Evaluation of CO_2 supermarket refrigeration systems

Field measurements in three supermarkets

S A R A H   J O H A N S S O N

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Sarah Johansson

Master of Science Thesis Energy Technology 2009:489
KTH School of Industrial Engineering and Management
Division of Applied Thermodynamics and Refrigeration
SE-100 44 STOCKHOLM

KTH Industrial Engineering and Management	Master of Science Thesis EGI 2009/ETT:489 **Evaluation of CO_2 supermarket refrigeration systems** **Field measurements in three supermarkets** Sarah Johansson	
Approved	Examiner Björn Palm	Supervisor Samer Sawalha
	Commissioner	Contact person

Abstract

This thesis is a part of a larger project where supermarket refrigeration systems with CO_2 will be evaluated and compared to more conventional systems without carbon dioxide. This will give an indication if CO_2 can be used in supermarkets and if the technology is ready to be more commercialized. In this thesis three supermarket using carbon dioxide as refrigerant has been studied. The systems under investigation are two trans-critical systems and one cascade system with R404A in the high stage. The supermarkets are fully instrumented to measure temperature, pressure and power. This makes it is possible to calculate the COP and capacity of the systems.

Carbon dioxide has many advantageous properties for being a refrigerant, it is environmental friendly, easy accessible and cheap, it has great thermal properties and allows the piping to be reduced in size. The main problem is that CO_2 works with high pressures and has a low critical point of 31°C. If the CO_2 operates at higher temperatures it will be trans-critical and there will be a loss in the cooling capacity and efficiency.

The result shows that the capacity and the COP of the system are dependent on the outside air temperature. During summer the COP is lower than it is in wintertime and that is due to the higher condensing temperature that is necessary but also due to that the humidity in the air is higher. In one of the systems some changes was made which allowed the system to condense at lower temperature, which increased the COP.

The oil cooler capacity has been a great challenge and the results shows that the oil cooler capacity has an impact on the COP of the system. It is higher in the winter than in the summer. One reason to this could be due to the higher surrounding temperature, which are not able to cool down the oil as much as in the wintertime. Further analysis of the oil cooler is needed in this project.

This thesis shows that it is difficult to compare refrigeration systems due to different operating conditions and system requirements. Therefore it is of importance to combine experimental work with computer simulations where the influence of different system parameters can be investigated separately.

In environmental aspect carbon dioxide is a great refrigerant to use in supermarkets. The overall COP for medium and low temperature side is between 3-5 and 1,2-1,8 respectively, depending on the season. Further analysis and comparison to more conventional supermarkets with different parameters is needed.

Acknowledgements

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Samer helped me with my calculations and they both gave me valuable advices and information about the systems.

I would also like to thank the owners of the supermarkets for letting us visit and do valuable measurements in the refrigeration systems.

Thank you!

Sarah Johansson
Stockholm, June 2009

Table of contents

Nomenclature

CC1	Cascade system 1
CFC	Chlorofluorocarbons
CO_2	Carbon Dioxide
COP	Coefficient of Performance
CP	Critical point
DX	Direct expansion
EES	Engineering Equation Solver
FA	Freezing unit
GWP	Global Warming Potential
HCFC	Hydrochlorofluorcarbons
HFC	Hydrofluorocarbons
HC	Hydrocarbons
HVAC-system	Heating, ventilating and air conditioning system
IHE	Internal heat exchanger
KA	Cooling unit
kW	Kilowatt
NH_3	Ammonia
ODP	Ozone Depleting Potential
PR	Pressure ratio
R404A	A common refrigerant that is a mixture of other refrigerants
R774	The name of carbon dioxide when its used as a refrigerant
R290	The name of propane when its used as a refrigerant
Q	Cooling capacity
TR1	Trans-critical system 1
TR2	Trans-critical system 2
°C	Degrees Celcius

Index of figures

Index of tables

1 Introduction

Global warming is a worldwide challenge nowadays and industries are fighting these problems by being more concerned of from where their energy is coming from and how they can contribute to a more sustainable interaction with the environment.

Related to the global warming impact caused by the refrigeration industry, supermarkets are main contributors by two ways, indirectly and directly. Indirectly with their high energy consumption, mainly due to the large use of energy to run the refrigeration systems. They are directly contributing by leaking refrigerants with high ozone depleting potential and high global warming potential (Campbell et al., 2006). New technologies for more efficient systems have progressed and the interest for using natural refrigerants has distinctively increased in the last few decades, mainly for carbon dioxide.

Carbon dioxide was already in use in the beginning of the 20th century, mainly in marine systems. Due to technical problems with high operating pressure and low critical temperature as well as the capacity and efficiency loss at high temperatures synthesized refrigerants, CFCs and later HCFCs, started to dominate the market and replaced most of the Carbon dioxide applications (Granryd et al., 2005). The halocarbons, CFCs and HCFCs, with their non-combustibility and technical properties operated at much lower pressures and where considered to be environmentally benign. They were shortly proven to be harmful to the environment as global warming gases and with their high ozone depleting potential (ODP). Therefore huge efforts were made for finding replacements. The CFCs were banned by the Montreal protocol in 1987 and replaced by the Chlorine-free substances, HFC. A few years later HCFCs were included in the Montreal protocol and today HFCs are dominating the industry. They have no ozone depleting potential (ODP) and even though their global warming potential (GWP) is relatively high it is still lower than the GWP for HCFCs and CFCs.

In the end of 1980s Norwegian professor Gustav Lorentzen had a proposal about re-introducing carbon dioxide as a refrigerant. Among the natural refrigerants, carbon dioxide is the only non-flammable and toxic free fluid that can operate in a vapour compression cycle below 0°. Therefore it is the most viable option for supermarket application (Kim et al., 2003). The major concern with the use of CO_2 earlier was the high operation pressure. Nowadays, when the availability of special designed components to handle CO_2 increased, some favourable thermophysical properties attached to the high operating pressure can be utilized (Sawalha, 2005). High pressure results in a high fluid density throughout the cycle, which reduces the size of the components and tightens up the system (Nekså, 2004).

In the beginning carbon dioxide was only operating as a secondary fluid in indirect system solutions in supermarkets. With more knowledge about CO_2 and more advanced technology, cascade solutions with CO_2 in the low stage as well as trans-critical solutions with CO_2 became viable options. There where at least 20 supermarkets in Sweden running with the trans-critical solution, by 2008. (Sawalha, 2008).

1.1 Energy use in supermarkets

Supermarkets are large consumers of energy. It has been shown that they use 3-5% of the total national energy use. It is estimated that the refrigeration system use 35-50% of that energy and thereby indirectly affecting the environment. The trend in Supermarkets indicates that the need for cooling will increase, resulting in an increased energy use if the systems do not become more efficient (Axell et al., 2004). Figure 1.1 is a diagram of the energy use in a Swedish supermarket (Axell, 2001). Lighting uses about 20% of the energy and about 30% of the energy goes to other application such as HVAC-system.

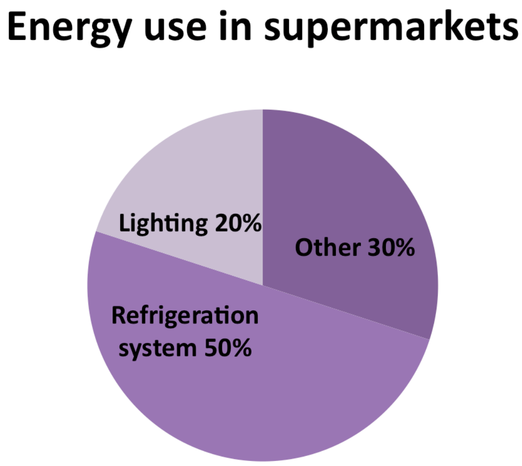


Figure 1.1 - Energy usage for a Supermarket in Sweden (Axell, 2001)

Many arrangements can be done when it comes to improving the energy use in the refrigeration systems. The design of the cabinets has a big influence of the energy consumption of the system. The retailers prefer large, open cabinets to be able to display the food in an attractive way for the customers. These open cabinets have a high infiltration of the warm ambient air forcing the system to work hard for keeping the temperature necessary for the products (Axell et al., 2004). It is estimated that 60-70% of an open vertical display cabinet losses is due to the infiltration (Axell, 2001). To reduce these losses glass lids can be used for the cabinets. The energy use is also affected by the way the food in the cabinets and freezers are stored. Wrong packing prevents the cold air from circulating correct and maintain the cold temperature of the groceries. Another well-known factor affecting the system load is the indoor climate. A high humidity and high surrounding temperature requires more energy to maintain its coldness. Still there has to be a certain humidity and temperature in a supermarkets for the customers to feel thermal comfort. The temperature and humidity in the air is higher during the summer. A high humidity increases the frost accumulation in the cabinets and they need defrosting more often to be able to work properly.

The refrigerant used has an influence on the energy use. To choose a refrigerant with good properties, such as good heat transfer properties etc, could reduce the energy use. The supermarkets are also directly affecting the climate changes by releasing refrigerant to the atmosphere. It is estimated that the annual refrigerant loss is 15-30% of refrigerant charge (heatpumpcentre.org, 2008). Changing to a better refrigerant could both decrease the energy consumption and reduce the effect on the environment by being friendlier for the atmosphere when released.

1.2 Objective

1.2.1 The project

This Master of Science thesis work is a part of a project that aims to evaluate refrigeration systems in supermarkets using carbon dioxide as refrigerant. Three different types of refrigeration systems using carbon dioxide will be evaluated. Indirect system with CO_2 pump circulation, cascade solutions and trans-critical systems with only carbon dioxide. The efficiency of the CO_2 refrigeration systems will be compared to more conventional systems to indicate if the technology with carbon dioxide as a refrigerant is profitable or if it still needs to be improved before it can be commercialized in larger scale. This study will also result in more knowledge about the application of trans-critical carbon dioxide in supermarkets refrigeration systems, which is a relatively new technology in the market.

To fulfil the objective in this project field measurements are done in a number of supermarkets in Sweden. The supermarkets use different system solutions, indirect, cascade and trans-critical.

The company that is in charge of the project is IUC Sveriges Energi och Kylcentrum. It is financed by the Energy Agency and consists of a project group with people from the refrigeration industry. The companies that are represented in the group is:

- ICA Sverige AB
- Ahlsell Kyl
- Huurre AB
- WICA Cold AB
- Kyl AB Frigoväst/Green & Cool
- AGA Gas AB
- Tranter
- Cupori AB
- Oppunda Svets och Mekanik AB

1.2.2 Master thesis

In this Master of Science thesis work following systems will be evaluated:

- Two trans-critical refrigeration system with carbon dioxide, referred to as:
  - Trans-critical system 1
  - Trans-critical system 2

- One cascade refrigeration system with R404A and carbon dioxide, referred to as:
  - Cascade system 1

The systems will be evaluated one by one but a short comparison will be done in the end of this report.

