyköping. The economic viability of this idea is only superficially covered. Factors like if the old furnace in Nyköping needs upgrading, which could be postponed, could possibly tip the project into go. Maitenance cost, of the existing cooling tower, is another such factor, initiating the project. A waste heat pipe between Oxelösund and Nyköping has been studied at least since the middle of the 1970:s by e.g. Lars Åke Cronholm [4]. Could it be the right time now?

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Table of Contents

Nomenclature

Abbreviations

IHP	Industrial heat pump
HTIHP	High temperature industrial heat pump
DH	District heating
R600	Butane
R600a	Isobutane
R245fa	Pentafluoropropane
R717	Ammonia
R744	CO_2
CO_2	Carbon dioxide
R718	water
R134a	Tetrafluoroethane
COP	Coefficient of performance
GWP	Global warming potential
ODP	Ozone depletion potential
TEWI	Total equivalent warming impact
CCC	Closed cycle compressor
MVR	Mechanical vapor recompression
TVR	Thermal vapor recompression
H_x	Heat exchanger
CDQ	Coke dry quenching
BF	Blast furnace
BOF	Basic Oxygen Furnace
AOD	Argon Oxygen Decarburization
DRI	Direct reduced iron
LPG	Liquefied petroleum gas
EAF	Electric arc furnace
RHB	Radiant heat boiler
TRT	Top pressure recovery turbine
COG	Coke oven gas
BFG	Blast furnace gas
ORC	Organic Rankine Cycle
TPV	Thermophotovoltaic
TEG	Thermoelectric generator
PCM	Phase change materials
SEK	Swedish krona

Symbol

h	Enthalpy (kJ/kg.K)
m	Mass flow rate (kg/s)
φ	Density (kg/m^3)
P	Pressure (bar)
Q	Heating or cooling capacity (kW)
T	Temperature (°C)
V	Volumetric flow rate
Cp	Specific heat (kJ/kg)
η	Compressor efficiency
is	Isentropic
In	Inlet

Out	Outlet
LP	Low pressure
HP	High pressure
T_b	boiling temperature (°C)
T_{cr}	critical temperature (°C)
P_{cr}	critical pressure (bar)
V_m	meat temperature difference between fluids
U	overall heat transfer coefficient $\left(\frac{W}{m^2K}\right)$

1 Introduction

1.1 Background

Growing energy prices and climate concern increases heat pump usage especially in household applications, to reduce the cost for energy consumption. Demands for CO_2 emission reduction and cost reduction in industry, has also lead to the design and utilization of high capacity heat pumps in commercial applications [5].

The purpose of industrial heat pumps is to upgrade low temperature waste heat in industry and offer it to customers. Industrial heat pumps are made for low, medium and large capacity [5] [6].

One of the most important advantages when using IHP is their environment benefit. Using waste heat, can in many cases replace oil and coal for heating. In Sweden though, biofuel or solid waste incineration is the most likely fuel to be replaced. The waste heat is being generated mainly in the pulp-, iron- and steel industry and also by petroleum refining. Replacing fuel also reduces the SO_x and NO_x emissions. The potential benefit when using industrial heat pumps in the food and beverage industry only would reduce CO_2 emissions totally 40 million tons per year in eleven countries, as illustrated in figure 1. This can be done solely by replacing steam boilers with heat pumps below 100 °C. Especially china and USA would benefit from this [7]. Figure 2 shows the energy utilization in Europe [5].

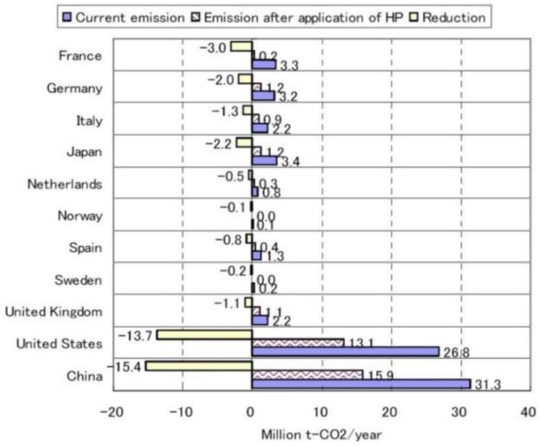

Figure 1: Potential of CO2 reduction in the food and beverage industry. (Source: HTPCJ, 2010) [7]

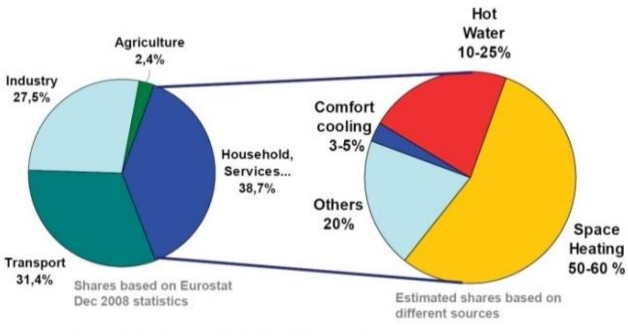

Figure 2: Final Energy Consumption -EU 27- by Sector (2006) [5]

Figure 3 illustrates the total final energy consumption in Sweden, 369 TWh. The industrial part is 144 TWh. Most of the electricity is used in the industry and residential sectors [8].

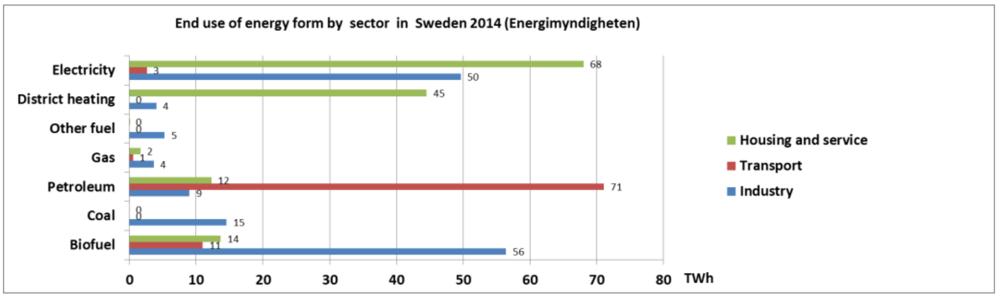

Figure 3: Total Final energy use in Sweden 2014 [8]

According to figure 2, (27,5)% of the energy consumption in Europe is used in industry. Figure 4 illustrates the main industry energy consumption in five countries. Around 1/3 of the total energy consumption is electricity and 2/3 is oil and gas. The steel industry is a large consumer [5].

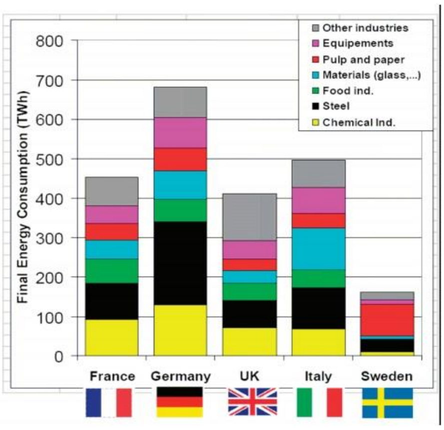

Figure 4: Energy consumption in industries in Europe [5]

Figure 5 illustrates that in Sweden, the pulp and paper industry is the largest energy consumer sector with 51%. The iron and steel sector comes next with 16% and the chemical sector after that with 9% respectively. Data from The Swedish Energy Agency – Energiläget 2015 [8].

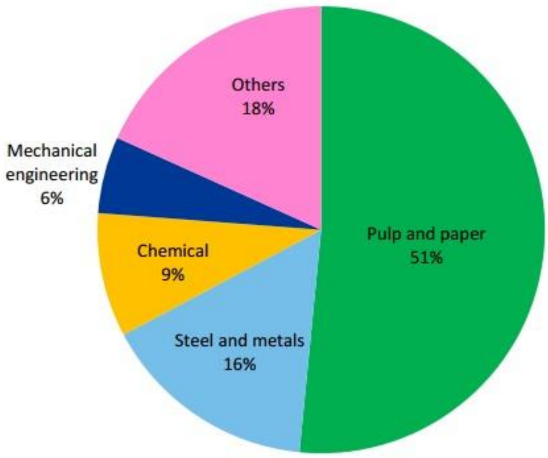

Figure 5: Energy use in Sweden industry 2015 [8]

According to the last long term projection from the Swedish Energy Agency, the total energy consumption would increase to 433 TWh in 2020 where after the consumption growth would then slow down. Thus by 2030 the total consumption would be 441 TWh. However industry consumption would increase faster. Due to the economic recovery, more coal and electricity is likely to be used in the iron and paper industry [7],[8].

The Swedish Energy Agency and the Swedish Environment Protection Agency, have proposed a climate Roadmap 2050, presented to the government. This roadmap aimed to achieve zero net GHG emissions, using cost efficient methods. One of the factors mentioned, was to reduce industry's emissions to about zero using new technologies adapted to the different sectors e.g. mining, iron and steel, cement, chemistry and paper industry [9].

SSAB EMEA Oxelösund Iron and Steel Company has two blast furnaces, a coke plant, a rolling mill and other facilities such as a lime plant, oxygen plant and power plant. They all produce waste heat flows in different part of the production process resulting in a potential both for direct use and by using high temperature heat pumps [10].

The power plant produces today district heating to the smaller town Oxelösund. However there is a larger heat demand in Nyköping, which could be covered using waste heat recovery from SSAB power plant in Oxelösund [10],[11].

Heat pumps are already utilized in the iron and steel industry, for instance in the Netherlands (Corus Ijmuiden) 1998. An absorption heat pump produces low pressure steam with 1.7-3.5 bar and 130°C using waste heat recovery from strip mill cooling water. As a result, there was a fuel saving about 11,1 (kWh/tonne steel) while increasing electric consumption with only 0.15 (kWh/ton)[13].

1.2 Aims and objectives

The aim of the project is to make survey of waste heat recovery in Swedish Iron and Steel industry and find out a way to cover 50% Nyköping's heat demand using waste heat from the SSAB plant in Oxelösund. 12 km piping for hot water to Nyköping would be needed.

Only using, "easy to come by", cooling tower heat, would be possible. Two stage high temperature industrial heat pumps producing forward water in the range 69°C up to 110°C have been modelled. This feasibility study is essentially mostly about finding of waste heat sources in the SSAB plant and proper ways to use them. Heat pump types and refrigerants are researched.

1.3 Methodology

To begin with, iron and steel production methods and waste heat recovery in USA and Sweden is surveyed, then potentials and current methods of waste heat recovery in iron and steel of Sweden is reviewed in the literature study. Afterward all waste heat sources in Oxelösund iron and steel company are investigated and a feasibility study is done covering SSAB company energy demand and waste heat potentials. It seems possible to recover heat from the blast furnace using a high temperature industrial heat pump instead of using the present cooling tower. A heat pump would thus be able to cool down the water temperatures from the blast furnace while delivering heat to Nyköpings district heating system.

Next, a high temperature industrial heat pump with components and applications was surveyed in a literature study. Proper components for an industrial heat pump application for the heat source, heat sink, temperatures and capacity were surveyed. Some types of industrial heat pumps were considered and finally one type recommended.

The refrigerant choice is critical. Therefore, many industrial refrigerants were reviewed in literature study. A proper refrigerant must have suitable boiling and evaporation temperatures and pressures, critical temperature and pressure, must fit the compressor type and preferably be non-toxic and non-flammable.

The heat pump system was modelled using the EES software while some other calculations were done using Excel. Due to the relatively high temperature in the condenser and relatively high pressure ratio, two types of two stage heat pump were modeled. A two stage cascade using different refrigerants and a two stages throttling one refrigerant. The latter has no heat loss between the two stages and therefore often achieves a higher COP. Because of the possible variations in the heat source temperature, the different heat pumps computer-models were simplified using just one average temperature. The, throttling system was simulated also using sea water as the heat source if Oxelösund's blast furnace would be unavailable.

Eventually, the simulation results such as the COP, condensation pressure, compressor energy consumption, pressure ratio and other indexes in the heat pumps using different refrigerants are compared and discussed. The CO_2 emission reduction and a possible project cost using a throttling heat pump including a 12 km district heating pipe between Oxelösund and Nyköping is very roughly estimated.

In summary these tasks were to be considered:

1. Literature study:
    a. Iron and steel production methods and waste heat recovery (USA, Sweden and SSAB)
    b. High temperature industrial heat pump and refrigerants.
2. Feasibility study:
    Oxelösund's SSAB Iron and Steel waste heat sources potentials
3. Model and simulate HTIHP (high temperature industrial heat pump) in EES and Excel
4. Analyze heat pump performance, CO_2 emission reduction and clarified total project cost

2 Waste heat recovery in Iron and Steel industry

This part considers iron and steel production methods and waste heat recovery techniques. Then the Swedish iron and steel industry is studied concerning the potential for waste heat recovery and the available techniques for that. For a relevant result each manufacturing process must be considered individually. In the SSAB case, also the energy flows with lower temperatures thus were considered. Finally, the potentials of low waste heat recovery using a heat pump implementation was suggested.

Production methods of iron and steel are similar in the US and Sweden. Mainly there are two methods in steelmaking:

1- Based on iron ore (Integrated method) using the alternative DRI method
2- Based on scrap (secondary method)

Figure 6 and 7 illustrates the iron & steel production in the US, similar to Sweden [12].

Integrated method:

Limestone, coke, iron ore are feed to a blast furnace. The high temperature in the blast furnace causes the iron oxide in the ore to convert into iron while carbon from the coke merges with oxygen in carbon dioxide. The liquid iron is conveyed to a BOF (Basic Oxygen Furnace) converter, which causes a reduction in the carbon content and conversion of iron into steel. This is done by blowing high pressure oxygen onto the molten iron's surface. Thereafter the crude steel can be processed further to e.g. alloys production.

Secondary method:

Scrap is fed to an electric arc furnace (EAF) and melted. Liquid iron similar to the previous method is conveyed and converted, the difference is the converter using Argon Oxygen Decarburization (AOD).

DRI (direct reduction):

This is a more direct way to reduce iron oxide to iron using a gas mixture of carbon monoxide and hydrogen gas. This gas mixture can be made from coal or natural gas.

Iron and steel industry energy usage in Sweden:

In Sweden the primary energy sources for iron and steel manufacturing are:

- Coke, for blast the furnace or replaced with tar, pulverized coal and oil
- Electricity, in an EAF, running and heat treatment in a rolling mill
- Liquefied petroleum gas (LPG) and oil for heating in a furnace
- Natural gas, in the west of Sweden - used instead of oil [13], [14].

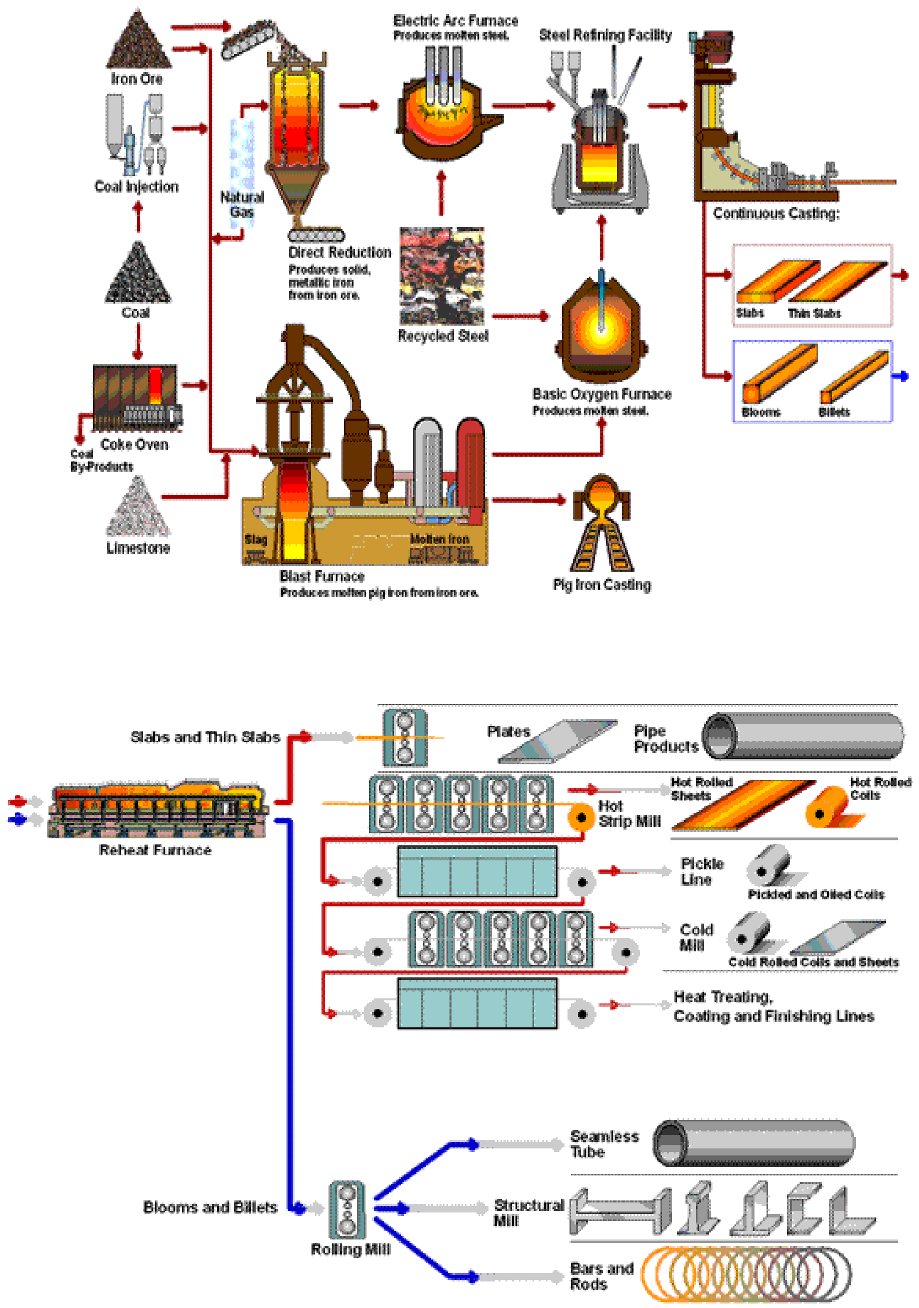

Figure 6: Flows line of iron & steel production and finishing in the US [12]

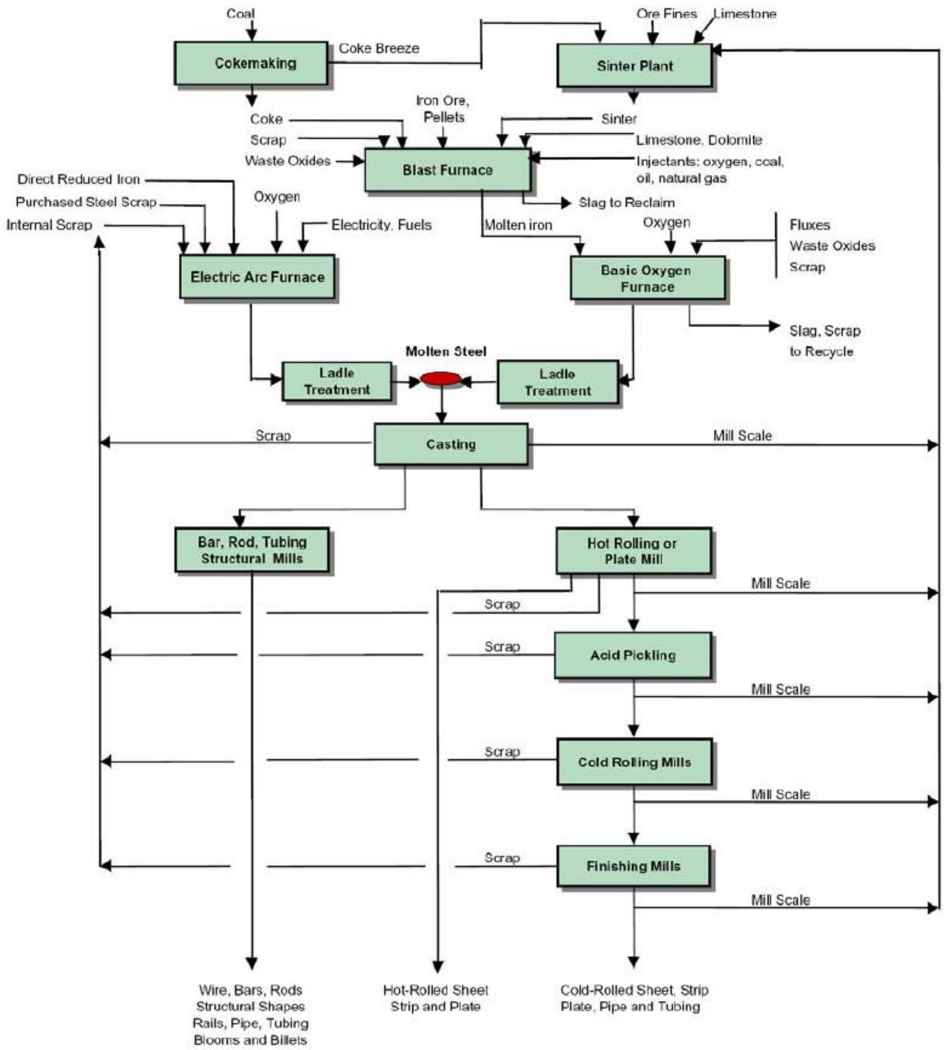

Figure7: Simplified scheme of Iron and steel production and finishing routs in the US [12]

2.1 Waste heat recovery in the US and Sweden

Essential factors for heat recovery are waste heat sources, available equipment and methods of recovery energy.

