



THE INSTITUTE OF REFRIGERATION

Ammonia Heat Pumps for District Heating in Norway – a case study

by

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Introduction

On our path to minimise humans impact on the planet's environment in a time where there is a growing number of people in the World there is a lot of focus on what can be done in the energy arena. One of the biggest issues is the reduction of emission of CO₂ and other GWP gases to minimise manmade changes to the global climate.

By using local resources like wind, water and bio fuels or optimise the usage of fossil fuel it is possible to significantly lower the global warming impact per kW heat or electricity produced.

After the Second World War, the neutral countries Sweden and Switzerland experienced a significant shortage in imported heating fuel and started investigating the use of local resources. Both countries have a large resource of hydro electric power. By encouraging installation of heat pumps they became less dependent on imported fossil fuel. In the years that followed many large districts heating system were installed in major cities using either waste heat from power stations or large scale heat pumps.

During the two oil crises in early and late 1970's the focus on local energy sources increased once more. The Nordic countries, with significant hydro and nuclear power production renewed their interest in large heat pumps (5 – 100 MW) and several plant were installed across Norway, Sweden

and Finland. All these plants used R22 as refrigerant. Several systems combined district cooling and heating.

Following the Montreal protocol signing in 1989 and the increased focus on ozone depletion and high GWP gases the process of phasing out R22 was started. Existing heat pumps are being changed to run on R134a and in recent years all new high temperature heatpumps (>80°C) for district heating were based on R134a. Although the performance of R134a is not as good as R22 it does not require higher design pressure and a lot of the technology from the R22 systems can be re-used in the R134a heat pumps.

The lower efficiency of R134a has made some large heat pump installation uneconomical to run and the district heating companies have been looking for higher efficiency and future proof alternative. Refrigeration is often used in the food industry as one of many items to add value to a product and optimisation of the refrigeration plant is not always the most important issue for the profitability of the company. For companies selling hot water the efficiency of the plant is all important to the profitability. Even a small efficiency improvement makes a big impact for the heating company's bottom line.

Although R134a does not have any ozone depletion effect and is therefore not due to be phased out in the foreseeable future, there is always a risk of further restriction on the use of high GWP working fluids. Denmark and Austria [K Madsen, IOR 2009] have already demonstrated that life goes on beyond HFCs and this may influence future EU regulations. We have already seen the implementation of the F-gas directive in EU and depending on how effective it is in reducing the amount of F-gas used in the heat pump and refrigeration industry there could be further restriction for usage of HFC refrigerant in the future.

Heat pump is the principal equipment asset in district heating companies, who are investing a large amount in infrastructure and utilities around the installation of a heat pump and they need to ensure that their investment is a safe long term solution without the risk of government policy intervention.

Environmental impact

When evaluating the environmental impact of any solution both indirect and direct impacts need to be taken into account. Large heat pumps have an estimated charge of 500 kg/MW of R134a and 190 kg/MW of ammonia. The system charge of R134a has significant influence on the direct environmental impact. Even with the tightest system it is unavoidable to lose some of the gas in the system during service and maintenance. One of the sites who have installed a large heat pump reported a 1% loss of refrigerant per year, which compares with HFC supermarkets systems that are reported to have refrigerant losses in excess of 25% per year.

If Drammen had chosen a heat pump using R134a a loss of 1% of refrigerant per year equates to 97.5 tons of CO₂ equivalents per year. Any loss of Ammonia has minimal effect on global warming.

Chemical	GWP ₁₀₀ (kg CO ₂ eq/kg)
R-22	1700
R-134a	1300
R-404A	3260
R-717	<1
R-744	1

Figure 1 – Summary of some Life Cycle Climate Performance Data (UNEP 2006).

The main contribution to global warming is the indirect emissions through the electricity consumption. In Norway 95% of the power consumption is generated through hydro-electric power station. The CO₂ emission from electricity in Norway is only 0.014 kg/kWh compared to UK's 0.580 kg/kWh

The project

Drammen is a town 40 km south west of Oslo (Norwegian Capital). During the last decade it has gone through a major transformation from being a run down industrial town to a newly developed town centre with new hospital, housing, ice rink, hotels and shopping centre. These new developments have all been connected to a district heating network. The first district heating plant in Drammen was installed in 2002 using 8 MW biomass boilers.