2 Methodology

In the beginning of this thesis some preparatory research was done, mainly by reading reports and articles about carbon dioxide as a refrigerant in supermarkets and its benefits but also about different system solutions. This knowledge was used as an introduction to this report.

Instrument for measurements has been installed in the supermarkets and data are automatically recorded. Data of interest are pressure and temperature in specific points identified before the installations. The energy consumption of the compressors in the system is also a key parameter for the evaluation. These values are stored in a database easily accessed by a web-based program called Iwmac (Iwmac, 2009). The measured values are logged for every five minutes. In the cascade system another type of logger called RDM was used which saves data for every 15 minutes.

For every system the cooling capacity and the coefficient of performance is calculated for every five minutes for the two trans-critical systems and for every 15 minutes for the cascade solution. It is then reduced to an average for every month. The monthly average of COP is studied for every system to see how different parameters influence the COP. There is also a shorter time period of one day in the analysis since the performance of the system is changed between opening-hours and night-time. The Analysis is done for one day in wintertime and one day during summer. The systems are evaluated based on the calculations and some assumptions that had to be made.

3 Refrigerants in supermarkets

There are mainly two groups of refrigerants, synthetic and natural. The synthetic refrigerants belong to the groups mentioned earlier, CFC, HCFC and HFC. One of the most common synthetic refrigerants used in supermarkets today is R404A. It is a mixture of HFCs refrigerants.

To be able to measure how the refrigerants are affecting the ozone layer and the amount of greenhouse gases, the ozone depleting potential (ODP) and global warming potential (GWP) is used (Naturvårdsverket, 2003). This makes it possible to compare different refrigerants in the environmental aspect. The ODP is based with R11 as reference (ODP of R11 is 1) and GWP is based on carbon dioxide (R-744). GWP relates how much 1 kg of a refrigerant released equals in kg of CO_2 release (GWP for CO_2 is 1). Table 3.1. Shows a comparison between different refrigerants where the natural fluids, ammonia (R717), carbon dioxide (R744) and propane (R290) are listed.

Type	Refrige-rant	GWP	ODP	Flamm-able	Toxic	Critical temp. (°C)	Critical Pressure (bar)	Vol. capac-ity
CFC	R12	8100	0.9	No	No	112	41.4	1
HCFC	R22	1500	0.055	No	No	96.2	49.9	1.6
HFC	R134a	1300	<0.0005	No	No	101.1	40.6	1
	R404A	3260	<0.0003	No	No	72.1	37.4	
Natural ref.	R717 (Ammonia)	<1	0	Yes	Yes	132.3	113.3	1.6
	R774 (CO_2)	1	0	No	No	31.1	73.8	8.4
	R290 (Propane)	20	0	Yes	No	96.7	42.5	1.4

Table 3.1 - Properties of CO_2 compared to other refrigerants (international institute of refrigeration, 2000; Granryd et al., 2005)

3.1 Requirements for refrigerants

There are certain requirements that need to be satisfied for fluids to be able to work as a refrigerant. They should have suitable thermodynamic properties, good heat transfer, reasonably low or no GWP and no ODP etc, but should also be inexpensive and easy to access. It is important that there is a chemical stability within the refrigeration system in order for it to operate properly and that the refrigerant does not react with the material used for the piping and components. All of the requirements cannot be fulfilled for one refrigerant but different refrigerants have different advantageous and drawbacks and are therefore suitable for different application.

The operating pressure in the systems should not be too high and not too low. A pressure below atmospheric pressure is not desirable since in case of a leakage air might be sucked in to the system and cause problems.

Oil is often a part of a refrigeration system for lubrication of bearings and other parts in the compressor and it is of importance that the refrigerant is compatible with the oil (Granryd et al., 2005).

3.2 Natural refrigerants

Common for the natural refrigerants is that they have an insignificant GWP and no ODP and as the name indicates they can all be found existing in nature. The used natural refrigerants are mainly water, air, hydrocarbons, ammonia and carbon dioxide.

Water is sustainable, absolutely harmless to people and nature, easily accessible and easy to dispose after use, excellent conditions for being a refrigerant. The problem is that it requires large volume flow and high pressure ratios. Furthermore, the high freezing point limiting the evaporating temperature to above 0°C is another disadvantage. Therefore water is not considered to be applicable in supermarket refrigeration (Kharazi et al., 2005).

Air is not attractive to use in refrigeration systems due to the fact that the air gives such a low efficiency and cannot compete to conventional refrigeration cycles (Pearson, 2003).

Methane, ethane, propane (R-290), propylene (R-1270) and iso-butane (R-600a) are all relevant as refrigerants and belong to the group of **hydrocarbons.** They have relatively high critical pressure and are miscible with oil. The main problem with hydrocarbons is their high flammability and explosive characters, thus limiting their use to smaller systems, such as household refrigerators (Pearson, 2003). The small charge will minimize the chance of forming an explosive mixture with air if a leakage would occur (Granryd et al., 2005).

Ammonia (R-717) is a fluid with eminent reasons for being used as a refrigerant, good heat transfer characteristics, inexpensive and easy to manufacture. Furthermore it has a GWP and ODP of zero (Naturvårdsverket, 2003). Due to its high toxicity and flammability it is not optional as a working fluid in systems running in occupied spaces but it is still possible to use it unoccupied spaces and outside. Ammonia has a distinguished smell that, in fact, is an advantage since a small leakage could immediately be detected. Ventilation systems are required in case of using ammonia and they should be placed close to the ceiling since ammonia is lighter than air (Granryd et al., 2005). Ammonia is not compatible with copper therefore steel pipes are used in ammonia systems (GTZ Proklima, 2008).

4 Carbon dioxide as a refrigerant

4.1 Properties of carbon dioxide

Carbon dioxide (CO_2) is a natural refrigerant with no ozone depleting potential (ODP) and an insignificant global warming potential (GWP=1) compared to other synthetic fluids (Arias et al., 2005). Carbon dioxide is classified in group A1 by ASHRAE standard, a group with non-flammable and non-toxicity at concentrations below 400 ppm (Granryd et al., 2005). When carbon dioxide is used as refrigerant it is known as R744. It is an odourless, non-flammable and non-toxic fluid, favourable properties as a refrigerant (Danfoss, 2008).

Its main characteristics are the relatively low critical temperature at 31,1°C at a relatively high critical pressure of 73,8 bar. Above that critical point CO_2 behaves as a supercritical fluid and shows properties of both a liquid and a gas. In the supercritical stage the pressure and temperature are independent and can be regulated separately to get the optimum conditions (Kim et al., 2003). At temperature higher than the critical temperature gas cannot be transformed into liquid by a change in only pressure. The triple point of carbon dioxide, where the three phases solid, liquid and vapour can coexist in equilibrium, occurs at a temperature of -56,6°C and a pressure of 5,2 bar. CO_2 is the only common refrigerant to have a triple point above atmospheric pressure (Butler, 2007). Figure 4.1 presents the phases of carbon dioxide in a pressure and temperature diagram.

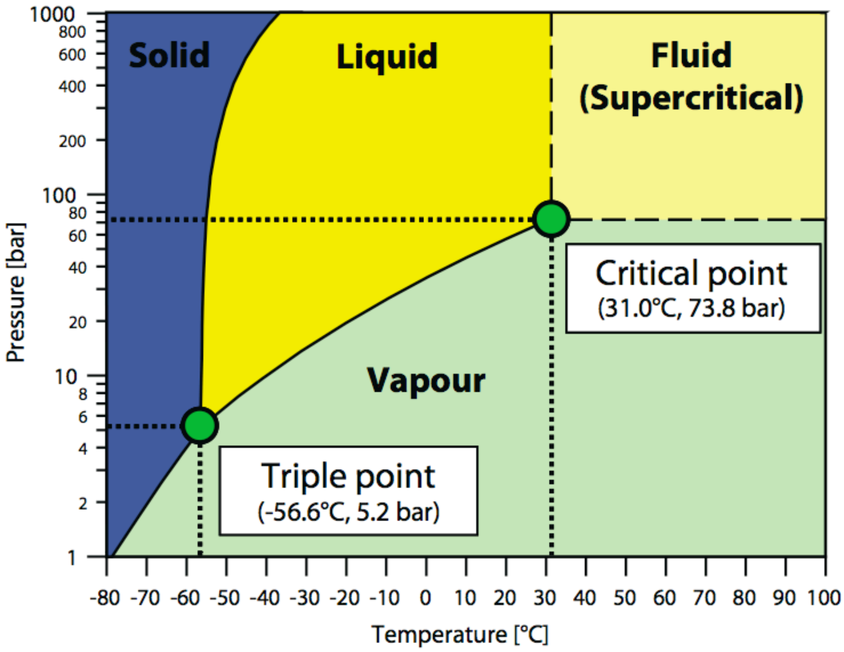


Figure 4.1 - Phase diagram for carbon dioxide (Danfoss, 2008).

4.2 Advantage and disadvantages

There are many benefits from using carbon dioxide as a refrigerant in supermarkets apart from the environmental advantages. Since it is not poisonous or flammable it is advantageous to use in supermarkets where the refrigerant circulates in long distribution lines through the public space. It is not damageable to food and in case of contact, the food is still consumable (Axima refrigeration, 2008). Carbon dioxide is denser than air so if a leakage would occur the highest concentration will be found close to the floor. Even though CO_2 is classified as a non-toxic and non-flammable refrigerant it can still be harmful to human at high concentrations. Since it is heavier than air it can displace the oxygen in the surrounding space. Carbon dioxide is colourless, odourless and hard to discover if it leaks and good ventilation and special detection systems for leakage are required. They should preferably be place in the lower level in the room (Sawalha, 2005).

Carbon dioxide is easily available since it is a waste product in many industries. Some industrial companies have developed systems to extract CO_2 from their plants to process it and sell it as refrigerant (R774.com, 2008). Due to this quantity of CO_2 it is inexpensive as refrigerant.