Two indexes are considered in waste heat recovery:

1- Quality (the exhaust temperature)
2- Quantity (the energy in the waste heat) - a function of mass flow rate, consumption and temperature

In the report ”Waste heat recovery: technology and opportunity in US industry “ reference [15], the temperature in the waste heat sources are classified into three temperature ranges considering industrial processes:

- High temperature T> 650 ˚C

- Medium temperature 230 ˚C <T> 650 ˚C

- Low temperature T< 230 ˚C

60% of the total - unrecovered heat - belongs to the low temperature. Low temperature heat exchange like economizers, indirect contact condensation heat recovery, direct contact heat recovery and heat pumps can be used for recovering low temperature heat. Economizers can be designed to cool down exhaust gases down to 70 °C to also resisting corrosion from acidic condensate leaving its surface. Indirect contact condensation is even used to cool down gases down to 40 °C with a shell and tube heat exchanger. Although these technologies are proper for low temperature waste heat recovery in industry, they are often costly and economic restrictions cause limitations for their commercialization [15].

Many different methods are used for heat recovery in the iron and steel industry and there are still many unrecovered waste heat flows.

Unrecovered and recovered waste heat from gases:

Although waste heat recovery techniques from dirty gases are available the economy an obstacle which must be addressed. Table 1 illustrates unrecovered and recovered waste heat potential from the exhaust gases. In the iron and steel industry gases are most frequent from exhaust the blast furnace (BF), Arc furnace, coke ovens and basic oxygen furnaces (BOF) [15].

Table 1: Unrecovered waste heat potential from exhaust gases process in the iron and steel industry (modified) [15]

<table>
<tr><th colspan="3">Sources</th><th>Average Exhaust Temperature °C</th></tr>
<tr><td rowspan="2">Coke Oven</td><td colspan="2">Coke Oven gas</td><td>980</td></tr>
<tr><td colspan="2">Coke Oven Waste Gas</td><td>200</td></tr>
<tr><td rowspan="3">Blast Furnace</td><td colspan="2">Blast Furnace Gas</td><td>430</td></tr>
<tr><td rowspan="2">Blast Stove Exhaust</td><td>Without Recovery</td><td>250</td></tr>
<tr><td>With Recovery</td><td>130</td></tr>
<tr><td colspan="3">Basic Oxygen Furnace</td><td>1700</td></tr>
<tr><td rowspan="2">Electric Arc Furnace</td><td rowspan="2">BOF Gas</td><td>Without Recovery</td><td>1200</td></tr>
<tr><td>With Recovery</td><td>204</td></tr>
</table>

Heat recovery from solid streams or solids products:

For example, for coke it is possible to recover sensible heat from the coke passing gas, such as nitrogen over it (Coke Dry Quenching CDQ), and then to e.g. transfer the heat to a boiler. Another way is to use a radiative boiler for other solid flows. These methods are used in Germany and Japan. It is also possible to recover the sensible heat using water cooling from hot rolled steel. This method usually results in a final water temperature of around 80 °C that can be used e.g. by a heat pump or directly [15].

Table 2 illustrates the waste heat losses in solid products and byproducts. It includes hot coke, slag from the BF (Blast Furnace), BOF (Basic Oxygen Furnace) and hot rolled and cast steel.

Table 2: Waste heat losses in solid products & byproducts (modified) [15]

Waste heat source	Maximum temperature °C	Heat recovery technique	status
Hot Coke	1100	Dry coke quenching	Commercial, not widely used in US
BF slag	1300	Radiant heat boiler (RHB)	Prototype, R & D stopped since end of 1980s
BOF slag	1500	RHB	Prototype, R & D stopped since end of 1980s
Cast Steel	1600	-RHB with heat pipe -slab cooler boiler -hot charging	RHBs are commercial, but not used in US. Hot charging is used for a small percentage of production.
Hot rolled steel	900	(cools down with water spraying), heat pump *	Commercial, not widely used in US

* Example of heat a pump heat recovery from cooling down water from a strip mill:
Spraying water on the rolled steel, cools it down to 80 °C. An absorption heat pump is used to generate low pressure steam 1.7-3.5 bar, 130°C and deliver it to the grid. This saves fuel and decreases electricity consumption. (Corus IJmuiden, Netherlands) [13],[15].

Water utilization in Iron and steel:

Huge amounts of water is used, but also circulated, in the iron and steel industry for many different purposes like washing and cooling.

For cooling, water is used directly e.g. when cooling of slabs, rollers, bearings, blooms and billets and as a result there are warm but contaminated flows that can be recirculated after cleaning. Cooling towers are used to cool down the water temperature coming from e.g. blast furnaces, sintering, hot forming, vacuum degassing , EAF (direct and indirect water cooling) and hot rolled steel mills (water spray) .

Water can be recirculated as one water cooling system for all this sections being cooled down in cooling towers. Usually the inlet water temperatures to the cooling towers is around 60 °C (less than 100 °C) and they are cooled down to around 30 °C, depending on process. The quantity of heat is varying greatly [15], [16].

Waste heat recovery potential and methods in Sweden:

Waste heat recovery in by-products from the iron and steel industry in Sweden is under consideration but usage of low temperature waste heat recovery can be an alternative to quickly decrease primary energy consumption and CO_2 emissions in society. Table 3 shows the potential and the available techniques for waste heat recovery in Sweden's iron and steel industry. Furthermore some lessons are commercially available though they are in small scale and a few methods are recommended [14], [17], [18].

Table 3: Waste heat recovery potential and available techniques in Sweden's iron and steel industry (modified) [14], [17], [18]

Waste heat source	**Availability**	**Temperature range °C**	**Technology**	**Application**
Combustible process gases : COG ,BFG,BOFG *a	Commercially available	250-1700 60(after heat recovery)*b	combined heat and power (CHP) plan Heat pump after heat recovery (proposal)	Electricity district heating steam
BF gas (heat and pressure)	Commercially available	250	TRT Turbine *c	eEectricity
Low temperature excess heat	Proposal	30-300	ORC & Kalina cycle & heat pump *d	Electricity district heating
Infrared radiation of heat generation in operation process : (flue gas and wall heat recovery)	Proposal	1000 – 1800	Thermophotovoltaic (TPV) *e	Electricity
Infrared radiation of heat generation in product : (blast furnace slag &slabs from the continuous casting)	Proposal	1000 – 1800	Thermophotovoltaic (TPV) *e	Electricity
Water cooled systems such as Cooling tower	Proposal	150–300 °C	TEG commercially available in small scale *f	Electricity
	Proposal	55–300	(ORC-commercially available) or heat pump (proposal)	Electricity district heating
	Proposal	25–95	(PCM engine-First customer installation planned in 2013) *g or heat pump	Electricity district heating
Hot rolled mill	Proposal *h	around 80	Heat pump	District heating

***a:** COG=Coke oven gas, BFG= Blast furnace gas, BOFG= Basic oxygen

***b**: Section recovery of waste contaminated gases - Coke Oven

***c**: TRT = (Top Pressure Recovery Turbine), heat and pressure in a gas turbine produces electricity

***d**: ORC = (Organic Rankine Cycle) by usage of organic working fluid to produce electricity

Kalina cycle, usage of a mixture of water and ammonia as working fluid to produce electricity

***e**: TPV= Thermophotovoltaic, electricity production by absorbing radiation in photovoltaic cells

***f**: (TEG) = Thermoelectric generator, the temperature differences between to junctions in a bi-metal circuit produces electricity

***g**: (PCM engine) = phase change materials engines to produce electricity when changing from solid to liquid

***h**: section **"**solid stream waste heat"

2.2 SSAB recovery potential for heat pump utilization

SSAB Company has a crude steel capacity of 6 million tons. It is mainly high strength steel. Production plants are located in Sweden (blast furnaces) and USA (scrap with electric arc furnace). The company is the largest manufacturer in Scandinavia [10], divisions:

- SSAB EMEA – Europe, Middle East, and Africa
- SSAB Americas – North America and Latin America
- SSAB APAC – Asia, Australia, and New Zealand

The crude steel capacity of SSAB EMEA is 3.5 million tons and its products are quenched, tempered plate and advanced high strength steels [10].

The Luleå plant produces steel slabs.
The Oxelösund plant produces heavy plate from steel slabs.
The Borlänge plant produces strip products from steel slabs.

One of the strategic goals of SSAB is the successive reduction of carbon dioxide emissions, e.g. -2% per ton of steel product in 2012 [10]. SSAB EMEA Oxelösund includes two blast furnaces, a coke plant, a rolling mill and facilities such as a lime plant, an oxygen plant and a power plant. The production process, energy flow diagram and information concerning the waste heat sources are summarized in figure 8 and 9 from [11],[19].

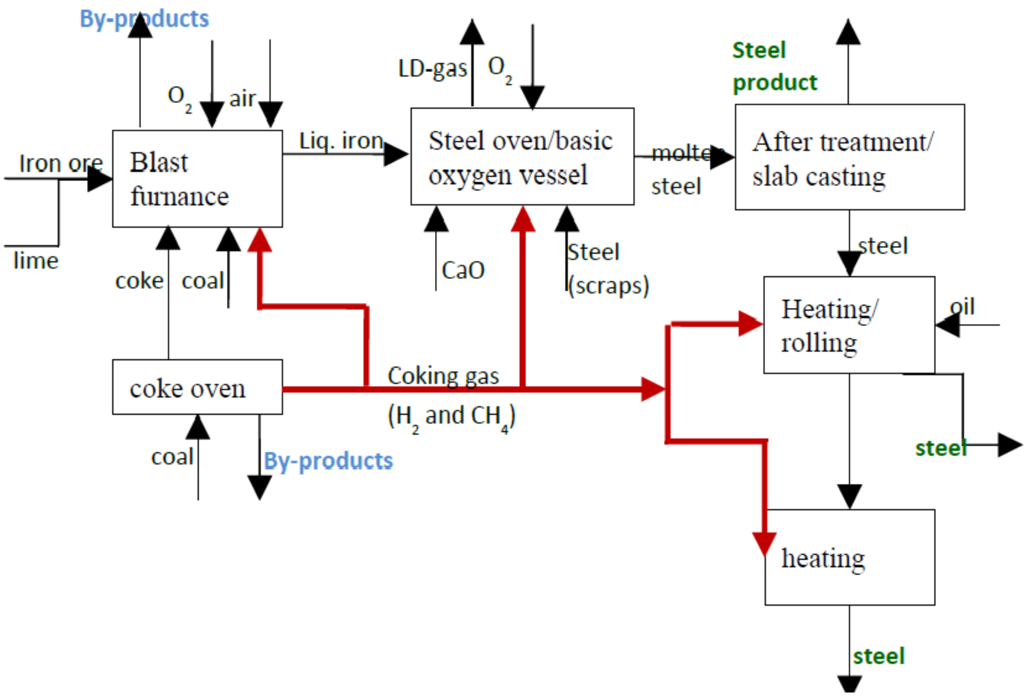

Figure 8: SSAB Oxelösund process with potential of waste heat sources [19]

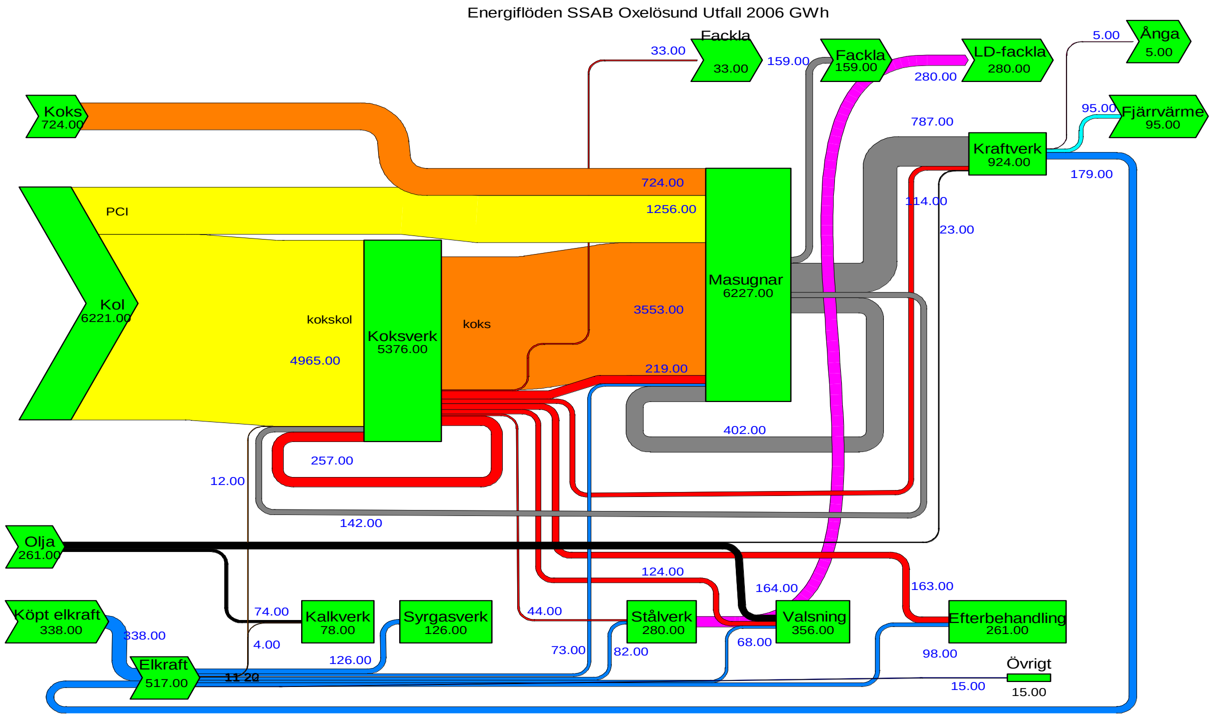

Figure 9: Sankey diagrams for major energy flows of SSAB Oxelösund (GWh) 2006 [19]

Coking plant:

In this part coal is heated whereby volatile hydrocarbons decompose to methane, hydrogen and carbon monoxide. Coal is converted to main product; coke, which is the raw material for the blast furnace and byproducts; like crude benzene, coke oven gas and ammonium soleplate which is used in blast furnace and heat treatment in process (red line in figure 8). The process takes about 18 hours and the temperature is above 1000 °C. The gas is cleaned in several process and the raw materials are recuperated [11], [19].

Blast furnace:

The largest blast furnace is located in Luleå, but there are two smaller blast furnaces in Oxelösund. Pig iron production is 200 tons per hour with a total energy consumption 8.5 TWh/year. The melting reduction process, is the blast furnace process type. Iron oxide is reduced from iron ore into iron. Iron is reduced, melted and collected in the bottom of the furnace. Gas is produced in the process resulting in e.g. 20% CO and a couple % H_2. This gas is utilized for preheating the blast furnace, power generation and district heating . Cooling is needed in the furnace body and in the bottom and pin sockets [11], [19].

LD Process:

In the LD convector, hot metal is converted to steel by reducing the carbon content to below 1.7-1.5 %. This process takes about 20 minutes, and a gas composed of CO and CO_2 is formed. The gas is recovered for energy usage after it is first vented and then cooled and cleaned [11], [19].

Continuous Casting:

In this process, molten steel at about 1600 ºC is cooled into slabs. Molten steel is gathered in a container (tundish) from the bottom of the ladle. The ladle is a container for later transporting or treating molten metal. In the next stage the molten metal is poured into a mould. Water cooling is utilized through the whole casting process from top to bottom of the mould with sea water using a temperature of about 10-15 ºC. An intensive water cooling in the mould side takes place when liquid metal changes to solid-. The temperature of this latent heat is around 1540 ºC. Then the cooling continues by first cooling the string in alcohol-water and then quenching using water pipes over the steel. The water temperature is then around 25ºC. A mixture of steam and gas leaves as a result. The temperature level decreases here from 1540 ºC to 1000 ºC. The steel is cut to slabs using an oxygen lance at the temperature level 1000 ºC after which it is reduced to 600 ºC [10],[11], [19].

Rolling mill:

There are six heat treatment furnaces for processes like rolling, hardening and annealing. They help to enhance hardness, strength and toughness [10],[11],[19].

All waste heat sources are surveyed and summarized with energy rate, temperature, process duration and other indexes in table 4-1 and table 4-2.

Table 4-1 illustrates the cooling of the media in each process of SSAB Oxelösund. It is divided into sections, sorted after the type of medium. The temperature range of the medium before and after cooling is given together with the energy potential per year [11].

Table 4-1: SSAB Oxelösund waste heat sources (modified) [10],[11],[19]

Process	Temperature range [°C]	Medium	Tempe. of inlet medium	Temp. of outlet medium	Energy (GWh /year)	Process duration (hrs/day)	process duration (hrs/ year)
Cooling of gas stream							
Flue gas, coking oven	280 → 150	gas	280	150	52	24	8760
Flue gas(fumes), preheating of air, blast furnace	250 → 150	gas	250	150	64	24	8000
Flue gas after treatment, N2	550 → 150	gas	550	150	24	24	8000
Flue gas after treatment, N1	400 → 150	gas	-	-	6	-	-
Flue gas, blast furnace 1+2	350 → 150	gas	-	-	26	-	-
Flue gas mill, N7/8	250 → 150	gas	-	-	2,7	-	-
Water cooled system							
Blast furnace	45 → 35	Water	45	35	187	24	8100
Steel furnace cooling	150→110	Water	150	110	150	20	6500
Cooling of steel slabs							
Cooling after casting	1000 → 75	air	1000	75	268	-	-
Rolling	1200 → 75	air	1200	75	140	-	-
Cooling of slag							
From blast furnace	1465 → 75	air	1465	75	65	-	-
From steel oven	1550 → 75	air	1550	75	48	-	-
Other							
LD-gas	800 before cleaning and 65 before flaring	gas	-	-	300	(24) 20 min/h	-
Quenching of coke	100	Water	40	100 (steam+air)	140	8	-
Cooling, LD gas in the venturi scrubber	100	-	-	100	163	-	-
Spirits Cooling, continuous casting	< 100	Water spray	25	< 100 Steam /air mix	250	12	-

Table 4-2: SSAB Oxelösund waste heat sources [10],[11],[19]

Process	Note
Cooling of gas stream	
Flue gas, coking oven	Flue gas from combustion of BFG/COG in coking ovens are emitted to atmosphere from three 70m height stacks and have 800m distance from Blast furnace. Flow rate: 110000 m^3/h
Flue gas(fumes), preheating of air, blast furnace	Flue gas from combustion of BFG/COG in hot stoves is emitted to atmosphere from two stacks and has 400m distance from Blast furnace
Flue gas after treatment, N2	Flue gas from combustion of COG in 6 heat treatment furnaces are emitted to atmosphere from three stacks ,2km from blast furnace
Flue gas after treatment, N1	-
Flue gas, blast furnace 1+2	-
Flue gas mill, N7/8	-
Water cooled system	
Blast furnace	Water is cooled in cooling tower and process duration is 24 hours per day .Flow rate is about 1500- 2000 m3/h
Steel furnace cooling	District heat sink is only 150 GWh/year and the source is intermittent
Cooling of steel slabs	
Cooling after casting	Air cooling of slabs from 1000 C
Rolling	Air cooling of plates from 800 C
Cooling of slag	
From blast furnace	BF Slag at 1450 C is deposited on ground
From steel oven	LD/BOF Slag at 1550 C is deposited on ground
Other	
LD-gas	LD gas is another name for BOF gas which is produced when oxygen blow into the melted iron. The gas consists of about 50%CO, 15%CO2, 10%N2, 25%H2O. Combustible and gas Heating value 6 MJ/M^3. Bath type, 20 min/h. LD gas flared in 70m flare stack and not possible to use.
Quenching of coke	10ton of Coke is quenched from 1100 to 50 C in a tower. Hot gas convert water to Steam, 1 bar, batch production everyday
Cooling, LD gas in the venturi scrubber	Batch steam
Spirits Cooling, continuous casting	Cooling during casting, mixture of air and steam emitted to atmosphere

2.3 Nyköping and Oxelösund heat demand

There are three problem which were proposed from SSAB company:

- Cover district heating demand in Oxelösund using waste heat recovery from SSAB
- Cover district heating demand in Nyköping including a pipe from Nyköping to Oxelösund
- Cover district heat demand in Nyköping even if there is no production in Oxelösund

Calculation and modeling of the first problem was done first, but the priority of the project later shifted to the second and third problem.

2.3.1 Oxelösund heat demand

There is already now a satisfied demand from Oxelösund's district heating. This demand is covered from waste heat sources from e.g. power production. Table 5 illustrates the power- and temperature demands [11].