With the second phase of the district heating network extension being a 15 MW of heat pump duty (for the base load) and additional 2 x 30 MW gas fired boiler (backup for the peak duties) have been installed. The maximum network peak heat demand is 45 MW duty.

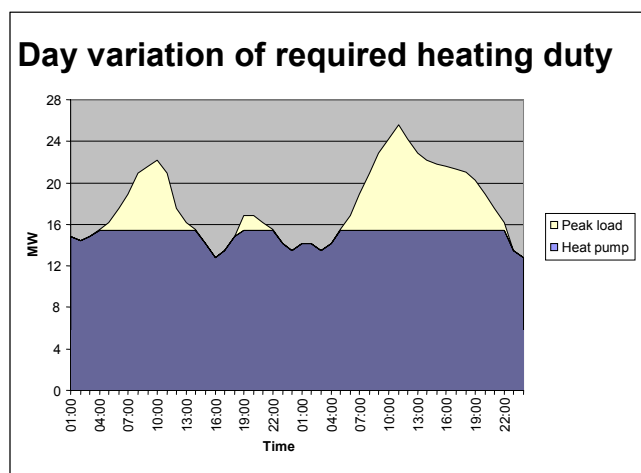


Figure 2 – Daily heatload variation

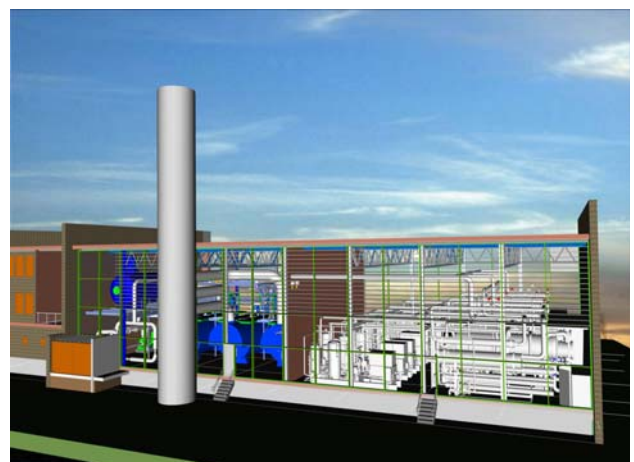


Figure 3 – 3D impression of Drammen facility

Figure 2 shows a typical heat load profile across a winter day in Drammen. Figure 3 shows the new District heating central with the gas fired boilers in blue to the left of the parting wall and the new ammonia heat pump to the right. Installation was completed Feb 2011.

The supply temperature of the district heating water varies across the year depending on the heat demand. In the summer time where there is a very small demand (less than 2 MW) the supply water temperature is 75°C, when the ambient temperature falls and there is an increase in heating demand, the supply water temperature increases up to 120°C at peak load. The return water temperature from the district heating loop is very steady at 60°C to 65°C all year around. When the gas boilers are being utilised they are working on a constant flow with temperature difference of 10°C between inlet and outlet. The water is then being mixed with the district heating water to achieve the desired outgoing temperature. To optimise the performance of the heat pump it was important to have variable flow system where the water is taken directly from the district heating return line as every degree subcooling is important and any degree overheating is wasted energy.

The heat source for the heat pump is sea water. Norway is famous a rocky coastline. The thermodynamic beauty of this landscape is that the water gets very deep just off the coast. When taking in the water at 40 m depth there is a constant water temperature of 8°C to 9°C all year. At this depth the water temperature is not affected by changes in the air temperature from +20°C in the summer to -20°C in the winter. The water intake pipe runs 800 m into Oslo Fjord and the return pipes are 600 m long to ensure that the 4°C outlet water is not mixed with the inlet water. The seawater pumps are situated on land but below sealevel.