CO_2 operates at very high pressure resulting in high vapour density. This results in a higher volumetric refrigeration capacity compared to refrigerants with the same latent heat of vaporization (Sawalha, 2005). It has a capacity that is 3-10 times larger than HCFCs, HFCs, CFCs and HC refrigerants (Kim et al., 2003). This allows the components such as pipes and compressor in the system to be minimized, which also gives a reduction in the amount of material used during manufacturing. Another benefit is that carbon dioxide does not react with metal as other refrigerants usually do and any metal able to restrain the high operation pressure can be used. Carbon dioxide is also compatible with oil (Arias et al., 2005).

CO_2 has excellent thermodynamic properties. It has a small density ratio between liquid density and vapour density. At 0°C, for instance, the liquid density is 927 kg/m^3 and the vapour density is 98 kg/m^3 giving a ratio of around 10. Other refrigerants such as R410a and R134a have a ration of 65 and 89 (Kim et al., 2003). The low-density ratio and the low surface tension results in a superior heat transfer coefficient. That contributes to smaller heat exchanger and close approach temperatures in CO_2 refrigeration systems (Irhace, 2008).

In a negative point of view the high operation pressure required for carbon dioxide demands good system components with high quality and the ability to handle the high pressures. They are harder to find but they have recently become more available and price effective. They will probably become even cheaper if the technology becomes more commercialized and the components can be manufactured in a larger scale (Butler, 2007). In Figure 4.2 a diagram is showing the high operating pressure for CO_2 compared to other commonly used refrigerants.

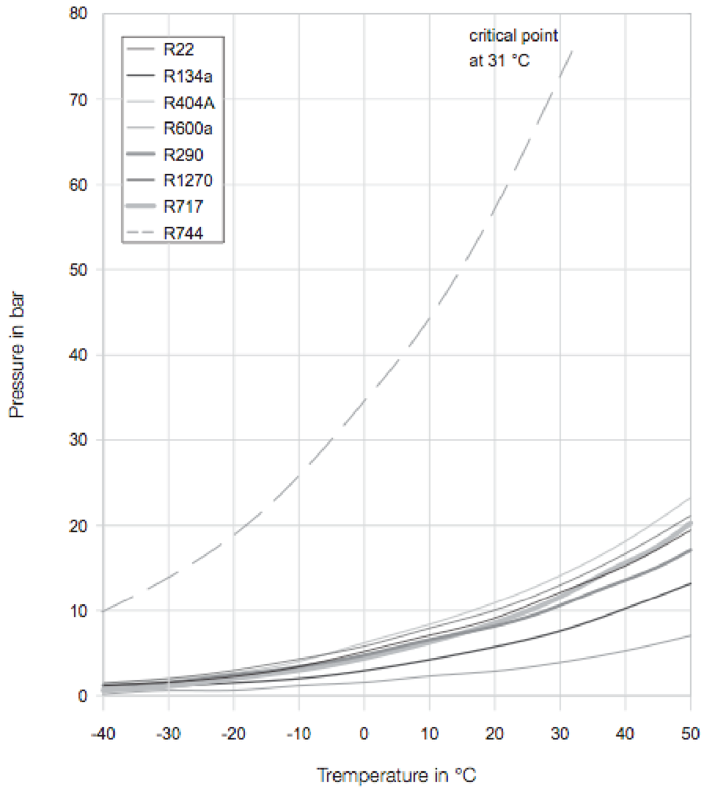


Figure 4.2 - Temperature and pressure diagram showing the vapour pressure curve of some common refrigerant e.g carbon dioxide (R744) (GTZ Proklima, 2008).

A previous problem contributing to the phase out of carbon dioxide was the low critical temperature of 31,1°C. It is still a problem for some applications. When rejecting heat above that critical temperature i.e. CO_2 is operating as trans-critical, there is a loss in cooling capacity and efficiency (GTZ Proklima, 2008).

4.3 System solutions for carbon dioxide in Supermarkets

Refrigeration systems in Supermarkets with carbon dioxide can be designed in several different ways. Carbon dioxide can be used as a secondary refrigerant (without compression) or as a refrigerant (with compression).

Carbon dioxide can be used as the only refrigerant in a system and operate in a trans-critical cycle or it can be combined with others in a cascade arrangement.

4.3.1 Indirect system

In an indirect system two fluids are used, one primary refrigerant and one secondary fluid (without compression). The refrigerant is operating in a conventional refrigeration cycle and evaporates by heat exchanging with the secondary fluid. The secondary fluid cools down the cabinets in a supermarket and is circulated by pumps.

In an indirect refrigeration system carbon dioxide can be used as a secondary fluid since it is non-flammable and non-toxic and therefore well suited to circulate in public areas. The needed pumping power for CO_2 is lower than for other fluids due to the low pressure drop and small volume flow rate (Pearson, 2003). The benefits from using indirect systems is that refrigerants that are not wanted in systems close to human still can be used in the primary system since it can be kept in a safe sealed room (SWEP, 2008). A typical indirect system can be seen in Figure 4.3. The blue circuit is the secondary fluid i.e. carbon dioxide, going out to the cabinets and the green circuit is the primary refrigerant e.g. ammonia, in the machine room.

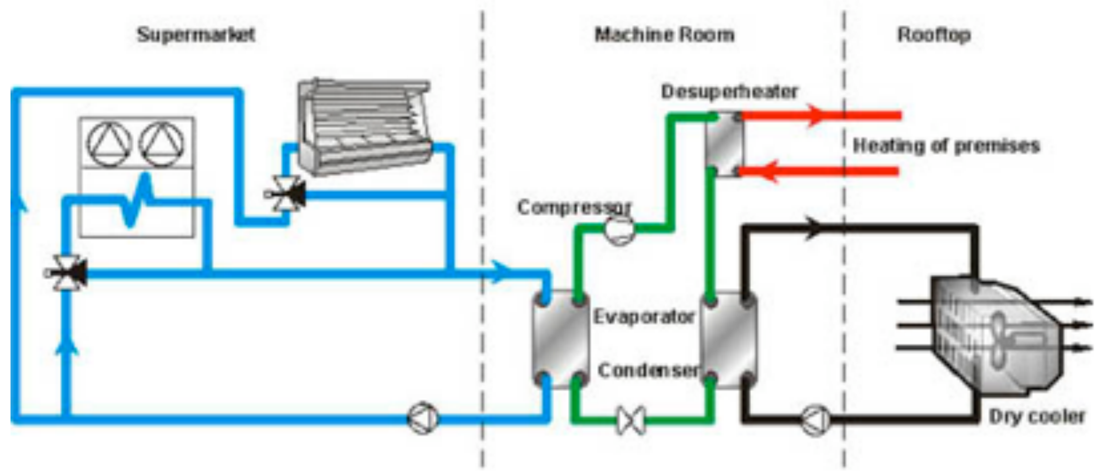


Figure 4.3 - Indirect system in Supermarket (SWEP, 2008).

4.3.2 Cascade system

In a cascade system two refrigerants are used. A difference from the indirect system is that both refrigerants are operating in separate refrigeration cycles. This arrangement is done to gain low temperatures without letting the compressor work in unfavourable conditions due to the large pressure ratios required at large temperature lift. It gives better efficiency than the one-stage compression systems. (Granryd et al., 2005). One fluid operates in the low stage and one operates in the high stage. The two cycles are connected by an intermediate cascade heat exchanger where the low stage refrigerant condensates and the high stage refrigerant evaporates. In this system carbon dioxide is suitable to operate in the low stage combined with e.g. ammonia, propane or R404A in the high stage (Sawalha, 2005). Carbon dioxide is working subcritically in a cascade system since the high stage refrigerant allowing it to condense below the critical point. Since carbon dioxide is used in the low stage the compressor size can be greatly reduced. Further advantage is the absence of a liquid pump (Kim et al., 2003).

A negative aspect with cascade systems is that an additional temperature difference is introduced between the two stages due to the cascade heat exchanger (Granryd et al., 2005). The temperature difference can be seen in figure 4.4 where the two cycles overlap. The blue cycle at the bottom is the low stage with CO_2. The red cycle at the top is a refrigerant working at the high stage.

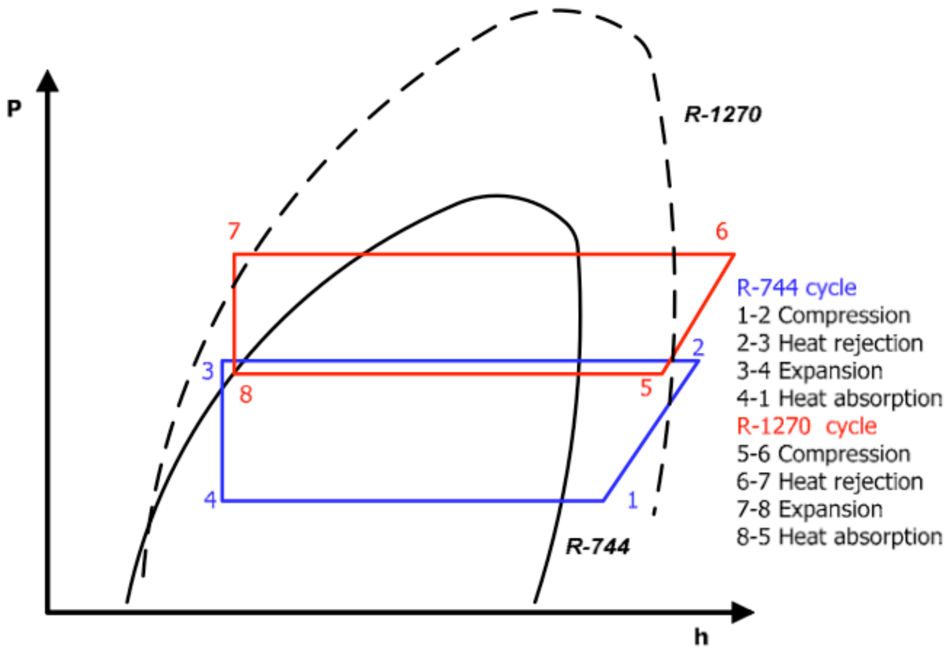


Figure 4.4 - Log P-h diagram showing operation of cascade refrigeration system (Campbell et al., 2007).

Cascade systems can be arrange in different ways. An alternative is to use carbon dioxide for the low temperature side and a brine for the medium temperature side that is heat exchanging with the high stage. The drawbacks with a medium temperature brine is that an additional heat exchanger with an extra temperature difference is needed. That can be avoided by letting the low stage cycle refrigerant e.g. CO_2, provide the medium temperature side as well as the low temperature side.

4.3.3 Trans-critical system

Carbon dioxide can be used as the only refrigerant in a refrigeration system. It can be a subcritical cycle working close to the critical point but when the ambient temperature is too high it operates as a trans-critical fluid meaning that it is subcritical on the low-pressure side and super-critical on the high-pressure side (Kim et al., 2003). Figure 4.5 compares a “traditional” subcritical refrigeration cycle to the left and a trans-critical process to the right.