Table 5: Oxelösund district heating energy demand

Process demand	**Medium**	**Return water temperature (T_r) C**	**Forward water temperature (T_f) C**	**Flow rate (m^3/h)**	**Power demand**	**Note**
District heating (summer)	water	55	80	250	Up to 7 MW	Demand to rise water temperature from 55 to 80 C
District heating (winter)	water	70	110-120	550	Up to 35 MW	Demand to rise water temperature from 70 to 120 C

T_r is temperature of return water from Oxelösund

T_f is temperature of forward water to Oxelösund

2.3.2 Nyköping heat demand

The heat demand of Nyköping is illustrated in table 6. This demand can be partly satisfied by SSAB in Oxelösund [11], [20].

Table 6: Nyköping district heating energy demand

Process demand	Medium	Return water temperature (T_r) C	Forward water temperature (T_f) C	Flow rate (m^3/h)	Power demand	Note
District heating (summer)	water	50	75	1295	Up to 20 MW	Demand to rise water temperature from 50 to 75 C
District heating (winter)	water	50	110	1295	Up to 90 MW	Demand to rise water temp. normally from 50 to 110 C

T_r is temperature of return water from Nyköpimg

T_f is temperature of forward water to Nyköping

2.3.3 Nyköping heat demand when heat from SSAB is not available

The heat from the SSAB plant in Oxelösund could be unavailable for shorter or longer periods. Even then the heat pump in Oxelösund should be able to cover Nyköpings heat demand using sea water as a source.

2.4 Feasibility study and potential for heat pump utilization

A feasibility study on waste heat sources of SSAB Oxelösund iron & steel process has been performed and the possibility of heat pump utilization is considered. The results are summarized in table 7 and most possibilities of a low temperature waste heat recovery using heat pumps are described in this table.

It would be possible to use a low temperature heat source, also in an absorption heat pump, if a high temperature gas or other high temperature heat source was available. The main alternative is to use a cooling tower with low temperature heat recovered using with heat pump.

Table 7: Feasible study of using heat pump from waste heat sources of SSAB Oxelösund plant

Process	Temperature range [°C]	Medium	Potential of using a heat pump
			Cooling of gas stream
Flue gas, coking oven	280 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
Flue gas(fumes), preheating of air, blast furnace	250 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
Flue gas after treatment, N2	550 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
Flue gas after treatment, N1	400 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
Flue gas, blast furnace 1+2	350 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
Flue gas mill, N7/8	250 → 150	gas	Temperature is high but , if it is clean and not used in the other part of heat recovery, possible to use in some part of the absorption or semi open MVR heat pump or directly use as heat recovery
			Water cooled system
Blast furnace	45 →35	Water	It is **possible** for heat recovery as heat source in heat pump, both of mediums are water and have better heat transfer. Cooling tower and district heating are close to each other, so there is no need for long piping.
Steel furnace cooling	150→110	Water	Maybe **possible** in absorption heat pump or use directly as heat recovery or
			Cooling of steel slabs
Cooling after casting	1000 → 75	air	**Yes** it can be possible.
Rolling	1200 → 75	air	**Yes** it can be possible.
			Cooling of slag
From blast furnace	1465 → 75	air	**Yes** ,but it should be considered if there is no erosion and not using as the heat recovery in the system
From steel oven	1550 → 75	air	**Yes** ,but it should be considered if there is no erosion and not using as the heat recovery in the system
			Other
LD-gas	800 - 65	-	**Yes,** possible to use but not continuously .
Quenching of coke	100	Water	**Yes** it can be possible but difficult, because of air/water mix.
Cooling, LD gas in the venturi scrubber	100	-	-
Spirits Cooling, continuous casting	< 100	Water spray	**Yes** it can be possible but difficult, because of air/water mix.

2.4.1 Why using the cooling tower?

Regarding to table 7 there are limitations to recover waste heat from many sources. Gases are mostly polluted and difficult to recover because of erosion, high cost and complexity. Also, Flue gas from combustion of BFG/COG in coking ovens are emitted to atmosphere from three 70 m height stacks and are situated 800 m distance from the Blast furnace. Flue gases from combustion of BFG/COG in hot stoves are emitted to the atmosphere from two stacks at 400 m distance from the blast furnace. Flue gases from the combustion of COG in six heat treatment furnaces are emitted to the atmosphere from three stacks with 2 km distance from the blast furnace.

LD gases, have a high temperature and would be a good source, however the process duration is just 20 (min/h), thus there is a need for a storage to achieve a continuous heat flow. On the other hand it is a good complementary heat flow to other processes.

Some other processes like quenching coke and cooling LD gas, is used in other parts of the plant. Sprits cooling is a mixture of steam and air cooling and therefore difficult to recover. Also, slag and steel furnace cooling are other alternatives but difficult to recover in a process. In general, there is little distance between the cooling tower and the blast furnace thus it would be a good source for heat recovery. This will also help reducing the cooling tower's maintenance costs.

When the cooling tower clogs it is difficult to maintain the efficiency and it also requires a lot of chemicals to avoid legionella. This tower also needs to be partly rebuilt, if its life length is to be prolonged. Electricity within SSAB plant is not so expensive and could help to improve the heat pump economy if that is allowed by the tax laws. If steel production will cease for a longer or shorter period the heat pump could use sea water as source.

In general, to cover the heat demand, high temperature industrial heat pumps can be used as low temperature waste heat recovery from the blast furnace. Sweet and clean water is coming from Nyköpingsån (a river) is used by the blast furnace and is also used in some other internal systems. The water delivered from the blast furnace to the cooling tower is today fluctuating between 40°C to 50°C and the return water temperature fluctuates between 30°C to 40°C. The water flow rate in summer and winter respectively is 1500 (m^3/h) and 2000 (m^3/h). Moreover, the cooling tower is running about 8100 hours per year (not working in July). July can be covered using sea water as heat source.

Maybe at a later stage also other higher temperature sources can help improving the efficiency of the concept.

2.4.2 Heat source (cooling tower) specification

Table 8 illustrated heat source and heat sink temperature with flow rates in summer and in winter. Because of fluctuating water temperatures in the cooling tower, a representative average temperature has been considered [11].

Table 8: heat source specification

Heat source	Average water temperature from blast furnace to cooling tower	**45 ºC**
	Average water temperature delivered from cooling tower to blast furnace	**35º C**
	cooling tower water flow rate in summer	**1500 (m^3/h)**
	cooling tower water flow rate in the rest of year	**2000 (m^3/h)**

3 The high temperature industrial heat pump

3.1 Industrial heat pumps

Their possible temperature range has increased significantly during the last decades. Industrial heat pumps temperature range is today up to over 100 °C and with power capacities ranging from maybe of 50 kW and many MW. This is achieved mainly by the development of new refrigerants.

Industrial heat pumps are implemented for many purpose such as waste heat recovery, air conditioning in industry , district heating, steam production and many other applications. The industrial heat pumps are of course designed to meet their specific needs and the specific conditions. As the conditions differ, the serial length of production is smaller than for e.g. domestic heat pumps. The energy consumption in the industry- in the household- and the service sector are rather equal, figure 2. Industrial heat pumps can sometimes have the following advantages compared to residential heat pumps:

- Higher COP due to a lower temperature span
- Lower investment cost due to a low distance between heat source and heat sink
- Higher duty factor 6000 h/year or more
- Simultaneously use being both heat source and heat sink
- Ability to use cheap waste heat in industry reducing total usage of primary energy and cost
-

Although IHPs thus often have advantages compared to residential heat pumps, the lack of experience, lack of consult IHP-experience in industry causes a lower amount of IHP installations rather than residential heat pumps. Industrial heat pump temperature range is sometimes divided to three levels [5],[21]:

- Medium temperature $T_{\text{heat sink}} < 80$ °C
- High temperature 80 °C $< T_{\text{heat sink}} <$ 140 °C
- Very high temperature $T_{\text{heat sink}} > 140$ °C (higher than 140 C in near future)

3.2 Industrial heat pump applications in general

Industrial heat pumps are able to recover waste heat in industry and make it usable for other industry processes. They depended on matching heat sink and heat source temperatures and capacities. The higher the temperature lift the higher the pressure ratio in the compressor and the lower the COP. In situations with a high temperature lift it is better to use multistage heat pumps.

Table 9 suggests many processes in industry where high temperature heat pumps can be used. The type of heat pump is indicated according to the temperature range [21], [22].

Table 9: Industrial applications with heating loads temperature (modified) [21], [22]

Sector	Process	Temperature range °C	Type of the heat pump
General	preheating	20-110	High temp. HP
	washing	30-90	Medium temp. HP
	district heating	70-120	High temp. HP
	Preheating boiler feed water	26-100	High temp. HP
	Space conditioning of facilities and warehouses	5-100	High temp. HP
	Preheating load	15-315	Very high temp. HP
	Preheating combustion air	315-871	Very high temp. HP
Chemical	bio chemical react	30-55	Medium temp. HP
	distillation	100-200	Very high temp. HP
	compression	110-170	Very high temp. HP
	cooking	85-110	High temp. HP
	thickening	130-140	Very high temp. HP
Food and Beverages	blanching	60-90	Medium temp. HP
	scalding	45-90	Medium temp. HP
	evaporating	40-130	Very high temp. HP
	cooking	70-120	High temp. HP
	pasteurization	60-150	Very high temp. HP
	smoking	20-85	Medium temp. HP
	cleaning	60-90	Medium temp. HP
	sterilization	100-140	Very high temp. HP
	tempering	40-80	Medium temp. HP
	drying	40-200	Very high temp. HP
	washing	35-80	Medium temp. HP
Paper	bleaching	40-150	Very high temp. HP
	de-inking	45-70	Medium temp. HP
	cooking	110-170	Very high temp. HP
	drying	90-200	Very high temp. HP
Rubber and Plastic	drying	45-150	Very high temp. HP
	preheating	45-65	Medium temp. HP
Textiles	bleaching	40-100	High temp. HP
	coloring	40-130	Very high temp. HP
	drying	60-100	High temp. HP
	washing	50-100	High temp. HP
Wood	steaming	75-85	Medium temp. HP
	pickling	40-70	Medium temp. HP
	compression	120-170	Very high temp. HP
	cooking	80-90	Medium temp. HP
	drying	40-155	Very high temp. HP

3.3 Heat pump principle

Mechanical heat pumps, frequently used in industry, using the common refrigeration cycle compressing and expanding a refrigerant thereby absorbing heat from a source and releasing heat to a heat sink. This type of heat pumps has four main parts:

- Evaporator
- Compressor
- Condenser
- Expansion valve

Heat is delivered from a waste heat source to a refrigerant in the evaporator. A good heat source has a steady and high temperature when needed. The refrigerant is compressed in the compressor and the temperature is increased. The refrigerant, with a higher temperature is then enters the condenser where the heat is delivered to the sink, whereby the refrigerant is liquefied, figure 10. After that the liquid refrigerant goes to the expansion device where the refrigerant is expanded and cooled down. Usually the expansion device is just a valve. The added energy needed for the compressor to compress the refrigerant is equal to the difference between the heat given to the sink and the heat absorbed from the source. Finally, the refrigerant is expanded through the expansion valve from the condenser to the evaporator. The circuit is closed. This the general principle for all kind of heat pumps. The $COP_{heating}$ = useful heat to sink / used compressor electricity.

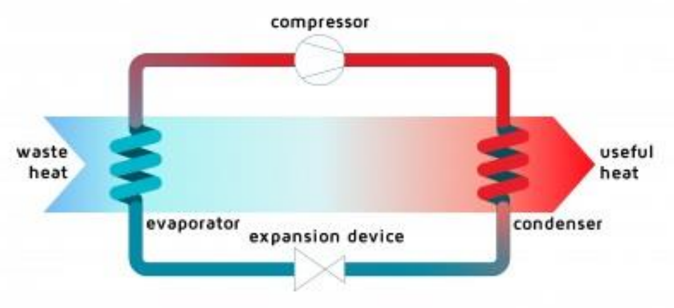

Figure 10: Heat pump principle work [6]

The heat pump efficiency is often measured using the Coefficient Of Performance (COPh). This is for heating the ratio between useful heat given to the sink and the compressor's energy consumption. When cooling, COPc is defined as the heat absorbed from the sink, divided by the electricity consumed by the compressor. According to the second law of thermodynamics, a higher temperature difference between heat source and heat sink, will decrease the COP [6].

3.3.1 Evaporator and condenser

The evaporator is used to transfer heat from the heat source to the refrigerant. The condenser is used to transfer heat from the refrigerant to the heat sink. The refrigerant changes from liquid to gas in the evaporator and in the opposite direction in the condenser. Thereby absorbing or exuding latent heat. The pressures in the evaporator and condenser depend on the boiling curve of the refrigerant. The heat transfer is proportional to the product of the heat exchange area and the heat transfer coefficient. Shell & tube heat exchangers are mostly used in industry for both evaporators and condensers. When using shell and tube evaporators with water there is always a risk of freezing which must be avoided.

In a shell and tube condenser the condensing normally takes part outside the tubes. The refrigerant can also be boiling outside the tubes in an evaporator (normally it is inside). It is much easier to clean the tubes on the inside when the source is polluted. However the volume outside the tubes is normally larger than inside the tubes so the total filling tends to get larger using shell and tube heat exchangers this way. The dimensioning of the condenser and evaporator is an optimization problem. Large surfaces give a high COP but also have a high investment cost. In the specific case of SSAB in Oxelösund another type of evaporator is suggested.

Concerning the two types of shell and tube evaporators with flow inside or outside the tubes. When refrigerant flows inside the tubes and is evaporated and superheated (dry expansion), there is less risk of oil accruing in the evaporator and the refrigerant charge is smaller. In the second case when brine flows inside the tubes and refrigerant boils outside, the oil in the refrigerant must be returned by skimming it of the boiling surface, heating it up and returning it to the suction line at a point where it can reach the compressor. Turbo compressors leak very little oil into the refrigerant (50 ppm). Thus the return of oil can be done even manually with long time intervals. In this later geometry the evaporation does require any following superheating which enhances the COP.

Presently the water (heat source) is originally coming from Nyköpingsån, (a river) which is sweet and is then cleaned in the SSAB plant before it is used in the cooling tower. Thus when using only this water as a heat source a horizontal shell and tube evaporator with refrigerant inside the tube could be recommended. However if also sea water should be used – other forms of heat exchangers would be better.

In shell and tube condenser, the refrigerant is condensed outside the tubes and water (the heat sink) flows inside the pipes. A typical horizontal shell and tube condenser is shown in figure 11. It is possible to sub-cool the liquid in a shell and tube heat exchanger slightly, if the inlet tubes from the sink are first passing through a liquid pool of refrigerant at the bottom [23], [24]. Often subcooling is however performed in a special heat exchanger after the condenser.

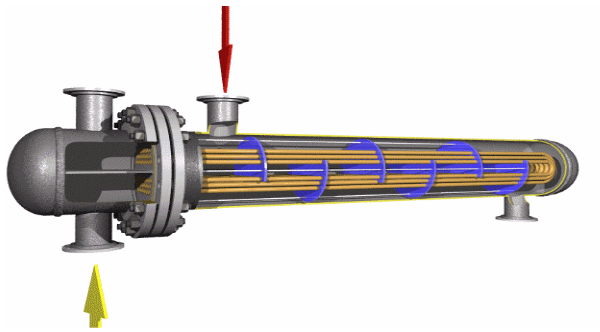

Figure 11: horizontal shell and tube condenser [25]

3.3.2 Compressor

Both the pressure and temperature of the working fluid increases in the compressor. Some compressor types can accept a wet inlet – a small fraction of liquid entrained in the gas. Other compressor types require a dry inlet (only pure gas). Most compressors require some oil-lubrication. Oil free types are much more expensive. The oil is entrained in the refrigerant. Turbo compressors require only a small amount of oil, but cannot accept drops, whereas screw compressors require a lot of oil and can accept drops. Compressors are classified as dynamic compression or positive displacement compressors. Both these two compressor types have several subsystems implemented and yield different characteristic data (figure 13).

In positive displacement compressors, like reciprocating- and screw compressors the fluid pressure increases due to reduction of its gas volume. A built in pressure ratio of the compressor can be a result of the physical design. The compressor should then be used around this built in pressure ratio. Reciprocating compressors do not have a built in pressure ratio. Normally it is also possible to achieve higher temperature lift with positive displacement, than with dynamic compressors, though the volumetric flow rate is normally lower than for dynamic compressors. Reciprocating compressors are like combustion engines working with valves and often piston rings [23]. The pressure is pulsating. A tank on the pressure side can smooth out this and a larger rotating balancing mass or a flywheel can smooth out vibrations. Positive displacement compressors often need more maintenance, than dynamic compressors due to wear and tear.

Screw compressor can achieve a high pressure ratio and rather high volumetric flow rate. Then for very high capacities many parallel screw compressor would be needed. Screw compressors have advantages compared to both dynamic and reciprocating compressors, but are not proper for very high capacities, and their necessary auxiliary components are expensive.

They are also able to work in high temperature machines. Some of them can vary their built in pressure ratio and most of them can easily vary their capacity using a built in sliding piston or the rotational speed. Normally oil is used for sealant and inner lubrication. They are also less sensitive to and wet compression and are often most cost effective when the shaft power is less than around 1 MW [23].

In different screw compressors, the volume (and pressure-) ratio can thus either be fixed or adjusted. Compressors with a fixed volume ratio are less energy efficient outside this ratio but have lower capital and maintenance costs and higher durability than variable volume ratio compressors have. Figure 12 illustrates side view of screw compressor [23].

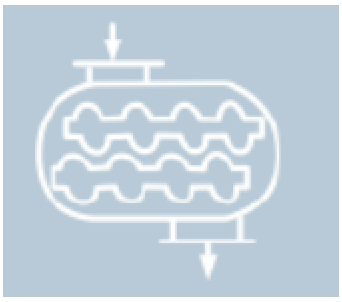

Figure 12: Screw compressor (side view) [23]

In radial dynamic compressors, an impeller wheel sets the gas molecules into motion. The kinetic energy of the molecules is then converted into pressure-, using a diffuser. Dynamic compressors have a large volumetric flow rate, but limitation in pressure ratio. For industrial applications where higher pressure ratios are needed multistage systems can be used. Therefore, most dynamic pressure heat pumps are placed in the lower-right corner of figure 13 (except multistage centrifugal types). Though multistage systems are used in industry, the type of refrigerant also sets limitations for their use. They are not proper for low-molecular-weight refrigerants. Dynamic compressors are thus proper for large capacities with limited pressure ratios. They are also called Turbo compressors and can be divided into the axial- and radial types depending on the direction of the gas flow through the impeller. Radial compressors, also called centrifugal compressors, have higher pressure ratios than the axial types. Generally they are not proper for low flow [23]. Centrifugal compressors capacity are normally controlled by guide vanes or impeller speed variations. Pressure surging can be occurred if the pressure ratio gets too high.

The main advantages of dynamic compressors are thus a small and compact package a lubricant that not interact necessarily blends with the refrigerant and that the maintenance is less costly (only one moving part). A disadvantage is that several stages are needed to reach a high pressure ratio.

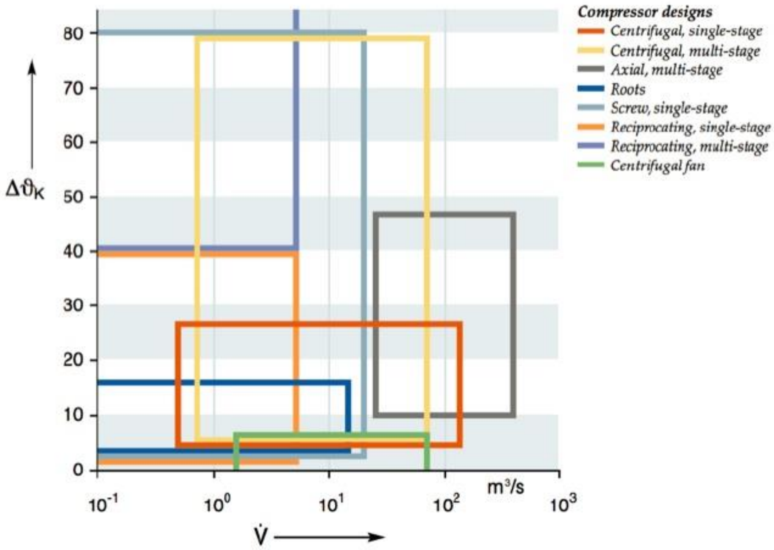

Figure 13: Functional ranges of compressor systems, $\Delta\vartheta_K$ is in this case the increased condensation temperature of water vapor at an initial state of 1 bar, 100 °C, $\dot{V}$ the vapor mass flow rate [23]

(Adapted from "Gaute Glomlien, High temperature heat pump for industrial applications".2013)

3.3.3 The expansion valve

Expansion valves decrease the pressure of the working fluid after the condenser (or sub-cooler) to that of the evaporator. The fluid flow rate to the evaporator is thus controlled by this valve. Ideally there is no energy transfer to the environment and the enthalpy in the inlet and the outlet is the same. A thermostatic expansion valve controls the temperature of the superheated refrigerant from the evaporator to the compressor [23]. There are other valve-types, especially in larger machines, controlled by the liquid refrigerant level in a tank on e.g. the low pressure side.