The seawater is cooled directly in spray chillers, where ammonia is sprayed across titanium pipes with the seawater inside. Each tube is 19 mm in diameter and with water velocity of 3 – 4 m/s there is a low risk of sediments being left in the tubes or algae growth.

The ammonia heat pump that has been installed on site consists of 3 x 2 stage single screw compressor systems in series each with a heating duty of approximately 4.5 MW.

R134a properties compared to R717

R134a has a critical temperature of 101.30°C, when operating at condensing temperatures around 90°C, we are operating close to critical point and the condensing heat through phase change from gas to liquid is decreasing significantly (3.7% per degree) for each degree increase in condensing temperature. R717 has a critical temperature of 132.28°C and small changes in condensing temperature have less influence (less than 1% per degree) on the condensing heat.

	Critical temperature	Critical Pressure	$\Delta h_{\text{cond}} @ 70^{\circ}\text{C}$	$\Delta h_{\text{cond}} @ 80^{\circ}\text{C}$	$\Delta h_{\text{cond}} @ 90^{\circ}\text{C}$
Units	$^{\circ}\text{C}$	Pa	kJ/kg	kJ/kg	kJ/kg
R134a	101.30	4065	124.01	106.63	83.04
R717	132.28	11304	939.77	877.80	808.50

Figure 4 - Thermodynamic properties for ammonia and R134a at high condensing temperature.

From the Figure 4 it shows that if there is no subcooling of the liquid there is required to be circulated 10 times the mass flow of R134a compared to R717 at 90°C condensing temperature. The gas density is higher for R134a (216.60 kg/m³) than for R717 (42.91 kg/m³), but it does not outweigh the difference in mass flow and you would still need a 92% bigger volume flow (larger compressor) to achieve the same heating duty.

The compressor technology used for R134a systems is multistage centrifugal compressors, which can handle large capacities. This compressor type is good for handling large volume flow with high molecular weight but less efficient at high differential pressures and therefore good for R134a (102.03g/mol) heat pumps with a total differential pressure of 30 bar and less efficient for ammonia (17.03 g/mol) with a differential pressure of 46 bar.

At the project in Drammen where there is 30°C temperature difference between the inlet and outlet temperature of the district heating water it is possible to achieve approximately 25°C subcooling of the refrigerant from the condenser. Subcooling is advantageous for heat pump systems as this is additional heating capacity without increasing volume flow and hence absorbed power (in effect “free heat”). All it requires if the return water temperature from the heating loop is low enough is a large evaporator as one kW subcooling equates to one kW additional evaporation duty but no additional compressor power consumption.

When evaluating the different refrigerants it is important to take the subcooling and de-superheat into account. The much higher mass flow of R134a compared to R717 is reflected in a much higher heat capacity per degree subcooling and de-superheat. The discharge gas temperature for ammonia is significant higher than for R134 [Stene 2008], for this system with a condensing temperature of 90°C the discharge gas temperature of ammonia is approximately 120°C compared to 98°C for similar R134a compression system.

Below is the calculation of the enthalpy change of ammonia at 90°C condensing pressure being cooled from 120°C super heated gas to 65°C sub cooled liquid and a similar calculation of the enthalpy change of R134a at 90°C condensing pressure being cooled from 98°C super heated gas to 65°C sub cooled liquid.

$$\text{Heating capacity per kg of ammonia} = h_{P,T\text{discharge}} - h_{P,T\text{subcooled}} = h_{90^{\circ}\text{C},120^{\circ}\text{C}} - h_{90^{\circ}\text{C},65^{\circ}\text{C}}$$

$$\text{Heating capacity per kg of ammonia} = 1586 \text{ kJ/kg} - 507.1 \text{ kJ/kg} = 1078.9 \text{ kJ/kg}$$

$$\text{Heating capacity per kg of R134a} = h_{P,T\text{discharge}} - h_{P,T\text{subcooled}} = h_{90^{\circ}\text{C},98^{\circ}\text{C}} - h_{90^{\circ}\text{C},65^{\circ}\text{C}}$$

$$\text{Heating capacity per kg of R134a} = 292.2 \text{ kJ/kg} - 145.4 \text{ kJ/kg} = 146.8 \text{ kJ/kg}$$