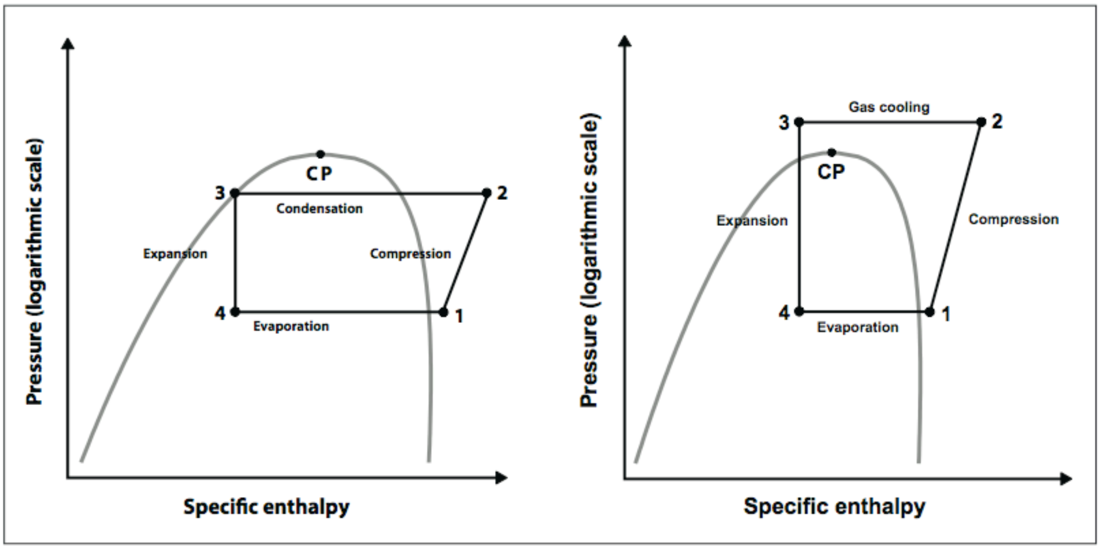


Figure 4.5 - Basic subcritical and trans-critical refrigeration cycles (Danfoss, 2008).

The main difference between a trans-critical cycle and a conventional cycle is that heat rejection can take place above the critical point (CP). A condition in the supercritical region, the region above the critical point, is often referred to as gas condition. Therefore the heat rejection process (between point 2-3 to the right in figure 4.5) is called gas cooling. The heat exchanger used in trans-critical processes on the high pressure side is called gas cooler (Nekså, 2004). The temperature is constantly changing during the gas cooling process which is different from the traditional refrigeration cycle where the condensing temperature is constant. The low side conditions in the trans-critical cycle stay subcritical.

The temperature is constantly changing during gas cooling. It can be very useful to use this temperature glide for heating water or air. If the gas cooler is designed correctly the temperature curve for the hot water can be adjusted to match the gas cooling temperature curve. This gives better and in some cases higher COP and is the reason for the success of using CO_2 in hot water heat pumps (Nekså, 2004).

The ambient air temperature affects the performance of the refrigeration systems. Systems with only carbon dioxide are most suitable for colder climates or where cold heat sinks are available since it can operate subcritical a major part of the time.

The benefit from using CO_2 as the only fluid is that there is no temperature difference between the fluids as it is in the cascade system (Sawalha, 2005).

A negative aspect with the trans-critical process is that the operating pressure is very high. In a conventional refrigeration cycle the discharge pressure is determined by the condensing temperature and is affecting the compressor work and thereby the COP. It is known that, in general, a high condensing pressure results in a loss in COP. The condensing temperature is the main function determining the enthalpy at the exit of the condenser (point 3 to the left in figure 4.5) and thereby affecting the COP. In a trans-critical process it is a bit different. No saturation conditions exist in the supercritical region and the pressure and temperature are independent and can be regulated separately. This means that both pressure and temperature have influence on the cooling capacity and COP of the system. An optimum pressure can be estimated to find the highest COP. The isotherms in a log P-h diagram are very steep close to the critical point (red curve in figure 4.6). A small change in pressure (small increase in energy input to compressor) can give a larger change in cooling capacity hence results in a higher COP. When the increase in cooling capacity no longer can compensate for the increment in compressor work the maximum COP is reached (Kim et al., 2003). Every trans-critical cycle has an optimum value for the high-side pressure that can be controlled by various methods to retain the optimum conditions. Figure 4.6 shows that different gas cooling pressures give different cooling capacity.

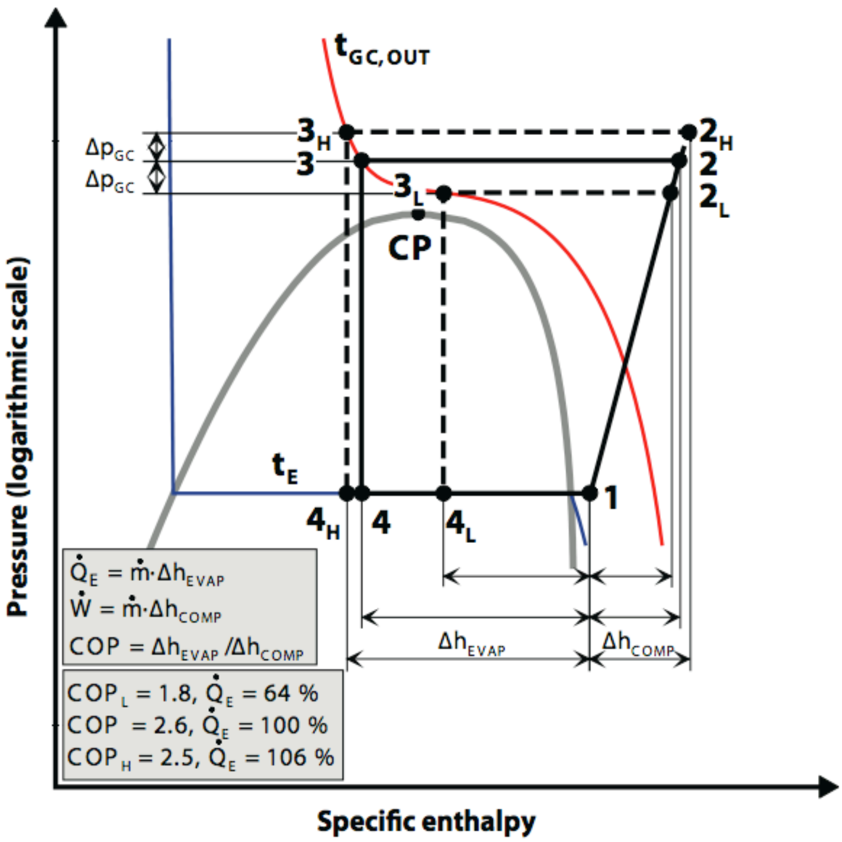


Figure 4.6 - Log P-h diagram for carbon dioxide. Different gas cooling pressure results in different COP (Danfoss, 2008)

An important factor to consider when deciding to use CO_2 is the high pressure in the system when the operation is shut down due to maintenance, failure, power cut etc. The carbon dioxide will gain heat from the surroundings hence increase the pressure in the system (Sawalha, 2005). The most common way to solve this problem is to release some of the CO_2 charge to the ambient when the pressure reaches a certain limit. The refrigerant is released through a relief valve, hence decreasing its pressure and temperature (GTZ Proklima, 2008). This procedure will be repeated if the system remains at standstill for a longer time. It is of big importance that the relief valve is placed correctly in the system so no liquid CO_2 will run through it. In case of liquid CO_2 running through the valve its pressure will decrease from plant pressure to atmospheric pressure (≈1 bar). The phases of CO_2 will then run through the triple point hence solid ice will be formed and might block the valve. This process could be advantageous in other places in the system since the solid ice might seal or shrink the leakage. Figure 4.7 shows the phase changes for CO_2 when solid ice is formed.

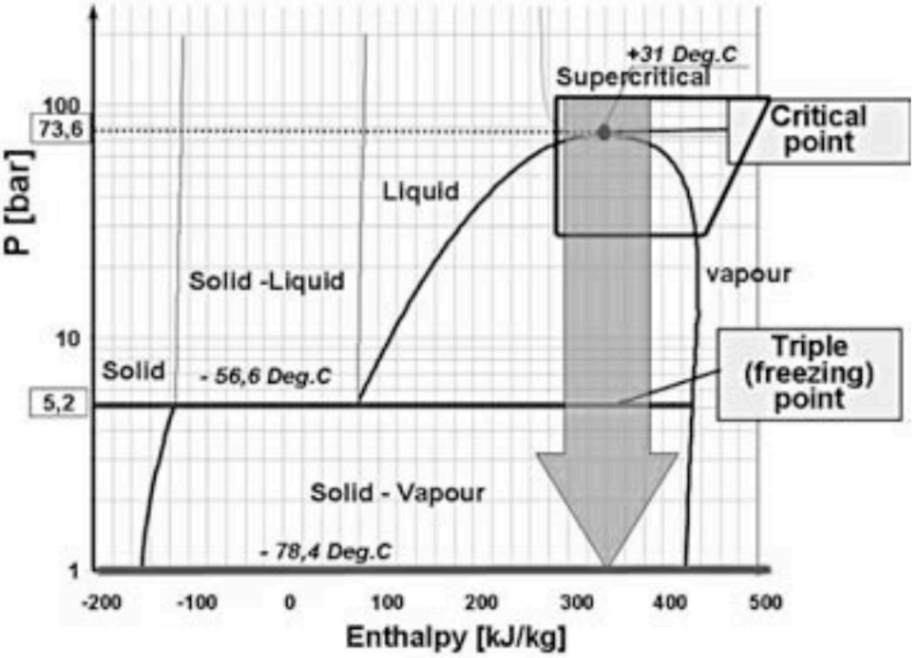


Figure 4.7 - Phase change for CO_2 when solid ice is formed (GTZ Proklima, 2008).

There are relatively large expansion losses between the gas cooler outlet and the evaporator inlet due to the high operating pressure. To further improve the efficiency of the trans-critical system these losses need to be reduced (Topping, 2004).

5 Test systems and evaluation method

In this Master of Science thesis two trans-critical systems and one cascade system are evaluated. The supermarkets will be named trans-critical system 1 and 2 (TR1 and TR2) and cascade system 1 (CC1). They are located in different places in Sweden meaning that the outside air temperature will be different for the different supermarkets. TR1 is far north and TR2 is the most southern supermarket located on the west coast. CC1 is located south west of Stockholm.