3.4 Many types of industrial heat pumps..

Many other types of heat pumps can be used in industry and they can be categorized using various criteria [6], [7],[21], [26]

- closed cycle compression heat pump: diesel and electric motor driven (CCC)
- mechanical vapor recompression (MVR)
- thermal vapor recompression (TVR)
- absorption cycle : absorption heat pumps & heat transformers
- hybrid heat pump
- Transcritical CO2 heat pumps

3.5 A high temperature heat pump for SSAB

Figure 14 illustrates different types of heat pumps classified after the heat source temperature and heat sink temperature. Using the SSAB waste heat temperatures as heat source in table 5, 6 and 8, another type of heat pump could be selected [27]. This is just shown in order to grasp the generality of the heat pump technology concept - and later we will return back to “normal” vapor compression heat pumps.

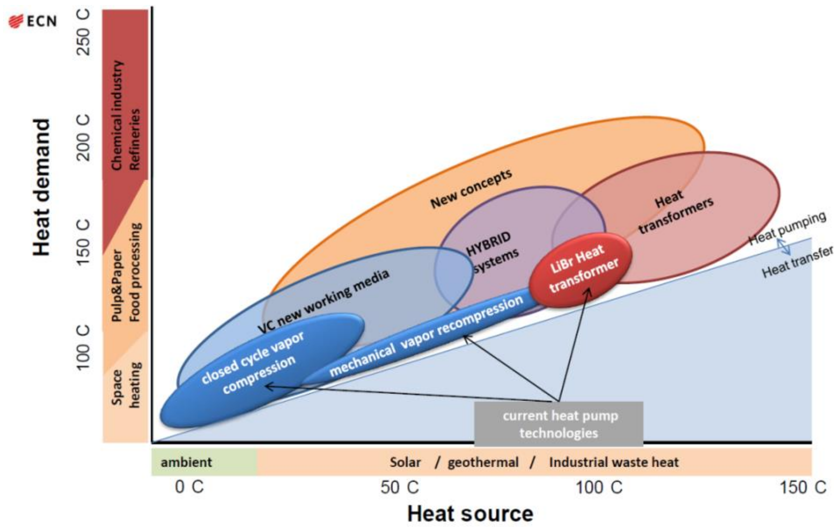

Figure 14: Schema of heat pump types in industry [27]

This summary shows various industrial heat pump types and their properties, like temperature lifts and sink temperatures as illustrated in table 10. Regarding the SSAB demand, possibilities of heat pump utilization with available energy sources are considered.

Thereafter a closed cycle compression heat pump with an electric motor and proper refrigerant suggested to cover 75°C - 120 °C heat sink demand with 35°C - 45 °C heat source temperature, is discussed [6],[7],[21],[26],[28].

Table 10: Feasible study on type of heat pump for SSAB cooling tower heat recovery

Industrial Heat pump type	Properties	Max sink Temp.	Max Lift Tempe.	Statues
MVR (Mechanical vapor compression water steam)	steam temperature more than 80 °C as heat source, also uses a mechanical compressor with electricity or combustion engine	190	90	If possible to use steam energy (SSAB waste heat, table 4-1) as heat source, then it could be proper to be used for waste heat recovery.
TVR (Thermal vapor recompression)	high pressure steam needed and need to have a high temperature waste gas /very low in maintenance because there are few moving parts	150	40	It would be possible to use high pressure steam (e.g. SSAB power plant) and high temperature heat sources gas (table 4-1)
Electric motor (CCC) (closed cycle compression heat pump)	closed cycle/no need of steam or high temperature heat source directly/use refrigerant /driven compressor by electricity for higher temperature/proper for separated heat source and heat sink / good for high temperature lift	120	80	Proper for heat recovery of cooling tower (SSAB waste heat, table 4-1). This is the most probable way for Nyköping.
Diesel motor (CCC)	It is like electric motor CCC but using a use diesel-engine	130	90	Proper for heat recovery of cooling tower using a diesel driven compressor.
Absorption (Lib/H2O)	driven by heat /complex/high investment cost/ COP is around 1,5	100	50	Not proper/driven by heat, but possible to use waste gas energy available to use (SSAB waste heat, table 4-1) and low waste heat as heat source.
Absorption (Heat transfer) (Lib/H2O)	driven by heat in medium temperature/low COP is around 0,5	150	60	Not proper, driven by heat/low COP, but possible if waste gas energy available to use (SSAB waste heat, table 4-1) and low waste heat as heat source.
Hybrid HP (Compression resorption heat pump) (ammonia & water)	Capacity of 250 KW to 1.5 MW/pressure below 25 bar/waste recovery from 15 to 60 C/uses natural working fluids (ammonia and water) and is based on absorption, compressor system.	110	50	Proper but complex and expensive and only 1.5 MW capacity/ used for district heating with max 110°C.
Transcritical CO_2 (Electric motor (CCC) with CO_2)	Low critical temperature /useful for low temperature heat source/ temperature lift depend on the limitations.	90	≈ 70	Not proper for heat recovery as high temperature heat pump/good for heat recovery of low temperature heat source and good for heating e.g. tap water tested to produce water up to 90 C without any problem [7].

4 Refrigerant

Closed- cycle compression (Vapor Compression) heat pumps need a working fluid to absorb, transfer and discharge heat from a low temperature heat source to a higher temperature heat sink. This is done manly by latent energy changes - from liquid to vapor and vice versa in a close cycle. All refrigerants are different and should be chosen specifically for each application. A good refrigerant also has low impact on the ozone layer (ODP) low greenhouse warming potential (GWP), low flammability, low toxicity, low corrosion on mechanical components, enabling a good oil choice and most important - proper thermodynamic properties [29].

4.1 Environmental indicators:

Below is a short explanation of some of the factors above:

ODP (Ozone Depletion Potential): comparison of ozone depletion using refrigerant and R11, (Trichloroflouromethane, ODP=1 as reference)

GWP (Global Warming Potential): relation between the greenhouse effect of the refrigerant and CO2 per kilo released into the atmosphere.

TEWI (Total Equivalent Warming Impact): there is a direct and an indirect share of greenhouse gases in heat pump cycle. The direct part comes from refrigerant leakage and the indirect part comes from the required energy running the cycle taking the carbon dioxide emissions from the electricity production into account. The sum of these both emissions is called "Total equivalent warming impact" [29].

4.2 Types of the refrigerants:

Table 11 illustrated old refrigerants replaced during the 90s':

Table 11: Traditional refrigerants (modified) [29]

Name	Temperature rang application	Max temperature (C˚)
CFC-12	Low- and medium temperature	80
CFC-114	High temperature	120
R-500	Medium temperature	80
R-502	Low-medium temperature	55
HCFC-22	Virtually all reversible and low-temperature heat pumps	55

4.2.1 CFCs

This group has high ODP and are currently banned. This group includes of R-11, R-12, R-13, R-113, R-114, R-115, R-500, R-502, and R-13B1 [29] [6].

4.2.2 HCFCs (hydrochlorofluorocarbons)

This group was first be used to retrofit CFCs. They have lower ODP (2-5% of CFC-12) and GWP (20% of CFC-12) compared to the CFCs. This group includes of H-CFCs contain R-22, R-401, R-402, R-403, R-408 and R-409 [6] [29].

4.2.3 HFCs (hydrofluorocarbons):

This group does not effect on ozone depletion at all and have been replacing the HCFCs. They have still an effect on global warming though – and many of them, especially R404a, are therefore going to be replaced in the future. This group includes among others R-134a, R-152a, R-32, R-125 and R-507 [6],[29].

4.2.3.1 HFC-134a

It is similar to CFC-12 in thermophysical specification. This refrigerant is used in medium and large heat pump systems. Its efficiency is normally better than both common refrigerants R407c and R410a but lower than NH_3. The pressure in this refrigerant is low, thus there is a need for larger swept volume by the compressor and thus more investment needed in the compressor.

4.2.3.2 HFC-152a

This refrigerant was mainly used before as part of R-500 and is now used as component in other blends. Its slight flammability is still too large for the A3L group which means it is considered flammable – A3. Although it is a very good refrigerant, it is costly to use it due to that it is considered an A3 refrigerant. The GWP is lower than 140.

4.2.3.3 HFC-32

It is slightly flammable and it has low GWP.

4.2.3.4 HFC-125 and HFC-134a

Similar to R-502 and HCFC-22.They implement in ternary mixture change with R-502 and HCFC-22.

4.2.4 Blends

A blend is a mixture of two or more pure working fluids and such blends have replaced many CFCs and HCFCs. They can be zeotropic, azeotropic or near-azeotropic. In an azeotropic mixture, evaporation and condensation occur under constant temperature but for the zeotropic mixtures condensation an evaporation happen under a varying temperature - a temperature glide. A temperature glide can actually help increasing performance, but it needs understanding processes and slight modification of the equipment. This type of refrigerant can be custom made and is very suitable sometimes. R-22, can be replaced with two blend fluids named R-410A and R407-C. Thermal specification and performance of R-407C is very close to R-22. Although the usage this type of refrigerant is increasing, there are still some engineering problems especially during maintenance – which part leaked out e.g.? R-410A is popular in USA and Japan and this refrigerant is able to increase COP for low condensing temperatures and also reduce the compressor size and thereby overall cost compared to R-22 [29], [6].

4.2.5 Natural working fluids

These refrigerants have are zero both ODP and GWP. They are naturally existing fluids in our environment thus they are seen as real long term alternatives to CFCs. Often though the safety rules are very strict. This applies to both flammable and toxic natural fluid types [6], [29].

4.2.6 Ammonia (NH3)

Ammonia is both slightly flammable and toxic. A very slight amount can though be used without any restrictions. Also it is a good alternative for HCFC-22 because of its thermodynamic character. With a high pressure compressor, it can reach a rather high condensation temperature range (slightly below 80°C.

4.2.7 CO_2

It is non-toxic and non-flammable refrigerant which is well-suited using normal lubrication and other common construction materials. It has an extremely high volumetric capacity and also a low pressure ratio. Because of it's operation near and over the critical point, it is difficult to reach a very high COP. However CO2 is very a very well suited refrigerant for heating a medium with a large temperature span – like tap water. The tap water enters with maybe 10 °C and exits with 65 °C e.g. [29].

4.2.8 Water

It is neither flammable nor toxic. It is mainly used in open and semi-open MVR and in some closed-cycle compression heat pumps as working fluid. The operation temperature range spans from around 80 °C to 150 °C. However the volumetric heating capacity is low – thus a large compressor is required [29].

4.2.9 Hydrocarbons

They are all flammable but have normally good thermodynamic data and material compatibility. They are already widely used in the petroleum industry. In the refineries the flammability characteristic and how to deal with that risk is well known [29].

4.3 How to choose the proper refrigerant - compilation

A proper refrigerant should preferably have data as below:

- Non flammable
- Non global warming potential (GWP)
- Non toxic
- Easy leak detection
- Highly stable in chemical and thermal characteristic
- Compatibility with material and miscibility with lubricants
- Low cost
- Related to the application has proper critical and boiling point temperature and compressor (molecular weight)
- Low viscosity
- High thermal conductivity

- Proper pressure range related to the application
- No ozone depletion (ODP)

Pressure: The saturation pressure curves vary a lot between refrigerants. A higher condensation temperature causes higher pressure in all refrigerants and all components have to be able to withstand the increased pressure. Low pressure refrigerants on the other hand increases the need for swept volume in the compressor, this means a have higher investment cost.

Critical point: When the temperature or pressure exceeds the critical point the supercritical area is reached. In this area it is not possible to distinguish the refrigerants gaseous- and liquid phase.

Energy efficiency of the heat pump: depends on the system design, on the heat pump and the refrigerant type [6], [30].

4.4 Refrigerant for high temperature heat pump

Figure 15 shows the temperature of evaporation versus the pressure range for different refrigerants. Water has a low pressure and low density in vapor form. Therefore a high compressor capacity is needed. The second lowest pressure option is R245fa, the third one is R600 and the fourth is R134a [6].

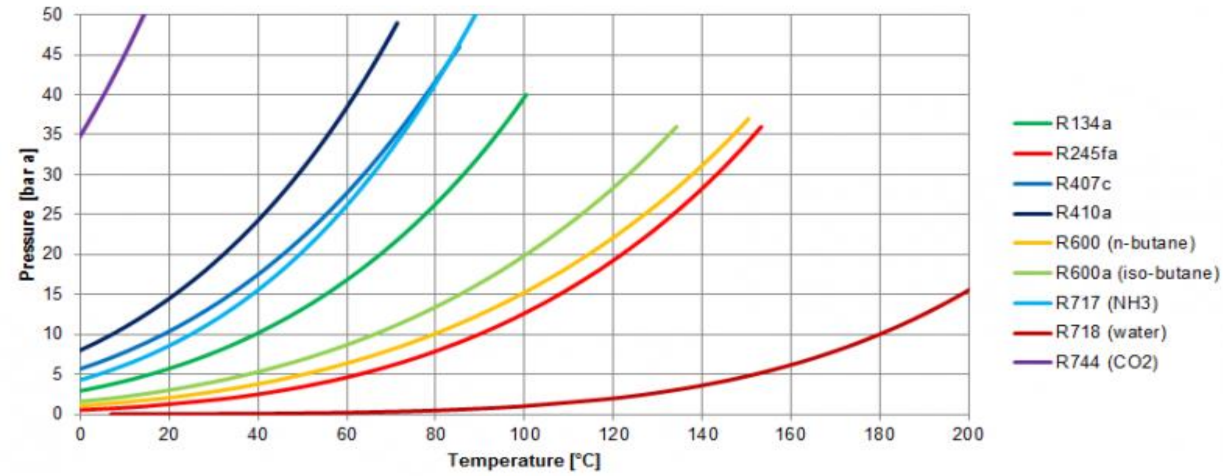

Figure 15: evaporator temperature vs pressure in refrigerants [6]

Table 12 illustrates some refrigerant specifications for refrigerants that have been used in high temperature industrial heat pumps [31], [32], [33], [34].

Table 12: Properties of high temperature refrigerants

Refrigerant	R717 (ammonia)	R600a	R600	R114	R134a	R245fa	R1234ze(Z)	R32	R1234ze(E)
safety class	B2	A3	A3	A1	A1	B1	A2L	A2L	A2L
Flammable (1-2-3)	yes	yes	yes	no	no	non	yes (very low)	-	yes(above 30C with air)
toxic(A-B)	yes	no	no	no	no	yes(very low)	no	-	no
ODP	0	0	0	0.85	0	0	0	0	0
GWP100	<1	8	8	9200	1430	858	<1	8.7	<1
Tb [°C], 1 bar	-33	-12	0	3.6	-26	15.1	9.8	-51	-18.95
Tcr [°C]	133	135	152	146	101	154	150.1	78.4	109.4
Pcr [MPa]	11.4	3.6	3.7	3.2	4	3.65	3.53	5.3	3.64
molecular weight (g/mol)	17	58	58	170.9	102	134	114	52	114
MAX pressure ratio for turbo comp.	1.2	1.7	1.7	4.7	2.6	3.7	3.1	1.7	3.1

R717, although ODP is zero and GWP is low the flammability, toxicity, critical temperature and pressure range is not proper for a high temperature heat pump even using e.g. a screw compressor (it needs too high pressure to reach a high temperature). Because of low the molecular weight it is not proper for turbo compressors (due to the low pressure ratio per stage)

R600 (butane) and **R600a** (isobutene) are both suitable to be used in a high temperature heat pump with a low pressure increase, but they are both flammable and installations should follow the safety NPR-7600 code or similar.

R1234ze(z) belongs to the hydrofluororolefins (HFOs) for high temperature applications which has been considered in the studies and our modeling. They would be proper replacement for **R114** which is banned for usage. The critical temperature of this refrigerant is 150 ˚C and the heat transfer coefficients are great. It also has a high molecular weight suitable for turbo compressors [34].

R1234ze(e) is another isomer of R1234ze(z) but with lower critical and boiling temperature (depends of application usage)

R134a has a high GWP and a low critical temperature and is not proper for, at least, very high temperature heat pumps.

R32 also has a low critical temperature and lower molecular weight, which is not proper for high temperature heat pump neither with screw nor turbo compressor.

R245fa is another viable option for a high temperature heat pump. The critical temperature of R245fa is 154 ºC and can be condensed around 125 ºC or higher. It has though a high GWP compared to R123ze.

The thermodynamic characteristics and COP concerning R1234ze(z) and R245fa are similar and the COP is higher than for other refrigerants.

4.5 Refrigerants suitable for District heating heat pumps

In district heating process, normally a large capacity heat pump is utilized running with refrigerants R134a or sometimes ammonia. The refrigerants should have zero ODP and a very low GWP because of environment demands.

R134a has a rater low critical temperature and needs subcooling at least to reach to the ammonia COP. Normal design in Sweden includes a two stage turbo compressor for to reach the desired condensing temperature within 30 or 40 bars maximum pressure. Shell and tube evaporators and condensers are used mostly. The outlet water temperature is always under 90 ˚C. The refrigerant GWP is 1430 and a typical high capacity heat pump with turbo compressor has a leakage around 1%.

R717 (Ammonia) normally utilizes a twin screw compressor with a maximum pressure about 40 to 50 bars. Mono screw compressors can even reach pressures around 75 bars. Then shell and tube heat exchanger is mostly utilized in the system. Furthermore the GWP of Ammonia is less than 1 which is very environment friendly. The outlet water temperature is almost always below 90 °C even in high pressure systems.

R744 (CO_2) has been used as the working fluid in district heat pump but yields almost always a lower COP of the system. Maximum outlet temperature today is about 80 ºC [35].

4.6 Result

Table 13 illustrates the specifications of some high temperature refrigerants and the feasibility to use them while recovering the heat from SSAB's cooling tower using a high temperature (electrical CCC) heat pump. **R114** is banned because of high GWP. **Ammonia** cannot be used by turbo compressors due to the low molecular weight. Refrigerants **R32** and **R134a** have comparatively low critical temperatures. **R245fa** and **R1234ze(Z)** are proper fluid for the high temperature heat pump with a high COP [31], [32], [33], [34], [35]. R1234ze(**E**) has a much lower critical point and is less favorable as compared to R1234ze(**Z**).

Table 13: High temperature refrigerants and their feasibility in the SSAB heat recovery by heat pump

Refrigerant	Feasible study of utilization in HTHP
R717 Ammonia (NH3)	**Maybe proper** Tcr is low, maximum outlet temperature is high/need high pressure in screw compressor/should be checked
R114	**Not proper** high GWP , it is banned
R134a	**Not proper** , Tcr is low, and high GWP
R245fa	**Proper** High Tcr/low pressure in high condensation temperature /but has higher GWP/heavy molecular weight proper for Turbo and screw compressor
R1234ze(Z)	**Proper** High Tcr/low pressure in high condensation temperature /low GWP/ proper molecular weight, proper for Turbo and screw compressor
R34	**Not proper** low molecular weight and low Tcr
R1234ze(E)	**Not Proper** High Tcr/low pressure in high condensation temperature /low GWP/ proper molecular weight/lower Tcr than R1234ze(Z), proper though for Turbo and screw compressor in other cases
R600	**Not Proper** High Tcr/low pressure in high condensation temperature **but flammable** and low molecular weight, good only for screw compressors
R600a	**Not Proper** High Tcr/low pressure and high condensation temperature **but flammable** and low molecular weight, good only for screw compressor

5 Simulation

5.1 Modeling description

An industrial heat pump was modeled in the EES software and simulated to evaluate the heat pump performance. The work flow is illustrated in figure 16 below:

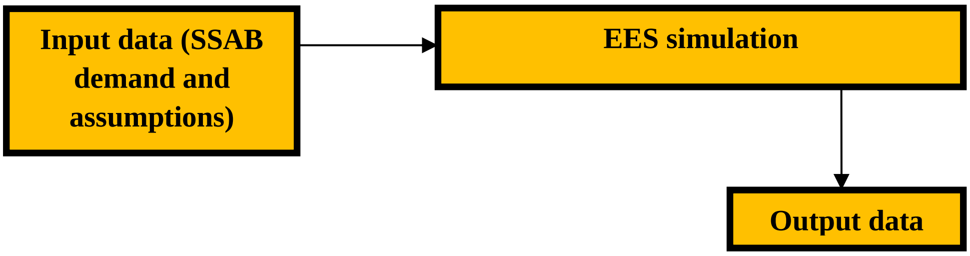

Figure 16: Modeling work flow

Referring to the mentioned types of heat pumps and refrigerants in previous sections, electric motor (CCC) closed cycle compression heat pumps were modeled using the EES software. The models showed the possibility, to deliver even 110°C, 115°C or 120 °C when using R245fa to the district heating net, having heat source temperatures between 35 °C to 45 °C. Simulation also showed that R1234ze(z) heat pumps easily could deliver 65°C to 75°C to Nyköping's district heating grid.

An experimental 700 KW high temperature heat pump using refrigerant R245fa was tested in France, 2012. The aim of that project was to evaluate the heat pump reliability and performance with R24fa. This heat pump had both a shell and tube condenser and evaporator. The temperature varied between 60 ºC and 100ºC delivered to the sink and 20 ºC to 60 ºC from the source [36],[37].