Taking the superheat and subcooling into account there is only required 45% more volume gas flow with R134a compared to ammonia. When calculating the efficiency of the two different systems based on all components offering same efficiency for the two system, the theoretical efficiency of the two different system is almost the same when having 25°C subcooling and condensing at 90°C, whereas it without taking the subcooling at de-superheat into account the efficiency of ammonia heat pump is approximately 40% better than R134a [Coolpack].

Heat exchangers

The customer requirement was for all heat exchangers to be shell and tube. All tubes in contact with the district heating water are stainless steel 304 and the evaporator tubes are in titanium. These materials offers the highest integrity of the system. For R134a system it is possible to use Copper Nickel tubes (CuNi10Fe). According to Copper development association [CDA Publication TN38, 1986] these tubes do not offer as high integrity as titanium tubes. Especially at high water velocity as is required in the evaporator to avoid algae growth and sediments settling in the tubes, there is a higher corrosion potential with Copper Nickel tubes.

Material	General Corrosion	Deposit Attack	Crevice Corrosion	Impingement Attack	Behaviour in the Presence of Sulphide Pollution	Effect of Chlorination	Marine Fouling	Mechanical Strength	Fire Resistance	Fabrication and Component Availability
90/10 Copper-Nickel CuNi10Fe1Mn	2	2	2	1	1	2	2	1	2	2
Carbon Steel	1	1	1	1	1	1	0	2	2	2
Austenitic Stainless Steel	2	0	0	2	1	1	0	2	2	2
Austenitic Stainless Steel *	2	1	1	2	1	1	0	2	2	1
Duplex Stainless Steel *	2	1	1	2	1	1	0	2	2	1
Ferritic Stainless Steel *	2	1	1	2	1	1	0	2	2	1
Plastics	2	2	2	2	2	2	0	1	0	1
Titanium	2	2	2	2	2	2	0	2	2	1

0 – Not resistant

1 – Potential problems, limited life or some design etc reservations

2 – Satisfactory

* High Molybdenum

Figure 5 - Comparison of Various Pipeline Materials

When comparing the thermodynamic properties of ammonia with R134a using the same tube material it is shown that ammonia has approximately 50% better heat transfer coefficient at the same temperature difference across the tubes. To achieve the same close approach on the heat exchanges it is necessary with 50% larger heat exchanger surface area with R134a than with ammonia.