5.1 Trans-critical system 1

The first system is a CO_2 trans-critical system with direct expansion (DX) on both the medium and the low temperature sides. A schematic diagram of the refrigeration system is shown in figure 5.1.

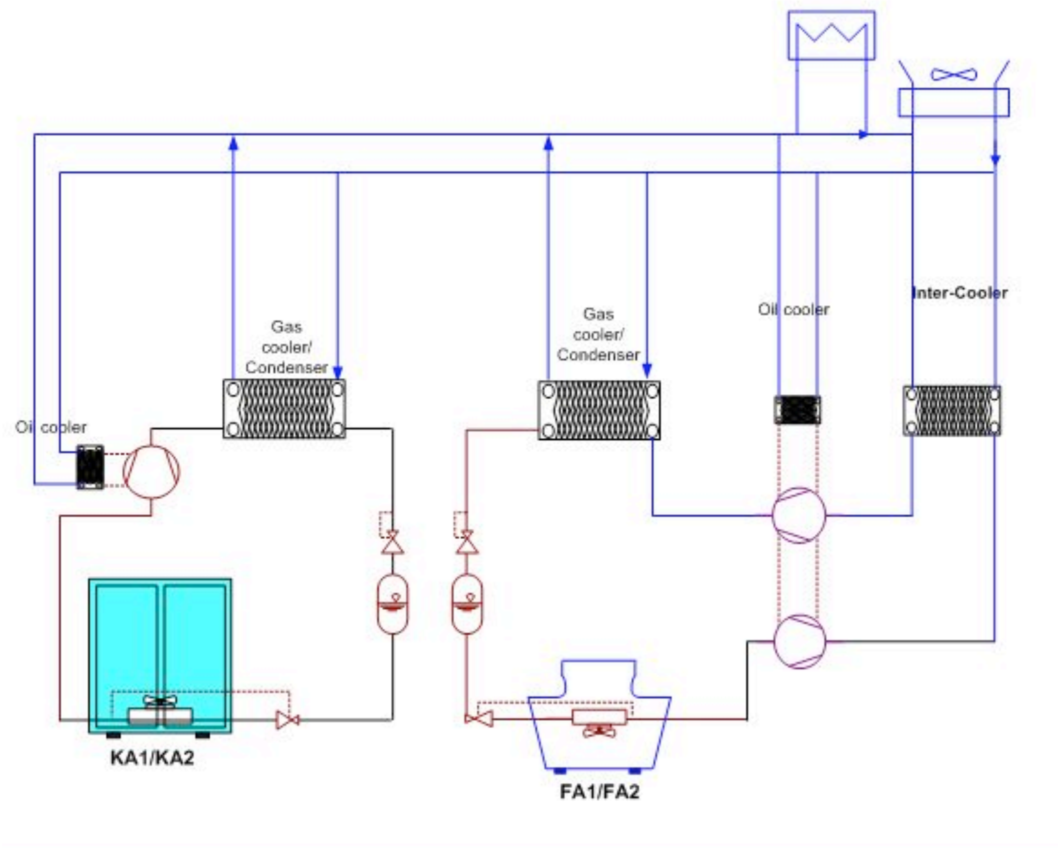


Figure 5.1 - A Schematic diagram of trans-critical system 1.

The system is a parallel solution where there are four separate carbon dioxide circuits, two for the medium temperature side (KA1 and KA2) and two for the low temperature side (FA1 and FA2). A benefit from using a parallel solution is that if one of the cycles fail, the other cycles can unaffectedly continue to work (Sawalha, 2008).

The cold temperature side, seen to the right in figure 5.1, has a two-stage compression with an intercooler in between. This is arranged to achieve cold temperatures but still keep low pressure ratios in the compressors. This will lower the inlet temperature to the second compressor, decrease the discharge pressure after the second stage and decrease the losses. The carbon dioxide is condensed in the condenser before it enters the expansion valve and the evaporator (freezers). The expansion valves are placed out in the supermarket close to the evaporators. After the freezer the refrigerant return to the machine room and enters a liquid separator (LS) before the compressors. This is done to make sure that no liquid enters the compressors. There are two units for the cold temperature side (FA1 and FA2). Each unit has two two-stage compressors.

The medium temperature side has a one-stage compressor since it does not need to operate with as high pressure ratio as the cold temperature side to maintain the chillers. After the condenser the refrigerant is expanded in the expansion valve where the pressure is reduced, before it enters the evaporator (chillers). As in the FA-units, the expansion valves are placed in the Supermarket area close to the cabinets. There are two units for the medium temperature side (KA1 and KA2). There are four one-stage compressors in every unit.

The refrigerant in both cycles is cooled by a coolant that circulates between the dry cooler and the four CO_2-cycles. The coolant is used for the oil coolers and the condensers/gas coolers in all units and for the intercooler in FA1 and FA2 units, see figure 5.1. The dry cooler is placed on the roof and uses the outside air to cool down the coolant. There is an additional heat exchanger in the coolant circuit, placed before the dry cooler, for supplying a heat pump that is servicing the supermarket with air conditioning and heating.

The compressors in this system are semi-hermetic compressors from Dorin with a swept volume 12,6 m^3/h at 2900 rpm (Dorin, 2008). The compressors are of one stage in the medium temperature side and of two-stage in the low temperature side. They are not frequency controlled, hence going on full speed when they are operating. The different compressor models can be seen in figure 5.2 and figure 5.3.

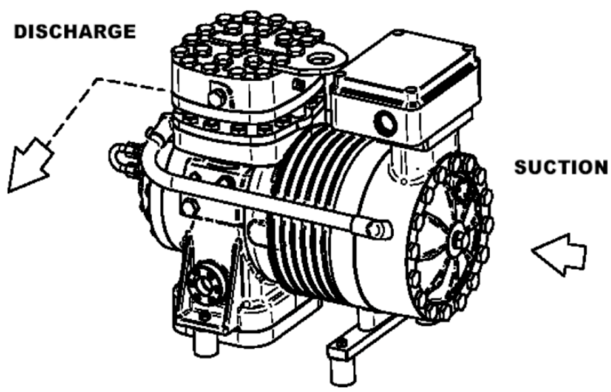


Figure 5.2 - A single stage semi-hermetic compressor from Dorin, model TCS373-D, used in KA-units (Dorin, 2009)

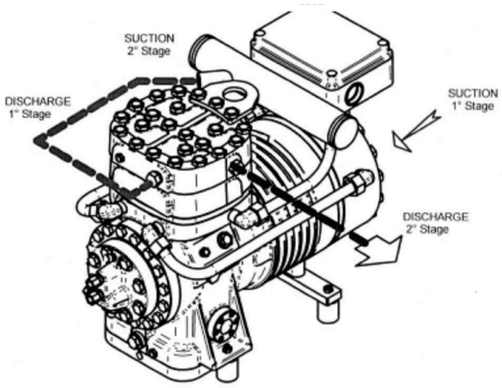


Figure 5.3 - A double stage semi-hermetic compressor from Dorin, model TCDH372B-D, used in FA-units (Dorin, 2009)

5.1.1 Evaluation method and calculations

Temperatures and pressure are measured in the essential points in the systems. The power to the compressors is also measured. By knowing these data it is possible to calculate the capacity and COP of the system.

At first the refrigerant mass flow rate in the system is calculated by using data for the compressor, see equation 5.1.

$$\dot{m}_{CO_2} = \frac{\eta_s \cdot \dot{V}_s}{\upsilon_{2k}} \quad [kg/s] \tag{5.1}$$

The swept volume, $\dot{V}_s$, is given by the manufacturer. By knowing the pressure and temperature at the compressor inlet, the specific volume υ_{2k} could be estimated. The volumetric efficiency η_s was calculated from compressor data. It is curve fitted and dependent on the pressure ratio in the system, see appendix 1. The volumetric efficiency for the medium temperature side compressors.

$$\eta_{s\_KA} = -0{,}4079 \cdot \left(\frac{P_{high}}{P_{low}}\right)^2 - 6{,}5843 \cdot \left(\frac{P_{high}}{P_{low}}\right) + 102{,}42 \quad [\%] \tag{5.2}$$

The volumetric efficiency for the low temperature side compressors:

$$\eta_{s\_FA} = 0{,}0251 \cdot \left(\frac{P_{high}}{P_{low}}\right)^2 - 1{,}1706 \cdot \left(\frac{P_{high}}{P_{low}}\right) + 93{,}424 \quad [\%] \tag{5.3}$$

where P_{high} is the discharge pressure after the compressor in KA-units and after the second compressor in FA-units. P_{low} is the pressure before the compressors.

After the mass flow is known it is possible to calculate the cooling capacity and COP of the system by equation 5.4 and 5.5. The enthalpies are calculated by knowing the temperatures and pressures in those specific points.

$$\dot{Q} = \dot{m}_{CO_2} \cdot \Delta h_{evap} \quad [kW] \tag{5.4}$$

where Δh_{evap} is the enthalpy difference over the evaporator (chillers/freezers).

$$COP = \frac{\dot{Q}}{\dot{E}_{comp}} \quad [-] \tag{5.5}$$

where $\dot{E}_{comp}$ is the total power consumption of the compressors.

To be able to estimate the temperature at the evaporator outlet, the outlet temperature from cabinets where used to make an average outlet temperature. This average was done for every five minutes.

The COP is calculated separately for every unit.

5.2 Trans-critical system 2

The system is a trans-critical carbon dioxide system with DX on both medium and low temperature levels. Figure 5.4 shows a schematic diagram of the refrigeration system.