The key strategy is to increase the discharge pressure to cover the heat demand with the required capacity. A one stage heat pump could not cover a high temperature lift while having an efficient performance thus a multistage system was utilized in the heat pump modeling.

Multi-stage heat pumps are implemented in applications with many different temperature requirements. Normally they have one or two refrigerants. In the latter case there is an intermediate heat exchanger between the two refrigerants.

It would be possible to utilize multi refrigerant cascade heat pump system with different flow rates and e.g. screw compressors. It is also possible to model a two stages throttling process with a two stage turbo compressor for high capacity. As this project is about large capacity, high temperature lift and pressure ratio, a two stages heat pump in cascade with a throttling system is the most interesting configuration.

5.2 EES Modeling, of Oxelösund's heat demand:

A modelling of Oxelösund's heat demand was also done first – thus not only of Nyköping's. The demands from Oxelösund's district heating system are shown in table 5. A cascade system was here modeled for a suitable heat pump. The heat source and heat sink assumptions are defined better in appendix A.

Table 5: Oxelösund district heating energy demand [11]

Process demand	Medium	Return water temperature (T_r) C	Forward water temperature (T_f) C	Flow rate (m^3/h)	Power demand	Note
District heating (summer)	water	55	80	250	Up to 7 MW	Demand to rise the water temperature from 55 to 80 C
District heating (winter)	water	70	110-120	550	Up to 30 MW	Demand to rise the water temperature from 70 to 120 C

T_r is the temperature of the return (inlet) water from Oxelösund's district heating net

T_f is the temperature of the forward (delivered) water to Oxelösund's district heating net

The cascade system can use different refrigerants in the high and low temperatures stages. Between the two stages a shell and tube heat exchanger is assumed. This heat exchanger works as a condenser for stage one and evaporator for stage two. Of course the power of the condenser in stage one equals the evaporator power of stage two. The division temperature between the two stages, is chosen as the average between the condensation temperature of stage two and the evaporation temperature of stage one. The separation between the two stages, enables them to work separately with different refrigerants and flow rates. It is also possible to utilize different compressors in the stages.

The compressor is a very important component in a heat pump, in the case of Oxelösund only it could very well be of the screw type [38].

First, a 25 MW heat pump was modeled and simulated based on the average inlet and outlet water temperature of the cooling tower which is 45 °C in and 35 °C out. The heat sink (DH-net) would be maximum 110°C out. R245fa was first used as refrigerant in both stages of the cascade system. The model produced essentially p-h and t-s diagrams, shown later. Because of the fluctuating cooling tower water temperatures, different heat source temperatures 30 °C, 35 °C and 40 °C were simulated while varying the sink temperature between 110°C, 115°C and 120 °C. The evaporator temperatures where correspondingly assumed to 25°C, 30°C and 35°C while the condensation temperatures were assumed to 115°C, 120°C and 125°C. Later the refrigerants R600, R600a and R717 were simulated in stage one. The simulation results were compared for energy consumption and CO2 emissions. Cost of fuel, electricity, district heating price and other indexes were superficially studied. No subcooling was assumed in these models and the isentropic efficiency of the compressor was set to 70% generally. A scheme of a typical heat pump is illustrated in figure 17.

5.2.1 Modeling Flow

Figure 17 illustrates a scheme of a heat pump together with the simulation flow.

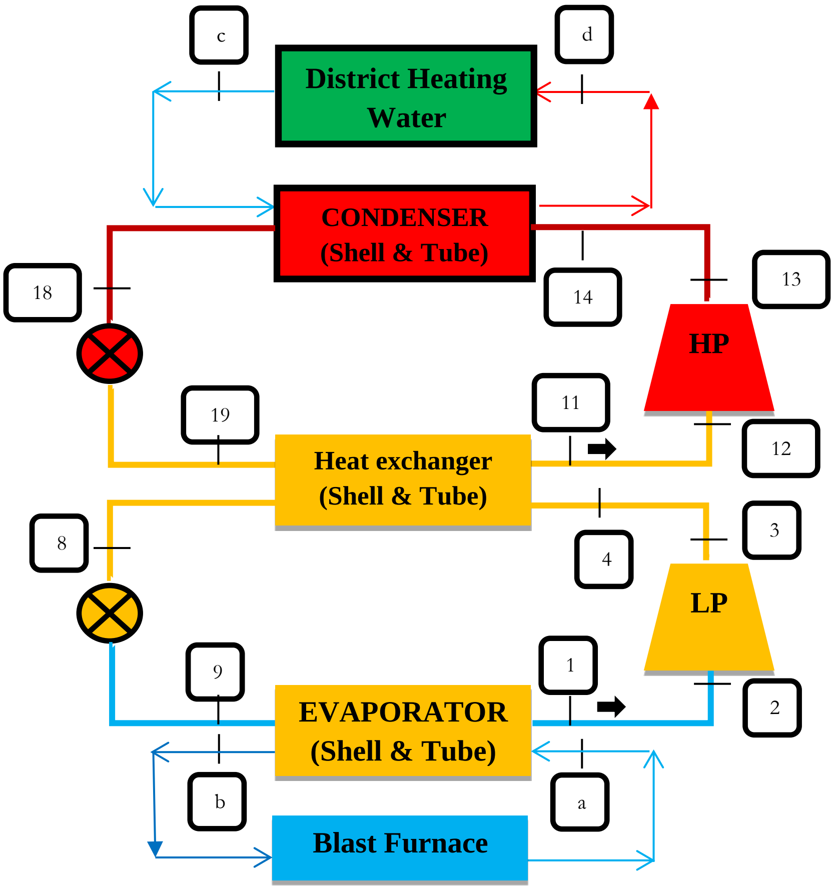

Figure 17: Schematic flow of a two stage high temperature heat pump

5.2.2 Heat Pump formulas and relations

In this part, some formulas used in each stage are described (see also appendix B). The indexes are defined in figure 17 and the p-h and t-s diagrams in figures 18 and 19.

5.2.3 P-h and T-s diagram

The figures below, illustrate the p-h and T-s diagrams of the models made in the EES software.

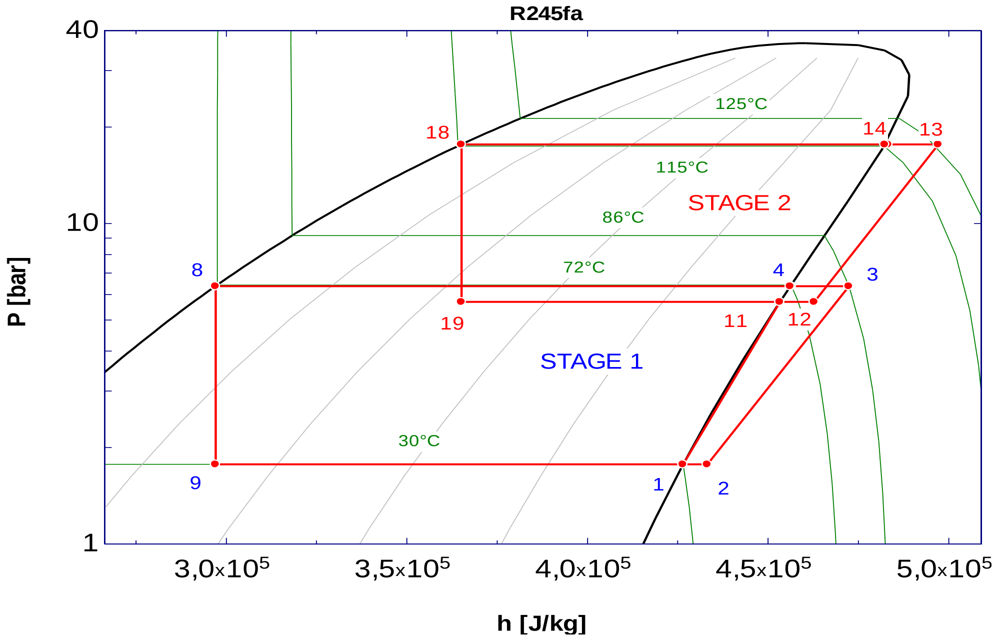

Figure 18: A two stage heat pump using a p-h diagram with R245fa in both stages, no subcooling

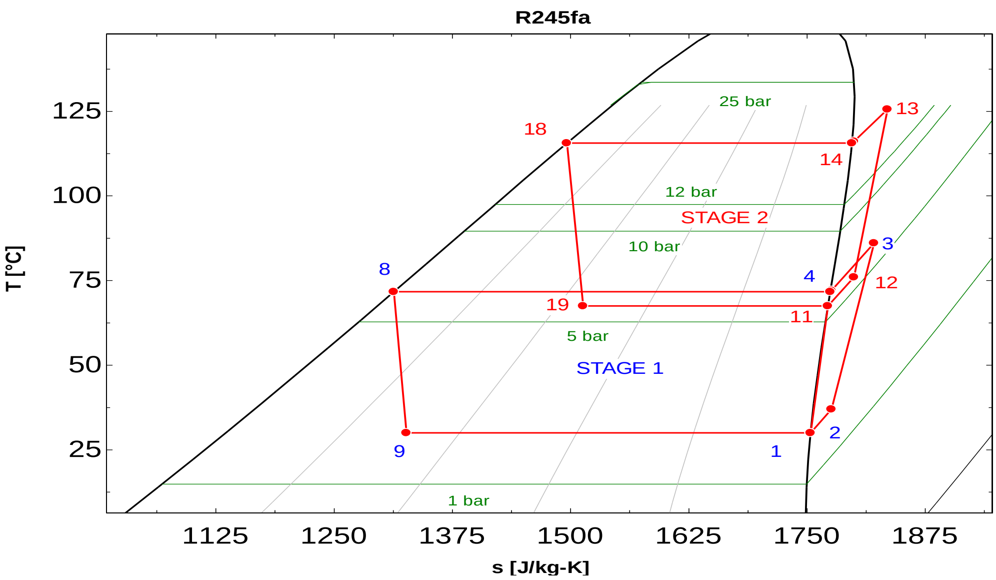

Figure 19: A two stage heat pump using a t-s diagram with R245fa in both stages, no subcooling

5.2.4 Simulation results

R600, R600a, R245fa and R717 were all tried in the first stage and R245fa was used in the second stage. The simulation result in figures 20 and 21 illustrates the heat pump COP variation. Further details are showed in tables 14 and 15.

R717-R245fa showed a higher COP but the condensation pressure rose to 42 bars which is close to the limits for ordinary compressor design. The R600-R245fa had a COP near that of R245fa-R245fa with lower refrigerant flow rate and only slightly higher condensation pressure.

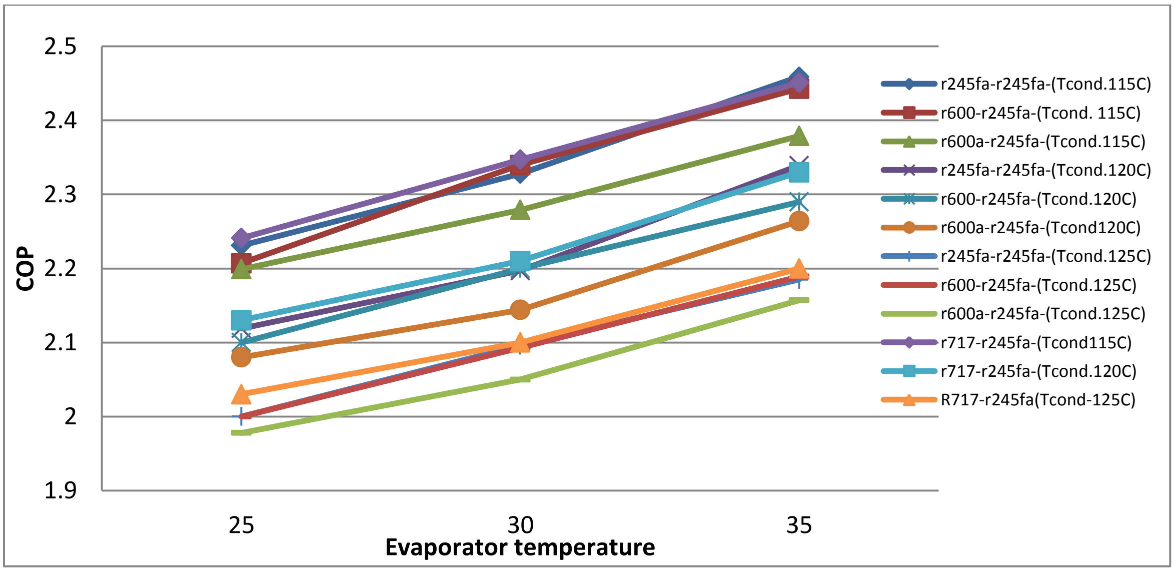

Figure 20 – The $COP_{heating}$ of the heat pump under extreme conditions in winter with evaporator temp. 25°C, 30°C, 35°C

Table 14 –simulation result in winter by condensation temperature 115°C, 120 °C , 125 °C

Winter										
Evp. Temp. (C)	Ref. stage 1	Ref. stage 2	p_{ev} stage 1 bar	$p_{ev.}$ stage 2 bar	ṁ stage 1 kg/s)	p_{cond} stage 1 bar	p_{cond} stage 2 bar	ṁ stage 2 kg/s	Condensation Tempe. (C)	condenser capacity (MW)
25	r245fa	r245fa	4.1	3.3	106	6	17.5	195	115°C	25
30	r245fa	r245fa	3.7	3.1	106	6.5	17.6	190		
35	r245fa	r245fa	3.3	2.9	110	7	17.7	191		

25	R600	r245fa	3.4	3.3	54	8.2	17.5	190		
30	R600	r245fa	3	3.1	57	8.5	17.6	196		
35	R600	r245fa	2.8	2.9	59	9.2	17.6	197		
25	R600a	r245fa	3.1	3.3	62	10.8	17.5	192		
30	R600a	r245fa	2.9	3.1	64	11.7	17.6	192		
35	R600a	r245fa	2.7	2.9	68	12.5	17.6	196		
25	R717	r245fa	3.3	3.3	15	33	17.5	191		
30	R717	r245fa	3	3.1	15	35	17.6	184		
35	R717	r245fa	2.8	2.9	17	38	17.6	200		

Evp. Temp. (C)	Ref. stage 1	Ref. stage 2	p_{ev} stage 1 bar	$p_{ev.}$ stage 2 bar	$\dot{m}$ stage 1 kg/s)	p_{cond} stage 1 bar	p_{cond} stage 2 bar	$\dot{m}$ stage 2 kg/s	(C)	(MW)
25	r245fa	r245fa	4.4	3.4	115	6.5	19.3	227	120°C	28
30	r245fa	r245fa	4	3.2	118	7	19.5	227		
35	r245fa	r245fa	3.4	2.9	123	7.1	19.5	222		
25	R600	r245fa	3.6	3.4	60	8.7	19.3	224		
30	R600	r245fa	3.2	3.2	62	9	19.5	228		
35	R600	r245fa	3	3	65	9.8	19.5	231		
25	R600a	r245fa	3.3	3.4	68	11.5	19.3	225		
30	R600a	r245fa	3.1	3.2	72	12.5	19.5	230		
35	R600a	r245fa	2.8	3	74	13	19.5	230		
25	R717	r245fa	3.5	3.4	17	35	19.3	235		
30	R717	r245fa	3.2	3.2	17	37.3	19.5	226		
35	R717	r245fa	2.9	3	18	39	19.5	230		

Evp. Temp. (C)	Ref. stage 1	Ref. stage 2	p_{ev} stage 1 bar	$p_{ev.}$ stage 2 bar	$\dot{m}$ stage 1 kg/s)	p_{cond} stage 1 bar	p_{cond} stage 2 bar	$\dot{m}$ stage 2 kg/s	(C)	(MW)
25	r245fa	r245fa	4.8	3.5	122	7	21	256	125°C	31
30	r245fa	r245fa	4.2	3.3	130	7.5	21.5	267		
35	r245fa	r245fa	3.8	3.1	135	8	21.5	268		
25	R600	r245fa	3.8	3.5	65	9.2	21.3	262		
30	R600	r245fa	3.4	3.3	67	9.6	21.5	264		
35	R600	r245fa	3.1	3.1	70	10	21.5	268		
25	R600a	r245fa	3.5	3.5	74	12.2	21.3	261		
30	R600a	r245fa	3.2	3.3	78	13	21.5	266		
35	R600a	r245fa	2.9	3.1	80	13.5	21.5	265		
25	R717	r245fa	3.7	3.5	18	37	21.3	267		
30	R717	r245fa	3.4	3.3	19	40	21.5	272		
35	R717	r245fa	3.1	3.1	20	42	21.5	275		

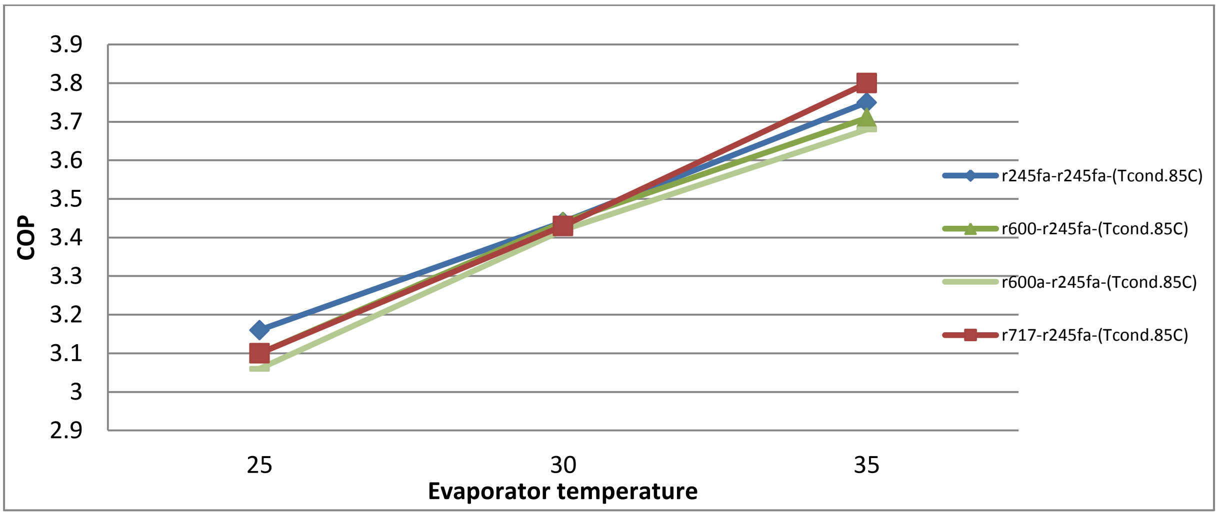

Figure 21 – COP in the summer varying evaporator the temperature 25°C, 30°C, 35°C

Table 15–simulation result in summer by condensation temperature 115°C, 120 °C, 125 °C

summer											
Evap. temp. (C)	Ref. stage 1	Ref. stage 2	COP	p_{ev} stage 1 bar	$p_{ev.}$ stage 2 bar	ṁ stage 1 kg/s	p_{cond} stage 1 bar	p_{cond} stage 2 bar	ṁ stage 2 kg/s	Condensation temp. (C) stage 2	condenser capacity (MW) stage 2
25	r245fa	r245fa	3,16	2,7	2,6	33	4	9	44	80	7,2
30	r245fa	r245fa	3,44	2,4	2,4	33	4,2	8,8	45		
35	r245fa	r245fa	3,75	2,2	2,2	35	4,6	8,7	46		
25	R600	r245fa	3,1	2,4	2,6	17	5,8	8,9	45		
30	R600	r245fa	3,44	2,1	2,4	18	5,9	8,8	46		
35	R600	r245fa	3,71	1,9	2,3	18	6,2	9,1	46		
25	R600a	r245fa	3,06	2,3	2,6	19	8	8,9	44		
30	R600a	r245fa	3,42	2	2,4	20	8	8,8	46		
35	R600a	r245fa	3,68	1,9	2,2	21	8,8	8,7	47		
25	R717	r245fa	3,1	2,3	2,6	5	23	8,9	46		
30	R717	r245fa	3,43	2,1	2,4	5	24	8,8	45		
35	R717	r245fa	3,8	1,9	2,2	6	25	8,7	51		

Figure 22 illustrates the compressor energy consumption for condenser temperature 115°C in winter. Although result of R245fa-R245fa and R600-R245fa are very close to each other, R245fa-R245fa has actually less electric consumption.