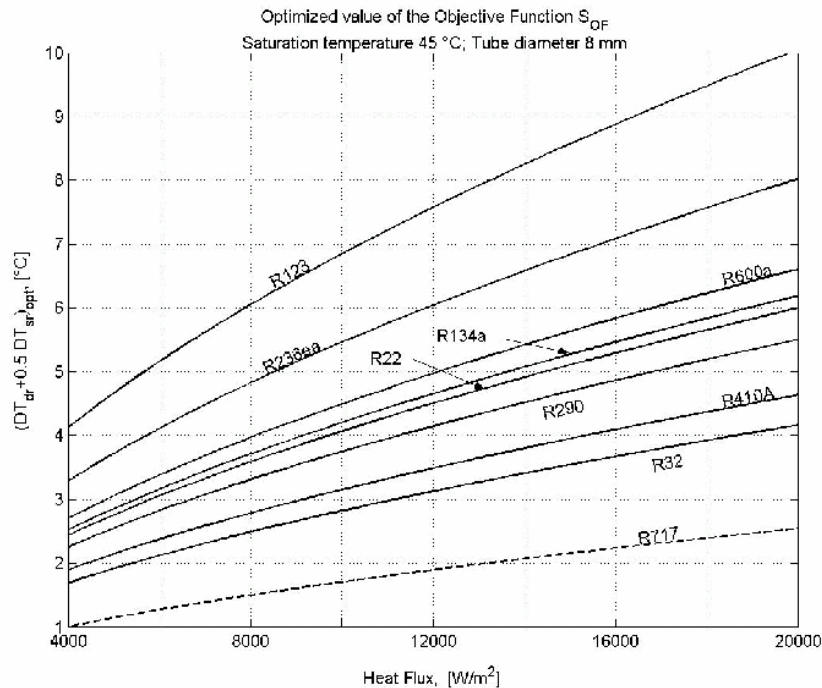


Figure 6 shows that for the same temperature difference across a vertical shell and tube condenser the heat transfer coefficient for ammonia is approximately 50% better than for R134a. [Prof. Alberto Cavallini, IIR2002]

All heat exchangers have been chosen with a close approach to optimise the performance. Condensers and evaporators have been chosen with only 2°C differences between water outlet and the condensing / evaporating temperature.

The intercoolers, sub coolers and oil coolers are all single pass shell and tube heat exchangers with counter flow, where the outgoing water temperature is higher than the outgoing refrigerant/oil temperature.

Compressor technology

The heart of the Drammen heat pump is the single screw compressor from Vilter (Emerson). Vilter has developed a high pressure version of their normal refrigeration compressor for gas compression systems. The single screw has balanced forces across the main rotor, both axially and radially.

The single screw compresses simultaneously on opposite sides of the rotor cancelling out the compression forces and balancing radial loads. This is in contrast to a twin screw where the compression is only on one side of the rotor, which creates a force trying to push the two rotors apart. The Vilter compressor isolates the discharge pressure from the end face and vents the suction pressure to discharge end of the rotor equalising any axial forces. The balanced forces give a very long bearing life at more than 120,000 hours for normal refrigeration compressors.

The compressor is manufactured in 3 different pressure ratings 76 bar (cast steel casing), 52 bar (nodular iron casing) and 36 bar (cast iron casing). The compressor is designed for a differential pressure up to 41 bar. The actuator operating the volumetric and capacity slides is operable up to a differential pressure of 26 bar. On projects like Drammen where 11 kV motors are utilised it becomes expensive to do capacity control with inverter drive so the pressure differential for each of the compressors are below 26 bar.

In addition to the actuated capacity slides, the volumetric slides are a significant advantage for heat pump systems. Where there can be big changes in both suction and discharge conditions across the

year, it is quite advantageous to be able to vary the volume ratio. These slides ensure that the compressor is always operating at optimum conditions. In addition the variable Vi can be adjusted during loaded running conditions independently of the capacity slides. This is not the case with twin screw compressors.

System design

For optimised performance of large scale heat pumps it is important to get the design right. The biggest challenge was to design the hot water flow through the heat pump to ensure that every kW is taken out of the system and at temperature where it is most useful.

With the water being heated from 60°C to 90°C the condenser part of the system is split into 3 off 2-stage systems working in series. The main water stream is being heated from 60°C to 69°C through the first condenser and from 69°C to 78°C in the second condenser and from 78°C to 87°C in the third condenser.

After the main flow has been heated to 87°C it is split into 3 streams going through the high stage de-superheater for each of the systems. The temperature is raised to 89°C through these heat exchangers.

Besides the main water flow there are separate streams of water going through subcoolers, high stage and low stage oil coolers and intercoolers. The intercoolers serve three purposes: they cool the superheated gas from the low stage compressors before entering the high stage compressors. Suction superheat reduces the isentropic efficiency of the compression. In addition the lower suction temperature gives rise to a lower discharge temperature thereby protecting the seals from too high a discharge gas temperature on the high stage compressors (maximum 135°C). The final reason of course is energy recovery.

With the main stream of water being mixed with the water from all auxiliary streams which has been heated to 92°C - 98°C the mixed outgoing water temperature from the heat pump is 90°C.

Although the three heat pumps are operating at different conditions the specifications for each of the three heat pumps are the same. This enables each of them to deliver 90°C water in case of a failure of one of the systems.

With 3 systems operating in a combination of series and parallel instead of simply parallel the average condensing temperature falls from 90°C to 80.5°C representing a 10% improvement in efficiency for the ammonia heat pump system.