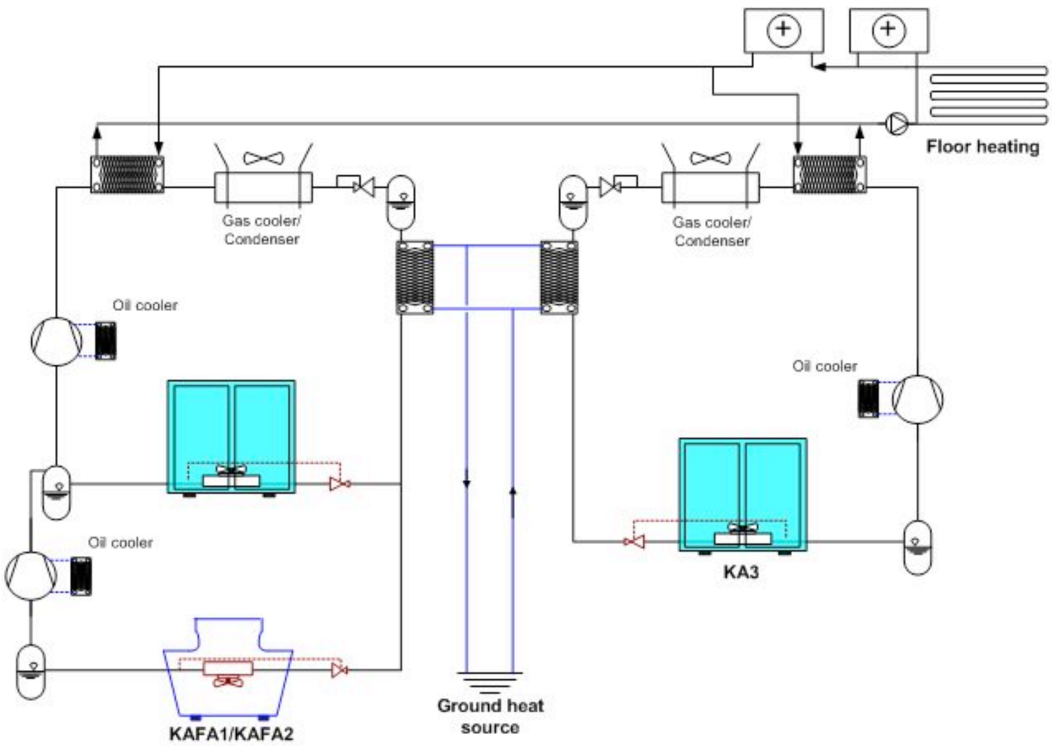


Figure 5.4 - A Schematic diagram of trans-critical system 2.

In this supermarket there are one circuit for the medium temperature side (KA3) and two circuits for a combined medium and cold temperature levels (KAFA1 and KAFA2).

The KA3 cycle can be seen to the right in figure 5.4 and is similar to the KA-unit in trans-critical system 1. After the evaporator (chillers) the refrigerant enters the one-stage compressor. An extra heat exchanger is placed after the compressor to recover heat to floor and space heating of the supermarket. The refrigerant is after that gas cooled/condensed in the gas cooler. The gas cooler is placed on the roof and rejects heat to the ambient. Before the refrigerant reaches the expansion valve an extra heat exchanger is placed to further cool down the refrigerant and gain some additional sub-cooling. This heat exchanger uses a ground heat sink for heat exchange with the carbon dioxide.

The combined circuit (KAFA1/KAFA2) side can be seen to the left in figure 5.4 and it serves both medium temperature cabinets and freezers. It is similar to KA3-unit but the mass flow of the refrigerant is separated before it reaches the expansion valves and cabinets/freezers. After the freezers two compressors called “booster compressors” are located. They increase the pressure of CO_2 to the same pressure as the CO_2 has during the evaporation in the medium temperature cabinets. The mass flows from the medium temperature cabinets and from the freezers are mixed in the liquid separator before the high stage compressors. The high stage compressors raise the pressure of the CO_2 to condensing pressure.

The compressors are of semi-hermetic type from Dorin. The compressor in the KA3-unit and in the high stage of the booster system is of the same type as KA-unit in trans-critical system 1, se figure 5.2. It has a swept volume 12,6 m^3/h at 2900 rpm. The booster compressors have the same external appearance as the model in figure 5.2 but it is of model SCS362-D and have a swept volume of 10,7 m^3/h at 2900 rpm. There is no capacity control of the compressors. The compressors have external oil coolers.

5.2.1 Evaluation method and calculations

As in the previous system the COP and capacity will be calculated. The calculations of KA3-unit are basically done in the same way as in trans-critical system 1. The refrigerant mass flow rate is calculated using equation 5.1. The compressors operating in the KA3-unit is of the same type as in KA-unit in trans-critical system 1, so equation 5.2 was used.

By knowing the, electric power consumption of the compressor and the mass flow rate the cooling capacity and COP of KA3-unit can be calculated in the same way as in trans-critical system 1, see equations 5.4 and 5.5.

In the booster system the mass flow is, as in the other cases, calculated using equation 5.1.

Since the high stage compressors and the booster compressors are located in different places in the system it is possible to calculate two mass flows. One mass flow is the total mass flow going through the high stage compressors and the other mass flow is what flows in the freezers. A mass balance can be applied to calculate the mass flow going through the medium temperature cabinets, see equation 5.6.

$$\dot{m}_{CO_2,medium} = \dot{m}_{CO_2,total} - \dot{m}_{CO_2,freezer} \quad [kg/s] \tag{5.6}$$

The compressor for the high stage is of the same type as in KA3 with the same volumetric efficiency correlation, equation 5.2, while for the freezer compressors it is given by the following equation, see appendix 1:

$$\eta_{s\_booster} = -0{,}1139 \cdot \left(\frac{P_{high}}{P_{low}}\right)^2 - 4{,}1854 \cdot \left(\frac{P_{high}}{P_{low}}\right) + 95{,}12 \quad [\%] \tag{5.7}$$

The separated mass flows provide the cooling capacity for the freezers and the medium temperature cabinets according to equation 5.4.

To calculate the COP for the medium temperature side of booster system, the total load, $\dot{Q}_{high}$, for the high stage compressor need to be identified by equation 5.8.

$$\dot{Q}_{high} = \dot{Q}_{medium} + \dot{Q}_{freezers} + \dot{E}_{freezer,shaft} \quad [kW] \tag{5.8}$$

$\dot{Q}_{medium}$ is the load for the chillers, $\dot{Q}_{freezers}$ is the load for the freezers and $\dot{E}_{freezer,shaft}$ is the power input to the refrigerant during compression. The COP of the high stage compressors can then be calculated by equation 5.9.

$$COP_{high} = \frac{\dot{Q}_{high}}{\dot{E}_{high}} \quad [-] \tag{5.9}$$

The power that goes to the high stage compressor, $\dot{E}_{high}$, in the booster system is contributing to the cooling capacity and COP of both the medium temperature side and the cold temperature side. Therefore the energy consumption of the high stage compressors needs to be separated before the COP for the cold temperature side can be calculated. The energy that goes to the cooling of the medium temperature cabinets from the high stage compressor can be calculated by equation 5.10.

$$\dot{E}_{high,medium} = \frac{\dot{Q}_{medium}}{COP_{high}} \quad [kW] \tag{5.10}$$

The energy that goes to the low temperature level from the high stage compressors are calculated by equation 5.11.

$$\dot{E}_{high,freezers} = \dot{E}_{high} - \dot{E}_{high,medium} \quad [kW] \tag{5.11}$$

When the energy from the high stage compressors to the freezers is identified the COP for the low temperature side can be calculated by 5.12.

$$COP_{booster} = \frac{\dot{Q}_{freezers}}{\dot{E}_{freezers} + \dot{E}_{high,freezers}} \quad [-] \tag{5.12}$$

Finally, a total COP for the KAFA-unit can be calculated with equation 5.13, by knowing the cooling capacity for low and medium temperature side and the total compressors electric power consumption.

$$COP_{total} = \frac{\dot{Q}_{medium} + \dot{Q}_{freezers}}{\dot{E}_{high} + \dot{E}_{freezers}} \quad [-] \tag{5.13}$$

Some assumptions have been made for KAFA-units to be able to perform these calculations. Before January there was only data available of the total energy that goes to the KAFA-unit and no separate measurement of the energy that goes to the booster compressors was done. From January the measurement of the energy to the high stage and booster compressors are separated for KAFA1. Based on that information an average of the energy that goes to the booster compressors of the total power consumption of the compressors was estimated. This was used to perform calculations of COP and cooling capacity for the months prior to the separate energy measurements on KAFA1 and for all the month for KAFA2.

5.2.2 Oil cooler capacity

A major challenge in this thesis has been to estimate the losses from the compressor to the oil cooler. It is necessary to be aware of these losses when applying a energy balance over the compressor to find out the mass flow rate of refrigerant. During this thesis it has been shown that a small change in oil cooler capacity has a significant impact on the COP of the system. Therefore it is of importance that oil cooler capacity is correctly estimated. Figure 5.5 shows the difference in COP for one of the systems when the oil cooler capacity is assumed to 15% and 20% respectively. The difference between COP is about 7%.

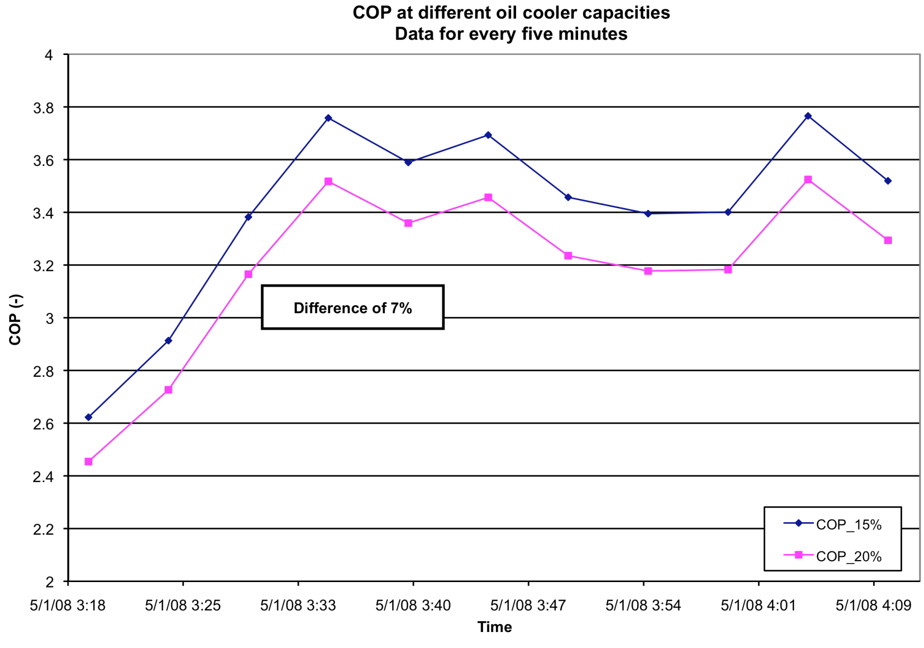


Figure 5.5 - COP at different oil cooler capacities.

Three different approaches have been discussed when estimating the oil cooler capacity in this thesis.

5.2.2.1 Compressor manufacturer

The manufacturer has estimated the oil cooler capacity to be about 20% of the compressor power (Dorin, 2009). This estimation is based on tests the manufacturer have done. It is good to have this information as a guideline when analysis of the oil cooler is done but data from every supermarket is needed to verify that 20% is a good estimation.