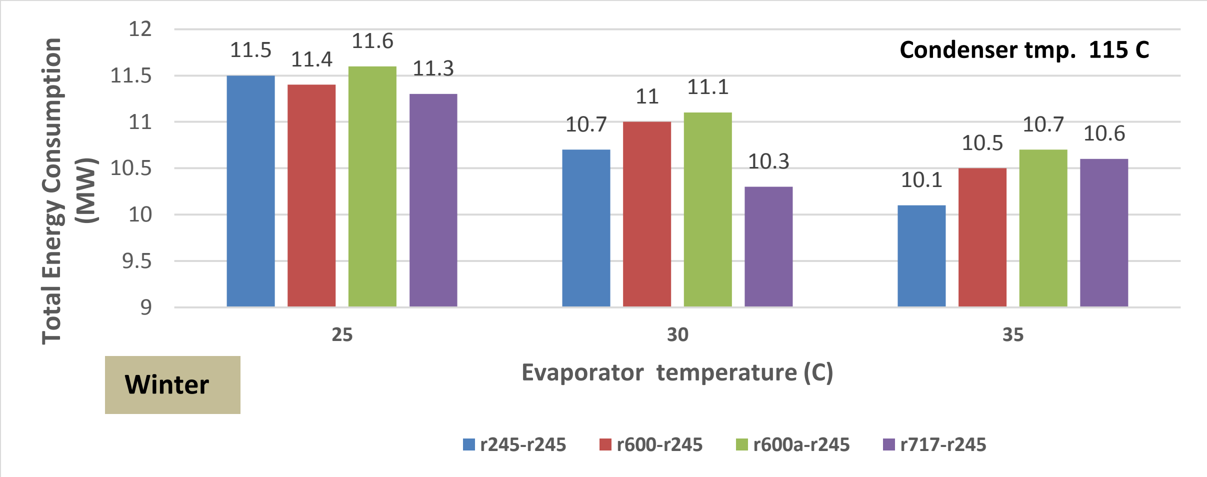

Figure 22: The compressor's electric consumption during winter (condenser temp. 115°C)

5.3 Modeling, of Oxelösund-Nyköping - also considering that waste heat could be unavailable.

It has historically been a challenge for both SSAB and Nyköping to find out how to use the waste heat for heating Nyköping's district heating network. Thus this simulation is focused on this. Another challenge is to cover the heat demand even if Oxelösund's SSAB plant is unavailable for a longer or shorter period, which means there must also be another heat source available than the cooling tower.

Table 6 shows the energy demand from Nyköping's district heat. Based on that a two stage throttling cycle heat pump was modeled in EES. The heat source and heat sink are further described in appendix A.

Table 6: Nyköping district heating energy demand [11], [20]

Process demand	Medium	Return water temperature (T_r) C	Forward water temperature (T_f) C	Flow rate (m³/h)	Power demand	Note
District heating (summer)	water	50	75	1295 Just est.	Up to 17 MW	Demand to rise water temperature from 50 to 75 C
District heating (winter)	water	50	110	1295	Up to 50 MW	Demand to rise water temperature from 50 to 110 C

T_r is the return temperature of (inlet) water from Nyköping's district heating net

T_f is the forward temperature of (deliver) water to Nyköping's district heating net

Nyköping is through 12 km from Oxelösund. Two pipes for delivering the heat pump's heated water in Oxelösund to Nyköping, is thus required, figure 23.

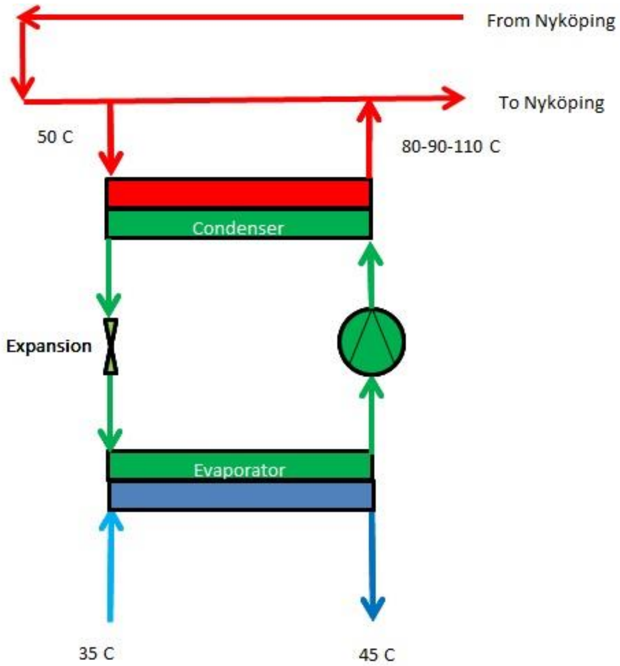

Figure 23: The pump's hot water delivery 12 km from Oxelösund to Nyköping's district heating

Maximum power demand in Nyköping is 90.4 MW out of which 15.4 MW could be considered for domestic hot water. Another way to see it is that 71 MW seems to be for outdoor temperature dependent power and 19 MW for temperature independent power. It is assumed there is 3.5 MW average heat leakage to the ground (the average leakage of a multi-family dwelling is at least 3.8%). It is also assumed that the forward temperature is 75°C summertime and 110°C wintertime (table 6). It is also assumed that indoor temperature where the buildings start to start their heating systems is 17°C [20].

Using data collected from 2000 to 2014, in Stockholm, yielding minimum and maximum diurnal outdoor temperatures of -18 °C and +25 °C. The sorted powers and temperatures expected are illustrated in figures 24 and 25 [20].

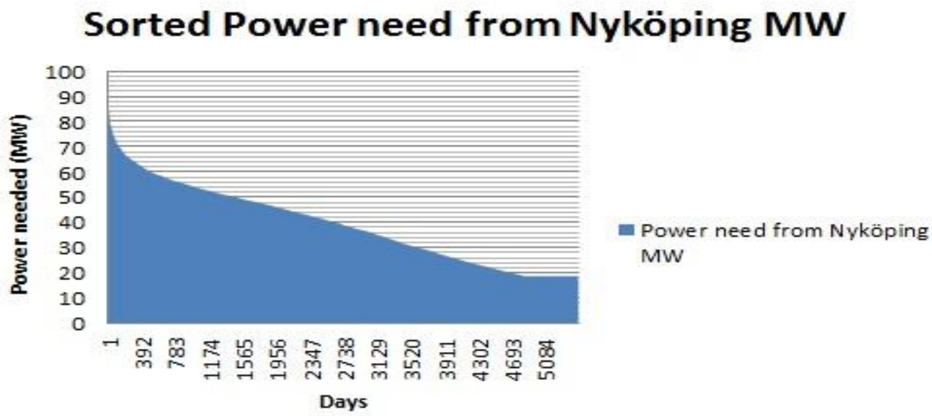

Figure 24: Sorted power need expected in Nyköping during 2000 -- 2014 years

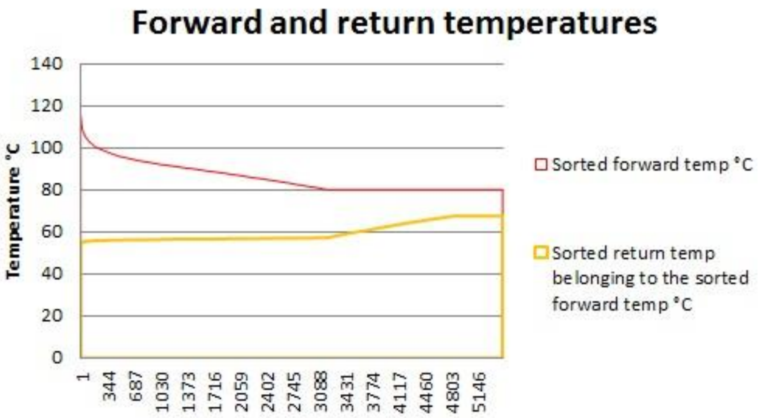

Figure 25: Sorted forward and return temperatures in Nyköping during 2000 – 2014

Regarding the cooling tower heat source, table 8, and the heat pump capacity, it would be possible to cover 50 % of Nyköping power demand with pure waste heat**,** figure 26. Taking also the electricity to the heat pump into account, the energy coverage would rise to around 62%.(Assuming a COP of around 5 and adding the electricity needed to run the heat pump, the total will result in totally 62% of the energy need for Nyköping.)

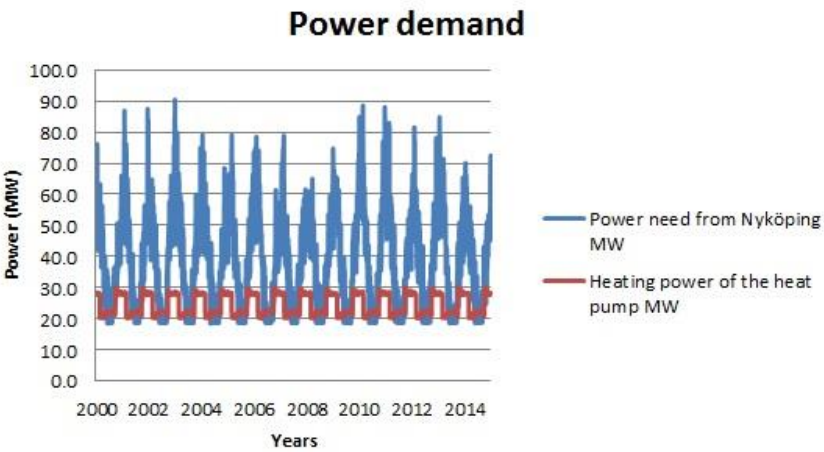

Figure 26: Comparing the power demand from Nyköping and the heat pump heat.

It is possible to model heat pump that reach 110°C the coldest day in Nyköping similar to the prior EES model but it is costly. **80** °C maximum forward temperature would be quite enough referring to figure 27. It would probably even be possible to deliver just around 70 °C by enlarging a few heat exchangers in some individual houses now craving 80 °C. The remaining temperatures between 80 °C and 110 **°**C, required in winter is expected to be taken with the existing biofuel furnace in Nyköping. About one degree would be lost in the12 km pipe. The expected condensing temperature is one degree higher than the outgoing water from the heat pump to Nyköping at any given moment.

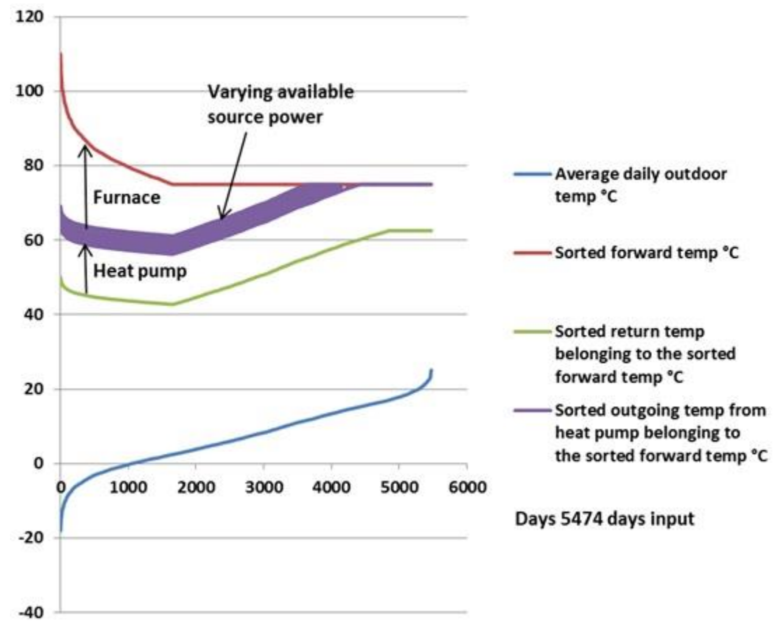

Figure 27: Covering Nyköping's heat demand via Oxelösund' heat pump

In a two stage throttling cycle, there is no heat loss between stage one and two. Thus only one refrigerant circulates through both stages. However the flow rate in each stage will be different. The heat loss between condensing temperature and the outgoing district heating water is expected to be about one degree. Compared to the cascade heat pump, with an internal heat exchanger, this model thus doesn't have any heat loss between the two stages. Instead a flash gas tank is used. It is assumed that there exist a 5 degree superheat and 5 degree subcooling in stage two in this modeling. Different refrigerants are simulated and the performances are compared. For covering a higher heat capacity, a two-stage turbo compressor is assumed. Factors related to Turbo compressors are taken into account in the simulation part, like as impeller diameter and rotational speed required.

According to table 6 and 8, the heat source and heat sink capacity are 23 MW and 28 MW during summer and winter respectively. The waste source water temperatures are then 35 °C and 45 °C. 70°C is the assumed condenser temperature. Because of the 12 km pipe and the condenser heat loss, the temperature arriving to Nyköping would be typically 68 °C. There are many more assumptions made – but nothing seems critical. It is easily possible to supply a higher temperature to Nyköping if needed.

If waste heat from the SSAB plant would cease, the heat pump would be able to work with sea water as heat source instead. Thus evaporator temperatures between 0 °C and 30 °C were simulated and the result using all refrigerants were compared. This is described more in detail formula in appendix B.´R245fa, R1234ze(z), R1234ze(e) and R1234a were simulated.

Eventually, the simulation results were compared concerning energy consumption, CO2 emissions and other indexes. The heat pump scheme is illustrated in figure 28.

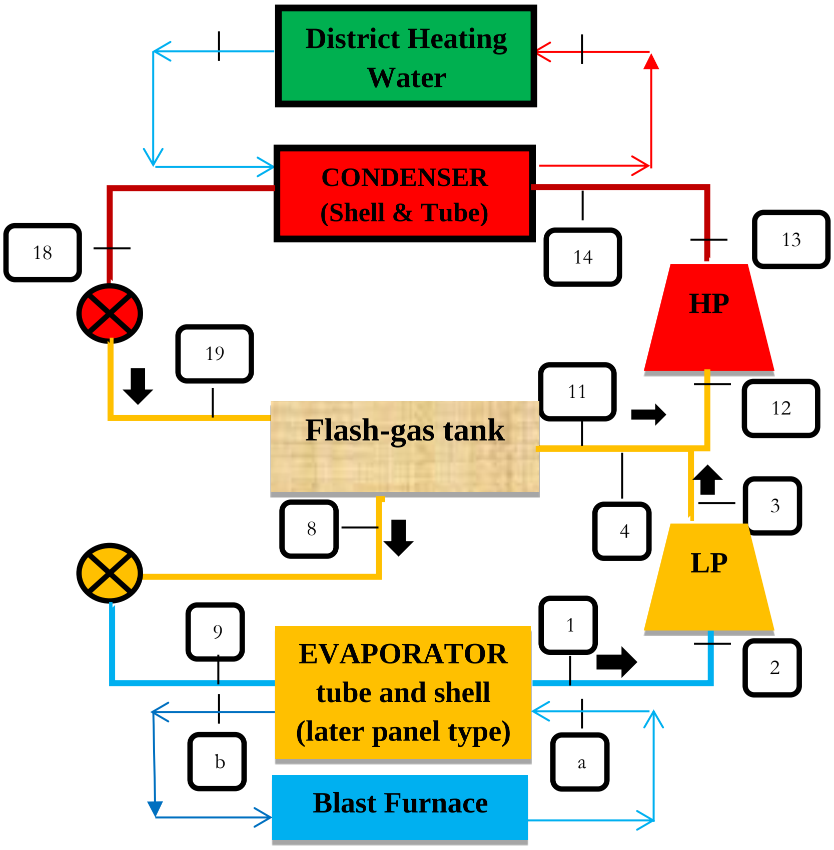

Figure 28: Schematic flow of a two stage throttling heat pump

5.3.1 Heat Pump formulas and relations

In this part, the underlying algorithms are described (see also appendix B). Indexes are defined in figure 28 and the p-h and t-s diagrams in figures 29 and 30.

5.3.2 P-h and T-s diagram

The figures below illustrates p-h and t-s diagrams modelled in EES software for an evaporator and a condenser temperature of 30 °C and 70 °C respectively using R1234ze(z) as refrigerant.

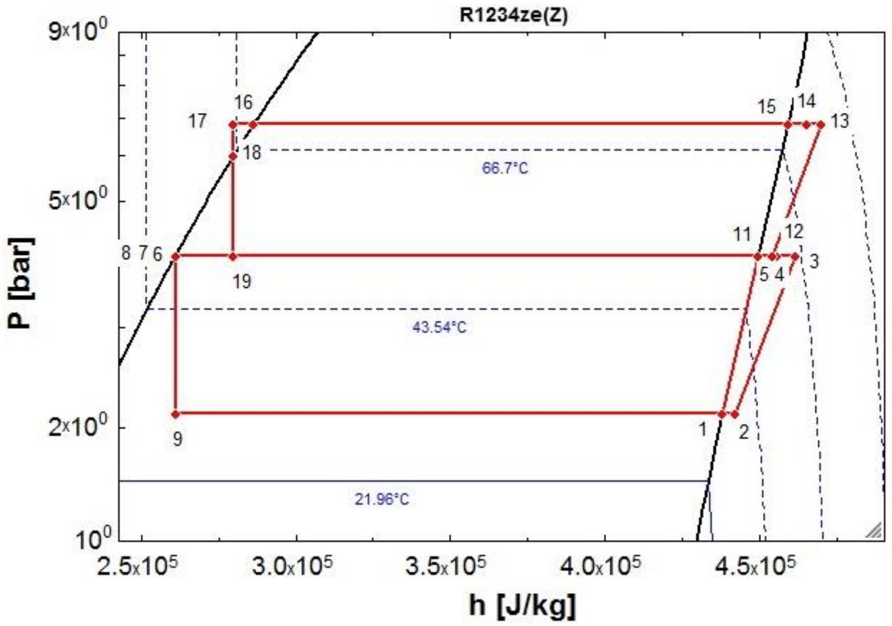

Figure 29: A two stage heat pump's p-h diagram, with R1234ze(z) in both stages

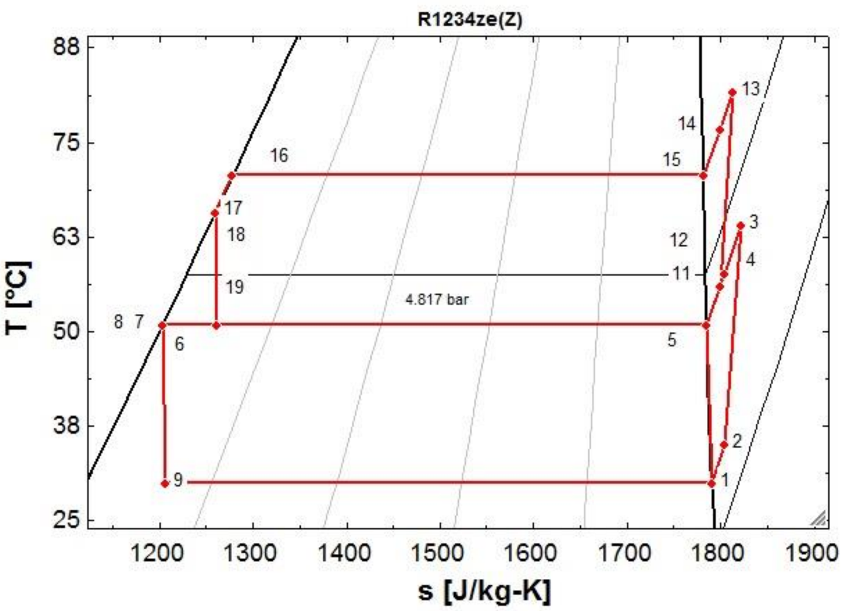

Figure 30: A two stage heat pump's t-s diagram, with R1234ze(z) in both stages

5.3.3 Simulation result

R1234ze(z), R1234ze(e), R245fa and R1234a were simulated with a turbo compressor assumed for both stages. Refrigerants like R600, R600a, R717 and partly R32 didn't have a high molecular weight enough and were not considered proper for turbo compressors with just two stages.

The condensation temperature was assumed to 70°C, but one degree loss between the condensation temperature and the district heating water, and another degree in the 12 km pipe would render a temperature to Nyköping of 68 °C. Both evaporators and condensers were considered counter flow and the compressor's rpm in each stage were in this case different. The isentropic compressor efficiency was assumed to 70%.

Simulation results are shown in figure 31 and 32. They illustrate the COP for a condensation temperature of 70°C versus three evaporator temperature (25 °C , 30n°C and 35 °C) in winter and summer. Figure 33 shows the compressor energy consumption. Formulas for the compressor are found in in appendix B. Turbo compressor details are described in table 16.

Generally, refrigerant R1234ze(z) has a higher COP both in winter and summer rather than R245fa.