71.9 Condensing Circuit	Suction Pressure		Press Diff Psi	Discharge Pressure		Discharge Temp °C	R717 Duty kW	Suction Temp °C	Discharge Mass Flow kg/s	Abs Power kW	Oil Cooling kW	System HOR kW	System Abs kW	System COP
	bar A	°C		bar A	°C									
Machine 1201 Cast Steel	19.4	48.3	225	34.9	72.4	114.5	3888	64.0	3.412	484.9	120			
VSS2101	4.4	1.0	224	19.9	49.3	99.3	327	1.0	3.201	805.8	281	4,616	1,342	3.44
80.7 Condensing Circuit	Suction Pressure		Press Diff Psi	Discharge Pressure		Discharge Temp °C	R717 Duty kW	Suction Temp °C	Discharge Mass Flow kg/s	Abs Power kW	Oil Cooling kW	System HOR kW	System Abs kW	System COP
	bar A	°C		bar A	°C									
Machine 1201 Cast Steel	23.5	55.8	274	42.4	81.2	115.1	3754	64.8	3.367	493.8	135			
VSS2101	4.4	1.0	285	24.1	56.8	104.6	415	1.0	3.279	939.5	389	4,693	1,491	3.15
88.5 Condensing Circuit	Suction Pressure		Press Diff Psi	Discharge Pressure		Discharge Temp °C	R717 Duty kW	Suction Temp °C	Discharge Mass Flow kg/s	Abs Power kW	Oil Cooling kW	System HOR kW	System Abs kW	System COP
	bar A	°C		bar A	°C									
Machine 601 Cast Steel	26.1	60.0	347	50.0	89.0	117.7	3662	65.0	3.343	528.3	155			
VSS2101	4.4	1.0	323	26.7	61.0	107.2	471	1.0	3.328	1016.2	453	4,740	1,606	2.95

Figure 7 - Operating conditions of each of the 3 heatpump systems at Drammen

Results from Drammen

The system is currently being tested and the COPh for the system is close to 3.3 which is better than expected.

The current gas prices in Norway are approximately £30 per MWh and the electricity prices is approximately £50 per MWh. By using the ammonia heat pump there is an estimated saving of £1,042,289 compared to using gas given the usage pattern shown in Figure 8. The system has shown to be 15% better than alternative heat pumps on R134a this equates to £159,534 per year.

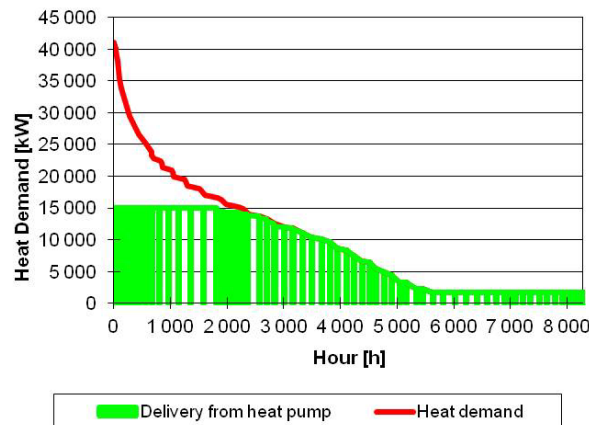


Figure 8 - Typical Daily Heatload served by heatpump and fossil fuel

The global warming benefit of the ammonia heat pump is also significant. With a yearly equivalent CO₂ emission of 317 tons, this compares to R134a 97.5 tons direct and 364 tons indirect or 46% more than the ammonia heat pump. Both these figures are low compared to burning gas which would give a CO₂ emission of 13,050 tons per year at the given usage profile.

Conclusions

With the availability of higher pressure compressors designed for natural refrigerants such as ammonia, the excellent performance of R134a and R22 heatpump systems can now be achieved without the significant environmentally damaging global warming impact associated with working fluid losses. The improved COP achieved with ammonia is a benefit particularly useful in a business with such emphasis on return on investment.

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