5.2.2.2 Oil cooler capacity by temperature measurements

One way to calculate the oil cooler capacity is to measure the inlet and outlet temperature of the oil. By knowing the type of oil and the flow rate, the capacity can be calculated by equation 5.14.

$$\dot{Q}_{oilcooler} = \dot{V} \cdot \rho \cdot c_p \cdot \Delta T \qquad (5.14)$$

The oil maintaining the compressors in trans-critical system 2 is BREOX RL 68 P with a density of 0,998 kg/m^3 at 20°C and volume flow of 0,1 l/s, given from the compressor manufacturer. The oil in the oil coolers is Breox RFL68 EP. The c_p of the oil can be estimated as an average from ISO46 and ISO100, see table 5.2.1. It is estimated to 1,6254 MJ/m^{3}K.

BREOX RL	ISO 46	ISO 68	ISO 100
C_p (MJ/m^{3}K)	1,6707	≈1,6254	1,5801

Table 5.2.1 - The c_p for BREOX RL 68.

The temperatures of the oil inlet and outlet for different compressors have been measured for trans-critical system 2. The results from the measurements were that the temperature difference of the oil was relatively constant but different for every compressor. Due to this no general assumption of the oil cooler could be made. Figure 5.6 shows the temperature difference and oil cooler capacity for two different compressors. Oil temperature at the inlet to the oil cooler is denoted T_comp_in and exit temperature is T_comp_out. Between 11:54 and 12:24 the measurements were made for one high stage compressor in KAFA2. It shows a temperature difference of about 15°C and a calculated oil cooler capacity of 2 kW. After that period the measurement was changed to another high stage compressor of the same type and in the same unit and it can be seen in the figure that the temperature difference is changed. The difference is about 5 °C and the calculated oil cooler capacity is below 1 kW. The compressors were kept in operating mode during the measurement period.

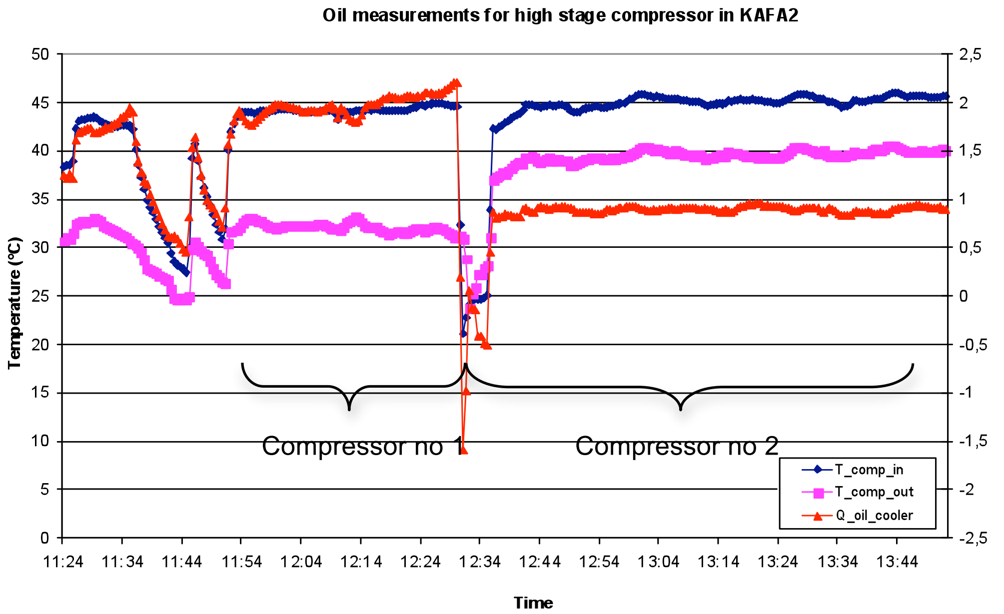


Figure 5.6 - Temperature and calculated oil cooler capacity for two different compressors during a two-hour test

Figure 5.7 shows the temperature difference for a compressor located in the same position as the compressors in figure 5.6 but in another unit. The difference in temperature is slightly increased and therefore there is also an increase in the oil cooler capacity during this period. The temperature measurements were done for other compressors in the system and all the measurements resulted in different temperature difference and capacities. The measurements for all the compressors can be found in appendix 2.

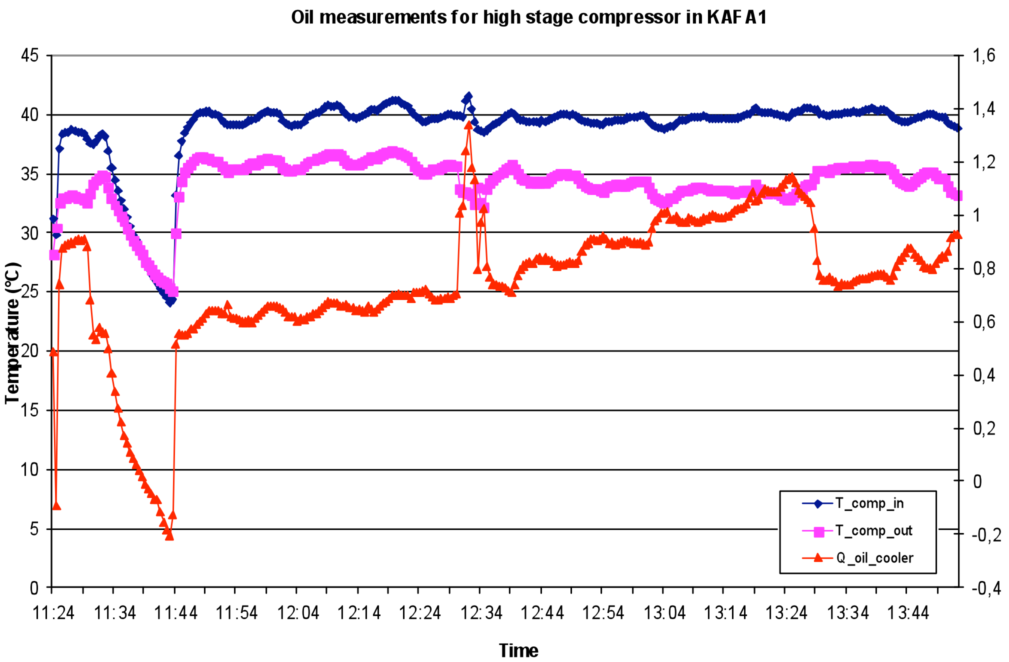


Figure 5.7 - Temperature and calculated oil cooler capacity for one compressor during a two-hour test

There are several factors that have an impact on the oil cooler capacity e.g. if the mass flow is different from the given value from the compressor manufacturer the result will be incorrect. During the operation some of the refrigerant dissolves in the oil changing the properties and makes it impossible to calculate with equation 5.18 and the properties data stated for pure oil. This could be one of the reasons why the capacity is slightly increasing in figure 5.7. Due to these uncertainties the general conclusion is that the capacity should not be estimated from the oil side measurements in these supermarkets.

5.2.2.3 Oil cooler capacity by compressor data

The oil cooler capacity is necessary to estimate if a heat balance is applied over the compressor to calculate the mass flow. To solve this problem, the mass flow in this thesis was instead, as mentioned earlier, calculated by the compressor data for all systems, see equation 5.1.

By knowing the mass flow of the refrigerant it is possible to calculate the cooling capacity and the COP of the system without estimating the oil cooler capacity. When the mass flow is known it is interesting to verify the oil cooler capacity with a heat balance over the compressors, see equation 5.15-5.17.

$$\dot{Q}_{oilcooler} = \dot{E}_{el} - \dot{E}_{shaft} - \dot{Q}_{losses} \quad [kW] \tag{5.15}$$

where $\dot{E}_{shaft} = \dot{m}_{oil} \cdot (h_{comp,out} - h_{comp,in})$ (5.16)

$\dot{Q}_{losses} = 7\% \cdot \dot{E}_{el}$ (5.17)

$\dot{E}_{el}$ is the electrical energy consumption in kW

These equations were used to verify the oil cooler capacity. The graph in figure 5.8 shows the oil cooler as a percentage of E_{el} for January in KA-unit in trans-critical system 1 and the graph in figure 5.9 show the percentage for the same unit but in July. The calculations are done for every five minutes during the month and the black line is an average.

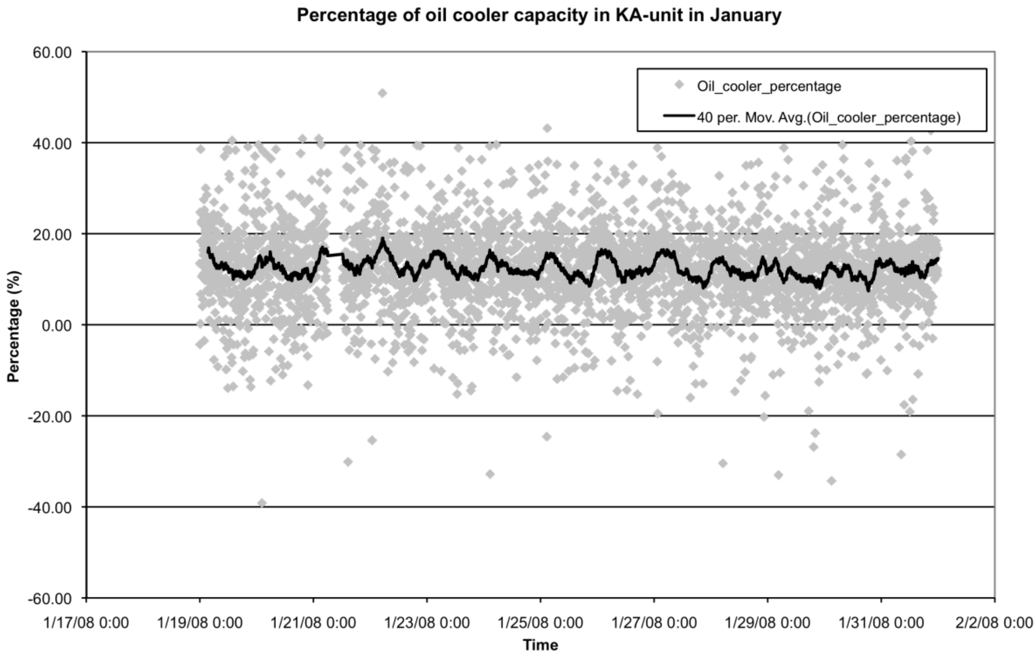


Figure 5.8 - The oil cooler as a percentage of E_{el} for KA-unit in TR1 in January.