Table 16: Turbo compressor specifications for each refrigerant

Refrigerant	D_{wheel} L_m	D_{wheel} H_m	RPM (min)	RPM (max)
R245fa	1.25	1.28	2129	3194
R1234ZE(e)	0.84	0.84	2997	4553
R1234ZE(z)	0.64	0.65	4366	6313

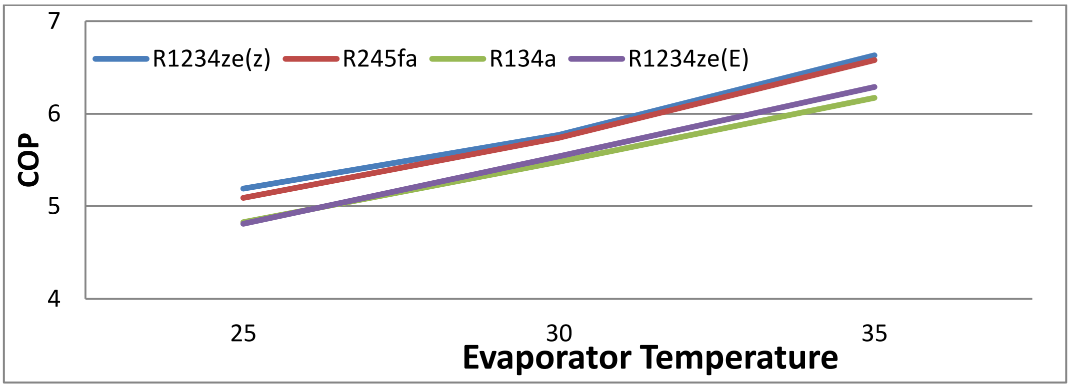

Figure 31 – COP of the heat pump in winter, evaporator temperature 25°C, 30°C, 35°C

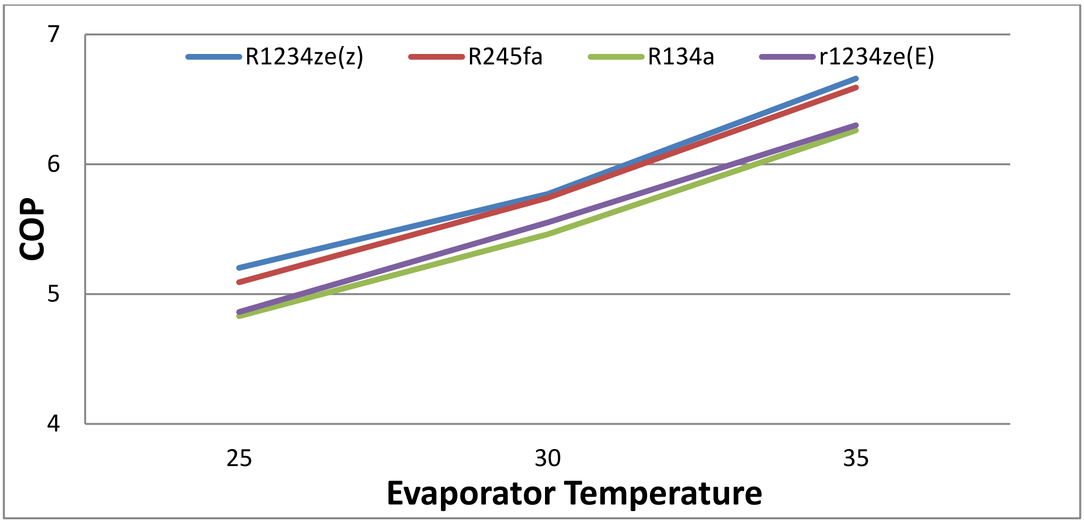

Figure 32 – COP of the heat pump in summer, evaporator temperature 25°C, 30°C, 35°C

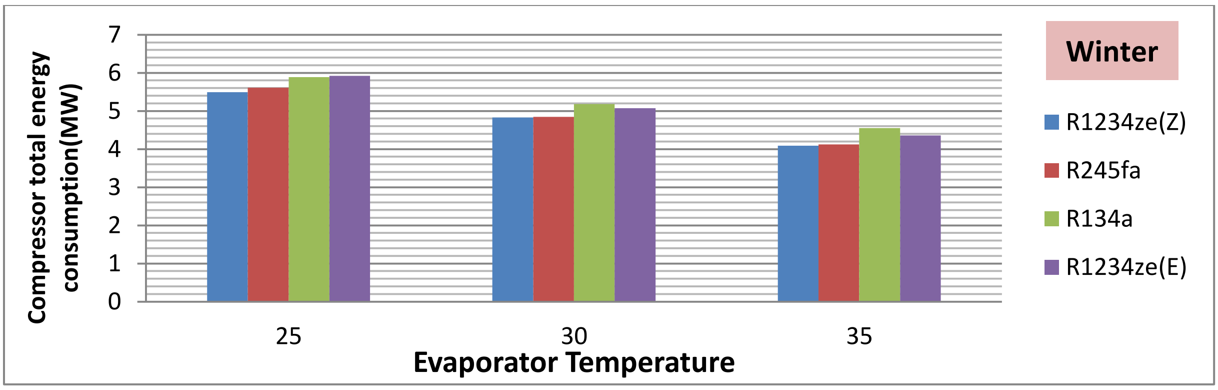

Figure 33: Compressor energy consumption winter

If the heat from the SSAB plant in Oxelösund would be unavailable, the heat pump should be able to work with sea water as heat source instead. The heat pump was therefore also simulated with evaporator temperatures from 0 °C to 30 °C and condenser temperatures from 65 °C to 75 C. Figure 34 and 35 shows the COP and heating capacity of the heat pump using R1234ze(z).

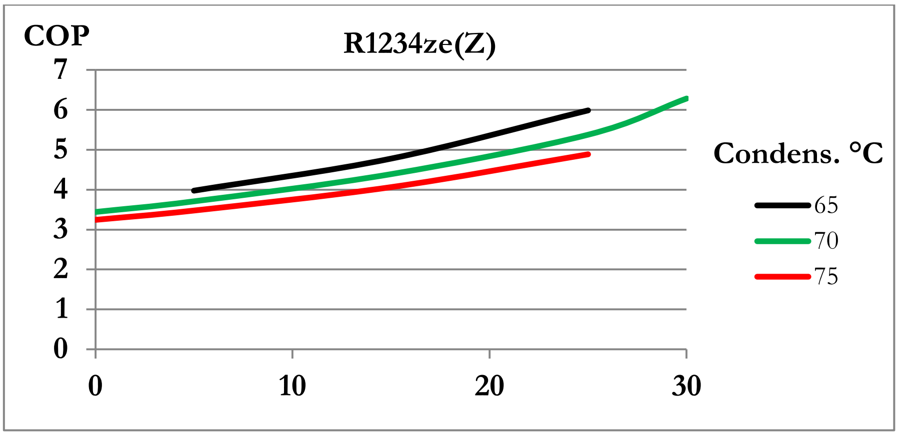

Figure 34: COP of the heat pump, evaporator temperature (0°C to 30°C)

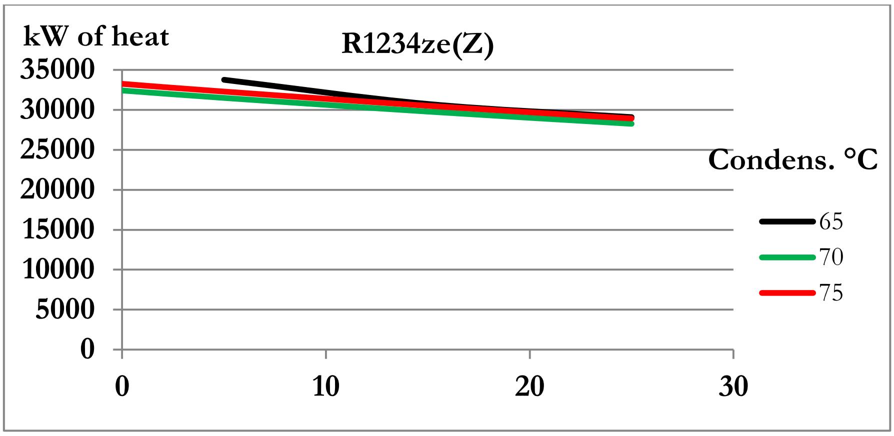

Figure 35: Heating capacity of heat pump, at evaporator temperatures 0°C to 30°C

Generally, R1234ze(z) was found to have a higher COP and heat capacity in 65 °C condensation

6 Environment and CO_2 emissions

It is very hard and almost impossible to estimate the consequences of waste heat utilization here. Some of the questions that have to be answered are:

- The waste heat is in this case created by burning coal to a large extent – how should that be accounted for? What kind of energy would be replaced by the heat pump using waste heat? Biofuel or coal?
    - If biofuel was replaced how that should be calculated?
        - With almost negligible greenhouse gas emissions (biofuel takes up CO2)?
        - Marginally as coal (the biofuel could have been exported and replaced coal?
- How is the electricity to run the heat pump generated?
    - Just using the fuels going into the Swedish national grid (almost 0 emissions!)?
    - Using CO2 data from the Nordic grid to which we are connected?
    - Using data for the total European grid for CO2 emissions.
    - Using marginal CO2 emissions from the worst plant running in the European grid?

The questions branch out into so many numerous alternatives – it is more a political than a technical question.

7 Cost

Table 17 illustrates very roughly the investment costs using waste heat and sea water as heat source. It is further discussed in Appendix C. Figure 36 illustrates a cost trend (heat vs electricity). It is also here very hard to come to a "true" value. Recently the interest rate has been low e.g. will that low interest rate remain for a long time? The price of electricity has been low during the last year – will that remain so? Some large heat pumps that were built 30 years ago – like Ropsten and Hammarby - seems to still have a viable economic life length left of another 25 years. The most vulnerable/costly parts seems to be the tubes in the heat exchangers that have to be changed.

Table 17: Capital cost of the heat pump and pipe

(Heat pump)Waste heat use		
Real interest rate	4%	(no inflation)
Time	40	years
Annuity factor	5.0%	
Specific cost for HP	10000	SEK/kW heat
Power of heat pump	30000	kW
Investment cost of HP	300	MSEK
Specific cost of pipe	10000	SEK/m
Length of pipe	12000	m
Investment cost of pipe	120	MSEK
Total cost of HP and pipe	420	MSEK
Yearly capital cost	21	MSEK

The total energy cost for the heat arriving in Nyköping can be very seen very superficially in fig 39 below.

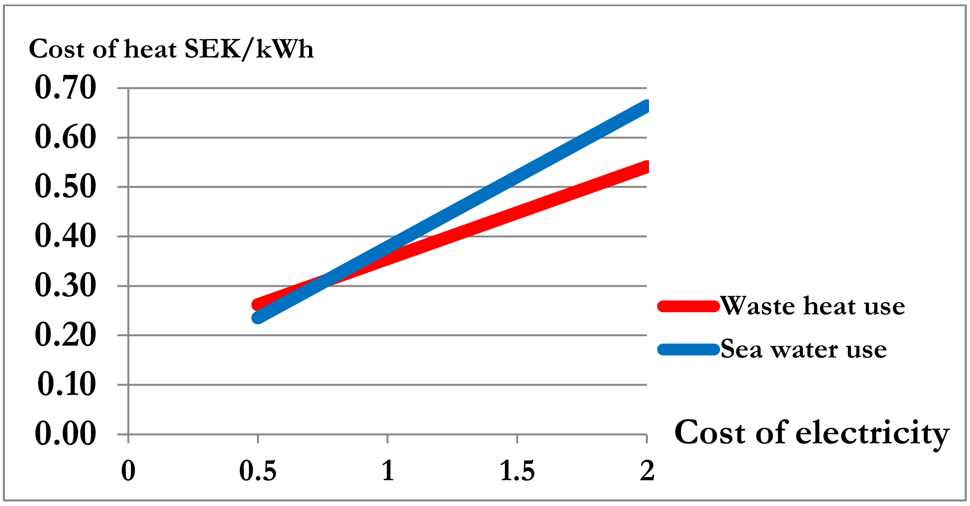

Figure 36: waste heat and sea water cost (heat vs electricity)

There are of course many other factors that must be taken into account like:

- What will the service cost?
- What will the alternatives cost in the long run?
- Should a sea water heat pump supply both Oxelösund and Nyköping?

This is just a very rough first cost estimation!

8 Results and discussion

Oxelösund's heat demand could be covered by a screw compressor cascade heat pump using either R245fa-R245fa or R600-R245fa. It could also be a good option to use R717-R245fa if a compressor can be found good for 42 bar. R717 and R600 are suitable for screw compressors but not for turbo type compressors. Large capacities, more than say 5 MW, are more economical with turbo compressors. However Oxelösund's district heat is today taken from a back pressure CHP-plant. It is unlikely that a new heat pump can compete with that unless the CHP-plant is taken out of production. However it would be possible to produce hot water for district heating with very small CO_2 emissions.

Nyköping's heat demand can be covered to about 62% with a turbo heat pump using the heat now just ventilated to the air by the cooling tower in Oxelösund. Out of these 62 %, 50 % of the energy would be waste heat and 12 % electricity to run the heat pump. R1234ze(z) is the preferred working medium as the COP is highest and the GWP is lowest. It would also be possible to cover Nyköpings heat demand using sea water as heat source, if heat from the blast furnace to the cooling tower is unavailable.

9 Conclusion

The iron and steel industry in Sweden has a 22 % share of the energy consumption in industry. The purpose of this project was to see if the uses of low temperature waste heat also could be upgraded using a high temperature industrial heat pump. As an example Nyköping's district heating demand could be covered from Oxelösund's SSAB-plant at rather low cost and with rather low CO_2 emissions.

First, a literature study was done comparing iron and steel production methods in USA and Sweden. Emphasis was waste heat recovery methods and their temperature ranges.

Secondly, a feasibility study of waste heat recovery in the SSAB iron and steel plant in Oxelösund was done. Many methods for recovering waste heat sources were considered. It was decided that recovering the waste heat from the blast furnaces cooling tower, utilizing a high temperature heat pump instead of the tower could cover 50% of heat demand in Nyköping (62% including the electricity to the heat pump). The cooling tower in SSAB is utilized 8100 hours per year and is situated close to blast furnace so there will be little need for internal piping.

The district heating forward temperature the coldest day in Nyköping is 110 °C, but it is not necessary to reach that temperature level. Delivering say just 75 °C water to the district heating system from Oxelösund and then subsequently heating up the water in Nyköping, with an existing biofuel furnace would be sufficient. Upgrading a few bad heat exchangers in the "sub centrals" could even lower the 75 °C further.

Thirdly, a literature study was done about high temperature industrial heat pumps. Applications, types of industrial heat pumps and heat pump components and were discussed. A proper type of industrial heat pump for SSAB and Nyköping's demand was recommended. Many types of refrigerants for industrial heat pumps were surveyed considering their stability, pressure, critical temperature, fitting compressor types and other indexes.

A one stage heat pump was found less suitable covering the rather high pressure ratio to lift 30°C temperature to 115°C in the case of Oxelösund or the lift 0 °C to 75 °C in the case of Nyköping using sea water. Two types of two stage heat pump systems were modeled in EES, this is better described in the appendices.

In the first type (cascade), the simulation was made using the heat source temperatures 30°C, 35°C and 40°C whereas the condenser temperatures were 115°C, 120°C and 125°C for covering mainly **Oxelösund's** heat demand. Combinations of refrigerants in the cascade were tried, to find a good COP. R245fa-R245fa, R600-R245fa and R717-R245fa had the best performances but there was a limitation in pressure. It was hard to find a compressor for ammonia able to compress to 42 bars or more. Both R717 and R600 can be used in screw compressors. But for capacities over say than 5 MW screw compressors are not economical. R600a and R600 are both flammable (A3) and ammonia is considered poisonous but not so flammable (B2)

The second type, a two stage turbo compressor using a throttling process with flash tank, was also simulated. Here temperatures are between 0 to 30 °C in the evaporator and 65 °C, 70 °C and 75 °C in the condenser to cover **Nyköping's** heat demand. It was even possible to cover higher temperatures in the condenser. Refrigerants R1234ze(z) and R245fa had the best performance, but R1234ze(z) had a lower GWP a slightly higher COP than R245fa. The best COPs were around 6 using the waste heat source. Turbo compressors technical limitations were considered whole modelling. If heat from the cooling tower/ blast furnace in Oxelösund's would be unavailable it would still be possible to use the heat pump with sea water as heat source instead of the cooling tower.

CO2 emission savings and economy is very hard to calculate and will vary depending on political view and expectations considering energy prices in the future. The investment is estimated to between 420 and 450 MSEK considering both the heat pump and the 12 km piping Oxelösund – Nyköping.

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10 Appendix

10.1 Appendix A

(Heat source and Heat sink capacity calculation)

Heat source capacity:

$Q_{water} = \dot{m}_{water} \times \varphi_{water} \times Cp_{water} \times (T_{in}-T_{out})$ Equation 1

Where:

Q_{water} is heat source capacity (MW)

$\dot{m}_{water}$ is cooling tower water flow rate ($\frac{m^3}{h}$)

φ_{water} is water density = 997.1 ($\frac{Kg}{m^3}$)

Cp_{water} is water heating capacity = 4.18 ($\frac{kJ}{kg\ K}$)

T_{in} is temperature of inlet water to cooling tower (ºC)

T_{out} is temperature of outlet water from cooling tower (ºC)

Heat sink capacity:

$Q_{water} = \dot{m}_{water} \times \rho_{water} \times Cp_{water} \times (T_{in}-T_{out})$ Equation 2

Where:

Q_{water} is heat source capacity (MW)

$\dot{m}_{water}$ is district heating water flow rate ($\frac{m^3}{h}$)

ρ_{water} is water density = 997.1 ($\frac{Kg}{m^3}$)

Cp_{water} is water heating capacity = 4.18 ($\frac{Kj}{kg\ K}$)

T_{in} is temperature of inlet water from district heating (ºC)

T_{out} is temperature of outlet water deliver to district heating (ºC)

10.2 Appendix B

(Heat Pump formula , two stage cascade)

Stage 1:

Evaporator:

Shell and tube evaporator:

$\dot{Q} = U \times A \times \upsilon_m$ Equation 3

Where:

$\dot{Q}$ is evaporator heat transfer (W)

U is overall heat transfer coefficient ($\frac{W}{m^2K}$)

υ_m is mean temperature difference between the fluids (K)

And

$\upsilon_m = \frac{Vi - Vo}{\ln\left(\frac{Vi}{Vo}\right)}$ Equation 4

Where:

υ_i is inlet heat source water and evaporation temperature difference (K)

υ_o is outlet heat source water and evaporation temperature difference (K)

Point 1 (saturated vapor pressure and temperature)

The evaporator temperature is assumed to be at least 5 degree lower than the water heat source outlet temperature on saturated vapor line. Mean temperature difference is calculated from inlet and outlet water temperature and evaporator.

Point 2:

Super heat temperature is assumed around to be 7 degrees higher than the evaporator temperature because of R245fa characteristics. With lower super heat temperature, isentropic work of the compressor would be in liquid and vapor region.

Evaporator capacity is calculated from:

$Q_{\text{evaporator}} = \dot{m}_{\text{refrigerant-1}} \times (h_2 - h_9)$ Equation 5

Where:

$Q_{\text{evaporator}}$ is evaporator capacity (W)

$\dot{m}_{\textbf{refrigerant-1}}$ is refrigerant mass flow rate in stage one ($\frac{kg}{s}$)

h_2 is refrigerant enthalpy of superheat refrigerant ($\frac{J}{kg}$)

h_9 is refrigerant enthalpy of refrigerant after expansion valve ($\frac{J}{kg}$)

Low pressure compressor:

Heat exchanger in two stage heat pump connected stage one and two to each other and worked as the condenser in stage one and evaporator in stage two. Refrigerant temperature increased with compressor up to average temperature between condensation temperature in stage one and boiling temperature in stage two (T_m) .Compressor kept the pressure constant in evaporator and condenser for evaporating and condensing .Pressure ratio is defined regarding average temperature, suction and discharge pressure line. Also isentropic efficiency is assumed to be between 0,6-0,8.

$$T_m = \left(\frac{(T_{evaporator} + T_{condenser})}{2} \right) \qquad \text{Equation 6}$$

Where:

T_m is average temperature $(°C)$

$T_{evaporator}$ is evaporator temperature in stage two $(°C)$

$T_{condenser}$ is condenser temperature in stage one $(°C)$

<u>Point 3 & 4</u>:

Pressure ratio is defined as below:

$$\text{Pressure ratio} = \left(\frac{P_3}{P_2} \right) \qquad \text{Equation 7}$$

Where:

P_3 is compressor discharge pressure (bar)

P_2 is compressor suction pressure (bar)

Isentropic work of low pressure compressor in stage one is calculated as below:

$$W_{Lp\text{-}C,is} = \frac{(\dot{m}\_refrigerant\_1) \times (h4 - h2)}{Isentropic\ efficiency} \qquad \text{Equation 8}$$

Where:

$W_{Lp\text{-}C,is}$ is isentropic work of low pressure compressor in stage one (W)

$\dot{m}_{refrigerant\text{-}1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_2 is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_4 is isentropic enthalpy of discharge refrigerant from isentropic compression process $(\frac{J}{kg})$

Isentropic efficiency is between 0,6-0,8.

Other losses such as mechanical, fluid friction, heat losses are assumed to be compensated in real work of compressor which is assumed to be 15%

$W_{Lp\text{-}C,Real} = W_{Lp\text{-}C,is} \times$ other losses Equation 9

Where:

$W_{Lp\text{-}C,is}$ is isentropic work of low pressure compressor in stage one (W)

$W_{Lp\text{-}C,Real}$ is total work of low pressure compressor in stage one (W)

And

$W_{Lp\text{-}C,Real} = (\dot{m}_{refrigerant\text{-}1}) \times (h3 - h2)$ Equation 10

Where:

$W_{Lp\text{-}C,Real}$ is total work of low pressure compressor in stage one (W)

$\dot{m}_{refrigerant\text{-}1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_2 is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_3 is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

Pressure would be constant in discharge line and condenser:

$P_3 = P_4 = P_8 \quad (bar)$ Equation 11

Volumetric flow rate at the compressor inlet:

$\dot{V} = \dot{m}_{refrigerant\text{-}1} \times v_{2k}$ Equation 12

Where:

$\dot{V}$ is volumetric flow rate at compressor inlet $(\frac{m^3}{s})$

$\dot{m}_{refrigerant\text{-}1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

v_{2k} is specific volume of the refrigerant at compressor inlet $(\frac{m^3}{kg})$

Heat exchanger between stage 1 and 2:

Shell and tube heat exchanger between stage one and two transfer heats indirectly. Also heat exchanger works as condenser in stage one and evaporator in stage two. This make possible to use different refrigerant in each stage.

Heat exchanger temperature with constant pressure is explained in equation 6, so:

$Q_{Hx} = \dot{m}_{refrigerant\text{-}1} \times (h_3 - h_8)$ Equation 13

Where:

Q_{Hx} is heat exchanger capacity (W)

$\dot{m}_{refrigerant-1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_8 is refrigerant enthalpy in expansion valve inlet $(\frac{J}{kg})$

h_3 is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

<u>Point 8:</u>

Heat exchanger pressure and temperature is on saturate liquid line. Refrigerant discharge pressure of compressor and heat exchanger is constant.

$P_3 = P_4 = P_8$ (bar) Equation 14

Expansion valve:

<u>Point 8:</u>

Enthalpy is the same in point 8 and 9.

Temperature and pressure after expansion vale is the same with evaporator pressure.

$h_8 = h_9$ (J) Equation 15

$T_8 > T_9$ $(^\circ C)$, $P_8 > P_9$ (bar) Equation 16

$T_9 = T_1$ $(^\circ C)$ Equation 17

$P_9 = P_1$ (bar) Equation 18

And regarding equation 13:

$\dot{m}_{refrigerant-1} = Q_{Hx} \times (h_3 - h_8)$ Equation 19

Where:

Q_{Hx} is heat exchanger capacity (W)

$\dot{m}_{refrigerant-1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_8 is refrigerant enthalpy in expansion valve inlet $(\frac{J}{kg})$

h_3 is enthalpy of discharge refrigerant from real compression process (J)

Different flow rate is tested in simulation to find out better result to cover condenser capacity in stage 2 and achieve higher COP.

Stage 2:

Stage2 is similar to stage one

Refrigerant in stage2:

Refrigerant mass flow rate is calculated from heat exchanger capacity.

$\dot{m}_{\text{refrigerant-2}} = \left(\frac{Q_{Hx}}{h_{12} - h_{19}} \right)$ Equation 20

Where:

Q_{Hx} is heat exchanger capacity (W)

$\dot{m}_{\text{refrigerant-2}}$ is refrigerant flow rate in stage two $\left(\frac{kg}{s}\right)$

h_{12} is refrigerant enthalpy of superheat refrigerant $\left(\frac{j}{kg}\right)$

h_{19} is refrigerant enthalpy after expansion valve $\left(\frac{j}{kg}\right)$

Heat exchanger:

<u>Point 11:</u>

Heat exchanger works as evaporator .Temperature is 5 degree less than heat exchanger temperature in stage 1 because of heat losses.

<u>Point 12:</u>

Super heat temperature of refrigerant is assumed to be around 7 degree higher than evaporator temperature because of R245fa characteristics in point 12.With lower super heat temperature, isentropic work of the compressor would be in saturate vapor and liquid region.

High pressure compressor:

High pressure compressor in second stage kept the pressure constant in evaporator (heat exchanger) and condenser. Pressure ratio is defined regarding condensation temperature, suction and discharge pressure line. Also isentropic efficiency is assumed to be between 0,6-0,8.

<u>Point 13 & 14</u>:

Pressure ratio in high pressure compressor:

Pressure ratio = $\left(\frac{P_{13}}{P_{12}} \right)$ Equation 21

Where:

P_{13} is compressor discharge pressure (bar)

P_{12} is compressor suction pressure (bar)

Isentropic work of high pressure compressor in stage two is calculated as below:

$$W_{Hp\text{-}C,is} = \frac{(\dot{m}\_refrigerant\_2) \times (h14 - h12)}{Isentropic\ efficiency} \quad \text{Equation 22}$$

Where:

$W_{Hp\text{-}C,is}$ is isentropic work of high pressure compressor in stage two (W)

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage two $(\frac{kg}{s})$

h_{12} is enthalpy of superheat refrigerant from heat exchanger in stage two $(\frac{J}{kg})$

h_{14} is isentropic enthalpy of discharge refrigerant from isentropic compression process $(\frac{J}{kg})$

Isentropic efficiency is between 0,6-0,8.

Other losses such as mechanical, fluid friction, heat losses are assumed to be compensated in real work of the compressor and assumed to be 15%

$$W_{Hp\text{-}C,Real} = W_{Hp\text{-}C,is} \times \text{other losses} \quad \text{Equation 23}$$

And

$$W_{Hp\text{-}C,Real} = (\dot{m}_{refrigerant\text{-}2}) \times (h_{13} - h_{12}) \quad \text{Equation 24}$$

Where:

$W_{Hp\text{-}C,Real}$ is total work of low pressure compressor in stage two (W)

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage two $(\frac{kg}{s})$

h_{12} is enthalpy of superheat refrigerant (J)

h_{13} is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

Pressure would be constant in discharge line and condenser:

$$P_{13} = P_{14} = P_{18} \quad (bar) \quad \text{Equation 25}$$

Volumetric flow rate at the compressor inlet:

$$\dot{V} = \dot{m}_{refrigerant\text{-}2} \times \nu_{2k} \quad \text{Equation 26}$$

Where:

$\dot{V}$ is volumetric flow rate at compressor inlet $(\frac{m^3}{s})$

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage two ($\frac{kg}{s}$)

ν_{2k} is specific volume of the refrigerant at compressor inlet ($\frac{m^3}{kg}$)

Condenser:

Shell and tube condenser in stage 2 transfer heat to water inlet from district heating. Condenser temperature is assumed to be at least 5 degree higher than water heat sink outlet temperature.

$$\dot{Q} = U \times A \times V_m \qquad \text{Equation 27}$$

Where:

$\dot{Q}$ is condenser heat transfer (W)

U is overall heat transfer coefficient ($\frac{W}{M^2K}$)

V_m is mean temperature difference between the fluids (K)

And

$$\upsilon_m = \frac{Vi-Vo}{\ln(\frac{Vi}{Vo})} \qquad \text{Equation 28}$$

Where:

υ_i is inlet heat sink water and condensation temperature difference (K)

υ_o is outlet heat sink water and condensation temperature difference (K)

Condenser capacity:

$$Q_{condenser} = \dot{m}_{refrigerant\text{-}2} \times (h_{13} - h_{18}) \qquad \text{Equation 29}$$

Where:

$Q_{condenser}$ is condenser capacity (W)

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage two ($\frac{kg}{s}$)

h_{18} is refrigerant enthalpy in expansion valve inlet ($\frac{J}{kg}$)

h_{13} is enthalpy of discharge refrigerant from real compression process ($\frac{J}{kg}$)

Point 18:

Condenser pressure and temperature is on saturate liquid line .Refrigerant discharge pressure of compressor and heat condenser is constant.

$P_{13} = P_{14} = P_{18}$ (bar) Equation 30

Expansion valve:

Enthalpy is the same in point 18 and 19.

Temperature and pressure after expansion vale is the same with evaporator pressure in stage two.

$h_{18} = h_{19}$ $\left(\frac{J}{kg}\right)$ Equation 31

$T_{18} > T_{19}$ $(°C)$, $P_{18} > P_{19}$ (bar) Equation 32

$T_{19} = T_{11}$ $(°C)$ Equation 33

$P_{19} = P_{11}$ (bar) Equation 34

Coefficient of performance:

Model calculated coefficient of performance base on two stage heat pump with heat exchanger

$$COP = \frac{\text{Q condenser}}{\text{WHp−C,Real} + \text{WLp−C,Real}}$$ Equation 35

Where:

$Q_{condenser}$ is condenser capacity (W)

$W_{Hp\text{-}C,Real}$ is high pressure compressor real work (W)

$W_{Lp\text{-}C,Real}$ is low pressure compressor real work (W)

(**Heat Pump formula, two stage throttling cycle**)

Stage 1:

Evaporator:

Shell and tube evaporator:

$\dot{Q} = U \times A \times \upsilon_m$ Equation 36

Where:

$\dot{Q}$ is the evaporator heat transfer (W)

U is the overall heat transfer coefficient ($\frac{W}{m^2K}$)

υ_m means the temperature difference between the fluids (K)

And

$\upsilon_m = \frac{Vi - Vo}{\ln(\frac{Vi}{Vo})}$ Equation 37

Where:

υ_i is the inlet heat source water and evaporation temperature difference (K)

υ_o is the outlet heat source water and evaporation temperature difference (K)

<u>Point 1</u> (saturate vapor pressure and temperature)

Evaporator temperature is assumed to be at least 5 degree lower than water heat source outlet temperature on saturated vapor line. Mean temperature difference is calculated from inlet and outlet water temperature and evaporator.

<u>Point 2:</u>

Super heat temperature is assumed around 5 degree higher than evaporator temperature. With lower super heat temperature, isentropic work of the compressor would be in liquid and vapor region.

Evaporator capacity is calculated from:

$Q_{evaporator} = \dot{m}_{refrigerant\text{-}1} \times (h_2 - h_9)$ Equation 38

Where:

$Q_{evaporator}$ is evaporator capacity (W)

$\dot{m}_{\mathbf{refrigerant\text{-}1}}$ is refrigerant mass flow rate in stage one ($\frac{kg}{s}$)

h_2 is refrigerant enthalpy of superheat refrigerant ($\frac{J}{kg}$)

h_9 is refrigerant enthalpy of refrigerant after expansion valve ($\frac{J}{kg}$)

Low pressure compressor:

Stage one and two are connected with one gas chamber, then there is no heat losses between stage one and two. Refrigerant temperature increased with Turbo compressor up to average temperature between condensation temperature in stage one and boiling temperature in stage two (T_m) .Compressor kept the pressure constant in evaporator and condenser for evaporating and condensing .Pressure ratio is defined regarding average temperature, suction and discharge pressure line. Also isentropic efficiency is assumed to be between 0,6-0,8 .

$$T_m = \left(\frac{(T\ evaporator + T\ condenser)}{2}\right) \qquad \text{Equation 39}$$

Where:

T_m is average temperature $(°C)$

$T_{evaporator}$ is evaporator temperature in stage two $(°C)$

$T_{condenser}$ is condenser temperature in stage one $(°C)$

$$W_{Lp\text{-}C,is} = \Psi \times \left(\frac{U2^{\wedge}2}{2}\right) \qquad \text{Equation 40}$$

Where:

Ψ is impeller head factor between 1 to 1.3

U2 is impeller tip speed (m/sec)

$W_{Lp\text{-}C,is}$ is isentropic work of low pressure turbo compressor in stage one (W)

$$U2 = \left(\frac{3.14 \times Dwheel \times RPM}{60}\right) \qquad \text{Equation 41}$$

Where:

D_{wheel} is impeller diameter (m)

U2 is impeller tip speed (m/sec)

<u>**Point 3 & 4**</u>**:**

Pressure ratio is defined as below:

$$\text{Pressure ratio} = \left(\frac{P_3}{P_2}\right) \qquad \text{Equation 42}$$

Where:

P_3 is compressor discharge pressure (bar)

P_2 is compressor suction pressure (bar)

Isentropic work of low pressure compressor in stage one is calculated as below:

$$W_{Lp\text{-}C,is} = \frac{(\dot{m}\_refrigerant\_1) \times (h4 - h2)}{Isentropic\ efficiency}$$ Equation 43

Where:

$W_{Lp\text{-}C,is}$ is isentropic work of low pressure compressor in stage one (W)

$\dot{m}_{refrigerant\text{-}1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_2 is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_4 is isentropic enthalpy of discharge refrigerant from isentropic compression process $(\frac{J}{kg})$

Isentropic efficiency is between 0,6-0,8.

And

$$Isentropic\ efficiency = \frac{(h4 - h2)}{(h3 - h2)}$$ Equation 44

Where:

h_3 is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

h_2 is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_4 is isentropic enthalpy of discharge refrigerant from isentropic compression process $(\frac{J}{kg})$

Isentropic efficiency is between 0,6-0,8.

$$W_{Lp\text{-}C,Real} = (\dot{m}_{refrigerant\text{-}1}) \times (h3 - h2)$$ Equation 45

Where:

$W_{Lp\text{-}C,Real}$ is total work of low pressure compressor in stage one (W)

$\dot{m}_{refrigerant\text{-}1}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_2 is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_3 is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

Pressure would be constant in discharge line and condenser:

$$P_3 = P_4 = P_8 \quad (bar)$$ Equation 46

Volumetric flow rate through the wheel :

$$V_2 = \varphi \times U2 \times 3.14 \times \frac{Dwheel^2}{4}$$ Equation 47

Where:

φ is volume flow rate between 0.04-0.06

U2 is impeller tip speed (m/sec)

V_2 is incoming volume flow rate $\left(\frac{m^3}{sec}\right)$

Point 8:

Refrigerant discharge pressure of compressor and flash gas chamber is constant.

$P_3= P_4= P_8$ (bar) Equation 48

Expansion valve:

Point 8:

Enthalpy is the same in point 8 and 9.

Temperature and pressure after expansion vale is the same with evaporator pressure.

$h_8=h_9$ $\left(\frac{j}{kg}\right)$ Equation 49

$T_8 > T_9$ $({}^{\circ}\ C)$, $P_8 > P_9$ (bar) Equation 50

$T_9=T_1$ $({}^{\circ}\ C)$ Equation 51

$P_9 = P_1$ (bar) Equation 52

And

$\dot{m}_{refrigerant-1} = Q_{cond1} \times (h_3-h_8)$ Equation 53

Where:

Q_{cond1} is condenser of stage 1 (W)

$\dot{m}_{refrigerant-1}$ is refrigerant flow rate in stage one $\left(\frac{kg}{s}\right)$

h_8 is refrigerant enthalpy in expansion valve inlet $\left(\frac{J}{kg}\right)$

h_3 is enthalpy of discharge refrigerant from real compression process $\left(\frac{J}{kg}\right)$

Different flow rate is tested in simulation to find out better result to cover condenser capacity.

Stage 2:

Stage2 is similar to stage one

Refrigerant in stage2:

Refrigerant mass flow rate is calculated from condenser capacity.

$\dot{m}_{\text{refrigerant-2}} = \left(\frac{Q_{cond2}}{h_{12} - h_{19}} \right)$ Equation 54

Where:

Q_{cond2} is condenser capacity in stage 2 (W)

$\dot{m}_{\text{refrigerant-2}}$ is refrigerant flow rate in stage two $\left(\frac{kg}{s}\right)$

h_{12} is refrigerant enthalpy of superheat refrigerant $\left(\frac{J}{kg}\right)$

h_{19} is refrigerant enthalpy after expansion valve $\left(\frac{J}{kg}\right)$

Point 11:

In flash gas chamber there is no heat loss between stage one and two. Then, point 11 has temperature the same as T_m .

Point 12:

Super heat temperature of refrigerant is assumed to be around 5 degree higher than T_m.

High pressure compressor:

High pressure Turbo compressor in second stage kept the pressure constant in evaporator (heat exchanger) and condenser. Pressure ratio is defined regarding condensation temperature, suction and discharge pressure line. Also isentropic efficiency is assumed to be between 0,6-0,8.

$W_{Hp\text{-}C,is} = \Psi \times \left(\frac{U2^2}{2} \right)$ Equation 55

Where:

Ψ is impeller head factor between 1 to 1.3

$U2$ is impeller tip speed (m/sec)

$W_{Hp\text{-}C,is}$ is isentropic work of low pressure turbo compressor in stage one (W)

$$U2=\left(\frac{3.14 \times Dwheel \times RPM}{60}\right)$$ Equation 56

Where:

D_{wheel} is impeller diameter (m)

U2 is impeller tip speed (m/sec)

Point 13 & 14:

Pressure ratio in high pressure compressor:

$$\text{Pressure ratio} = \left(\frac{P_{13}}{P_{12}}\right)$$ Equation 57

Where:

P_{13} is compressor discharge pressure (bar)

P_{12} is compressor suction pressure (bar)

And

$$\textit{Isentropic efficiency} = \frac{(h_{14}-h_{12})}{(h_{13}-h_{12})}$$ Equation 58

Where:

h_{13} is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

h_{12} is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_{14} is isentropic enthalpy of discharge refrigerant from isentropic compression process $(\frac{J}{kg})$

Isentropic efficiency is between 0,6-0,8.

$$W_{Hp\text{-}C,Real} = (\dot{m}_{refrigerant\text{-}2}) \times (h_{13} - h_{12})$$ Equation 59

Where:

$W_{Hp\text{-}C,Real}$ is total work of low pressure compressor in stage one (W)

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage one $(\frac{kg}{s})$

h_{12} is enthalpy of superheat refrigerant $(\frac{J}{kg})$

h_{13} is enthalpy of discharge refrigerant from real compression process $(\frac{J}{kg})$

Pressure would be constant in discharge line and condenser:

$P_{13} = P_{14} = P_{18} \quad (bar)$ Equation 60

Volumetric flow rate through the wheel :

$V_2 = \varphi \times U2 \times 3.14 \times \frac{D_{wheel}^2}{4}$ Equation 61

Where:

φ is volume flow rate between 0.04-0.06

U2 is impeller tip speed (m/sec)

V_2 is incoming volume flow rate $\left(\frac{m^3}{sec}\right)$

Condenser:

Shell and tube condenser in stage 2 transfer heat to water inlet from district heating. Condenser temperature is assumed to be at least 1 degree higher than water heat sink outlet temperature, because of one degree heat loss.

$\dot{Q} = U \times A \times V_m$ Equation 62

Where:

$\dot{Q}$ is condenser heat transfer (W)

U is overall heat transfer coefficient $\left(\frac{W}{m^2K}\right)$

V_m is mean temperature difference between the fluids (K)

And

$\upsilon_m = \frac{Vi - Vo}{\ln(\frac{Vi}{Vo})}$ Equation 63

Where:

υ_i is inlet heat sink water and condensation temperature difference (K)

υ_o is outlet heat sink water and condensation temperature difference (K)

Condenser capacity:

$Q_{condenser} = \dot{m}_{refrigerant\text{-}2} \times (h_{13} - h_{18})$ Equation 64

Where:

$Q_{condenser}$ is condenser capacity (W)

$\dot{m}_{refrigerant\text{-}2}$ is refrigerant flow rate in stage two $(\frac{kg}{s})$

h_{18} is refrigerant enthalpy in expansion valve inlet $(\frac{J}{kg})$

h_{13} is enthalpy of discharge refrigerant from real compression process (J)

Point 18:

There is 5 degree subcooling in condenser and point 18 has lower pressure than point 17.

$T_{18}=T_{16}\text{-}5°C$ Equation 65

$P_{13}= P_{14}= P_{15} =P_{16}= P_{17}$ (bar) Equation 66

Expansion valve:

$h_{17}=h_{18}$ Equation 67

$h_{19}=h_{18}$ Equation 68

Enthalpy is the same in point 18 and 19.

Temperature and pressure after expansion vale is the same with evaporator pressure in stage two.

$h_{18}=h_{19}$ $(\frac{J}{kg})$ Equation 69

$T_{18} > T_{19}$ $(^{\circ}C)$, $P_{18} > P_{19}$ (bar) Equation 70

$T_{19}=T_{11}$ $(^{\circ}C)$ Equation 71

$P_{19} = P_{11}$ (bar) Equation 72

Coefficient of performance:

Model calculated coefficient of performance base on two stage throttling heat pump

$$COP_{total} = \frac{Q_{condenser\ 2}}{W_{Hp\_C,Real} + W_{Lp\_C,Real}}$$ Equation 73

Where:

$Q_{condenser\ 2}$ is condenser capacity in stage two (W)

$W_{Hp\text{-}C,Real}$ is high pressure compressor real work (W)

$W_{Lp\text{-}C,Real}$ is low pressure compressor real work (W)

10.3 Appendix C

(COST)

Annuity factor $a = \dfrac{r}{(1-e^{-rT})}$ $r = rate\ and\ T = time$ Equation 74

Investment cost for heat pump (MSEK) = $\dfrac{\textbf{heat pump specific cost} \times \textbf{heat pump power}}{\textbf{1000000}}$ Equation 75

Investment cost for pipe (MSEK) = $\dfrac{\textbf{pipe specific cost} \times \textbf{pipe lenght}}{\textbf{1000000}}$ Equation 76

Total cost of pipe and heat pump = Cost of heat pump + Cost of pipe Equation 77

Yearly capital cost = Annuity factor × Total cost of pipe and heat pump Equation 78

Total cost of electricity $(\frac{MSEK}{year})$ = $\dfrac{\textbf{consumed el.} \times \textbf{el.specific cost}}{\textbf{1000000}}$ Equation 79

Total cost per year $(\frac{MSEK}{year})$ = Total cost of electricity + Yearly capital cost Equation 80

Specific cost of heat $(\frac{SEK}{KWh})$= $\dfrac{\textbf{Total cost per year}}{\textbf{Delivered heat}}$ Equation 81