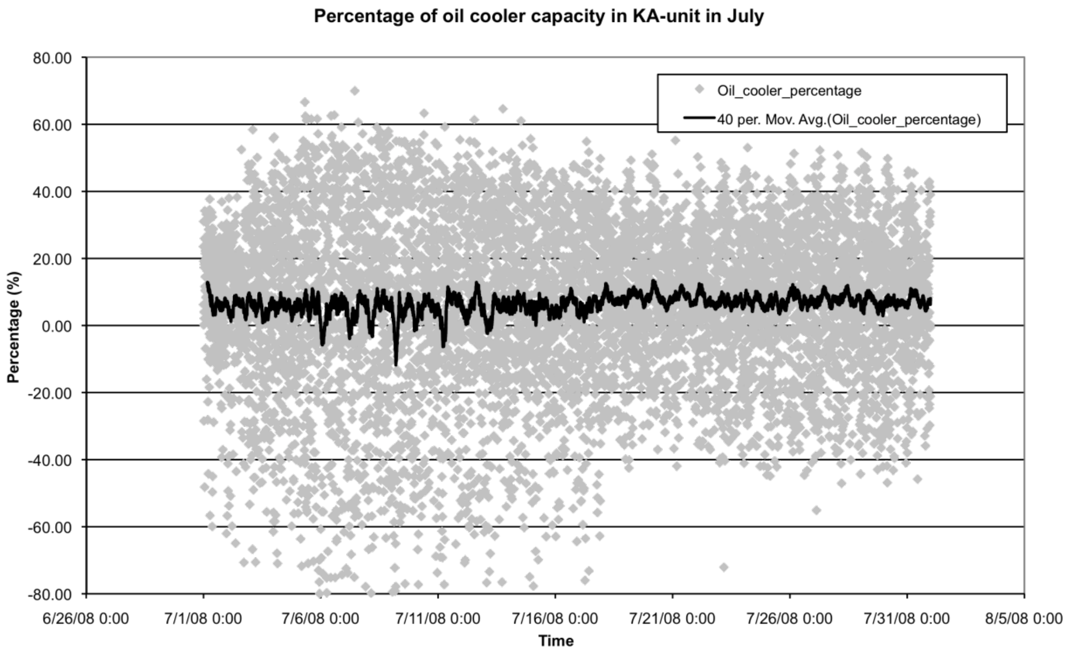


Figure 5.9 - The oil cooler as a percentage of E_{el} for KA-unit TR1 in July.

What can be seen from the figures is that the percentage varies widely and is sometimes negative which should not normally be the case. The reason for this could be because the frequent starts and stops in operation of the compressors. The percentage is more spread for July than January and this could be because there are more compressors that cut in and out of operations more frequently than in the winter case. These figures also show that the oil cooler capacity is higher in the winter than in the summer. This could be because of that the surrounding temperature is higher in the summer and therefore are not able to cool down the oil as much as in the wintertime.

5.3 Cascade system 1

The system solution is a cascade solution with R404A in the high stage and CO_2 in the low stage. There is direct expansion with CO_2 for the freezers and indirect with a brine for the medium temperature side. Figure 5.10 shows a schematic diagram of the system. There are two units of the low stage (KS5 and KS6) and two units for the high stage (VKA1 and VKA2). There is only one brine circuit.

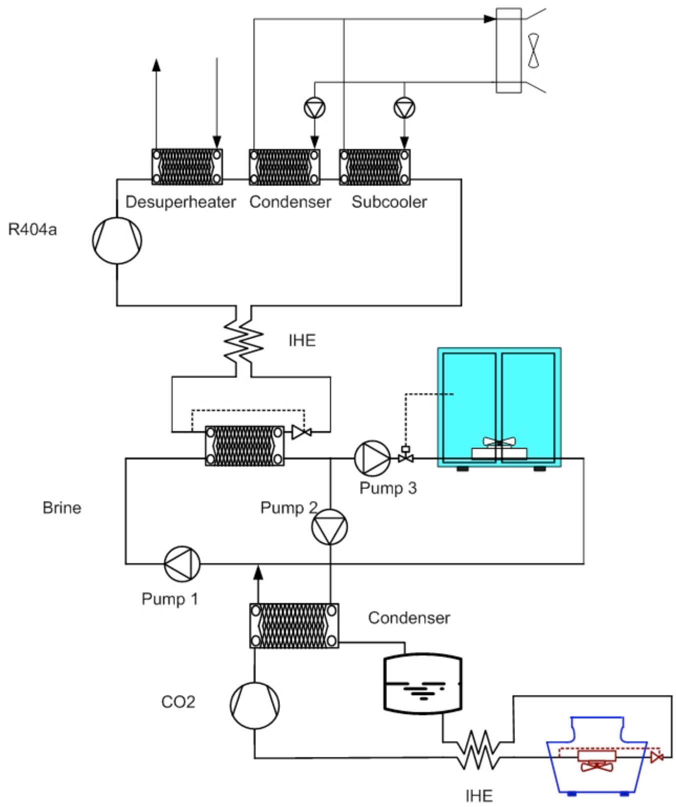


Figure 5.10 - A Schematic diagram of cascade system 1.

R404A in the high stage is condensed by a coolant that rejects heat to the ambient. There is a a desuperheater located before the condenser to recover some of the heat after the compressor, and a subcooler that uses the coolant. There is an internal heat exchanger (IHE) in the system where the refrigerant is further subcooled by transfering heat to the refrigerant after the evaporator. After the evaporator and the IHE the refrigerant enters the compressors before returning to the desuperheater. There are two units of the high stage R404A and three compressors in every unit.

The brine evaporating the R404A is cooling the medium temperature cabinets and is circulated by pumps. The brine condenses the CO_2 in the low stage. After the condenser the CO_2 flows through an IHE to be subcooled before the expansion valve and the freezers. After the freezers the refrigerant enters the

IHE before it enters the compressors and then back to the condenser. There are two units of the low stage CO_2 and four compressors in every unit.

The compressors are from Bitzer. The R404A-compressors has a swept volume of 73,6 m^3/h at 1450 rpm. It is of model 4H-15.2Y-40P. The CO_2 compressors has a swept volume of 4,06 m^3/h at 1450 rpm. It is of model 2KC-3.2K-40S.

5.3.1 Evaluation method and calculations

The capacity and COP will be calculated in this system as in the previous systems but this system is a bit different since it is of cascade solution.

In this supermarket the measured temperatures and pressures are saved in an RDM. They are logged every 15 minutes. There is no energy measurement in this supermarket instead the temperature after every compressor is measured to indicate if the compressor is running. The assumption, that the compressors are operating when the outlet temperature is above 50°C, was made. When the compressors are assumed to be operating a mass flow runs through.

The mass flow in the system is calculated by compressor data in the same way as for the previously studied supermarkets, see equation 5.1. This is done both for the R404A-unit and for the CO_2-unit. The density is calculated at the inlet of the compressor from measured pressure and temperature. The swept volume and the volumetric efficiency can be estimated by the compressor data, see appendix 3.

By using the mass flow, the energy consumption of every compressor can be calculated. The total energy consumption is then obtained by adding the consumption of all compressors.

$$\dot{E}_{shaft} = \dot{m} \cdot \Delta h_{comp} \quad [kW] \tag{5.18}$$

$$\dot{E}_{shaft,total} = \dot{E}_{shaft,comp1} + \dot{E}_{shaft,comp2} + \dot{E}_{shaft,comp3} \ldots . etc \ [kW] \tag{5.19}$$

To get the total electric energy going in to the compressors a curve fit is done from the compressor data. The electric power consumption during different operating conditions is plotted vs the pressure ratio for the compressor. The total electric power consumption of the CO2 and R404A compressor is calculated using the correlations in equations 5.20 and 5.21. For the calculations see appendix 4.

$$\dot{E}_{el,CO_2} = 0{,}263 \cdot \left(\frac{P_{high}}{P_{low}} \right) + 1{,}2111 \quad [kW] \tag{5.20}$$

$$\dot{E}_{el,R404A} = -0{,}4715 \cdot \left(\frac{P_{high}}{P_{low}} \right) + 16{,}394 \quad [kW] \tag{5.21}$$

With mass flow, temperatures and pressure it is possible to calculate the cooling capacity of the R404A- and CO_2-units, see equation 5.4.

The condenser load of the CO_2-unit can be calculated with equation 5.22.

$$\dot{Q}_{cond,CO_2} = \dot{Q}_{evap,CO_2} + \dot{E}_{CO2,shaft} \quad [kW] \tag{5.22}$$

Three pair of pumps is circulating the brine. In every pumping pair there is always only one pump running at a time. The pumping pairs are shown in figure 5.10 and are called pump 1, pump 2 and pump 3. The pumping power for every pumping pair is shown in table 5.3.1 and it is estimated by pump data. The pumps are of fixed speed.

	Pump 1	Pump 2	Pump 3
Power (kW)	2,2	1,5	3

Table 5.3.1 - The pumping power for every pump pair.

The total power of the pumps is calculated by equation 5.23 and is 6,7 kW.

$$\dot{E}_{pumps,total} = \dot{E}_{pump\_1} + \dot{E}_{pump\_2} + \dot{E}_{pump\_3} \quad [kW] \tag{5.23}$$

To decide the load of the medium temperature cabinets equation 5.24 is used.

$$\dot{Q}_{cab} = \dot{Q}_{evap,R404A} - \dot{Q}_{cond,CO_2} - \dot{E}_{pumps,total} \quad [kW] \tag{5.24}$$

The COP for the medium temperature cabinets is calculated by

$$COP_{cab} = \frac{\dot{Q}_{evap,R404A}}{\dot{E}_{R404A,total} + \dot{E}_{pump\_1} + \dot{E}_{pump\_3}} \quad [-] \tag{5.25}$$

The energy from the high stage compressors that goes to the cabinets is estimated by

$$\dot{E}_{R404A,cab} = \frac{\dot{Q}_{cab}}{COP_{cab}} \quad [kW] \tag{5.26}$$

The energy from the R404A compressors that goes to the freezer is estimated by

$$\dot{E}_{R404A,freezers} = \dot{E}_{R404A,total} - \dot{E}_{R404A,cab} \quad [kW] \tag{5.27}$$

The COP for the freezers can be calculated by equation 5.28.

$$COP_{freezers} = \frac{\dot{Q}_{CO_2}}{\dot{E}_{el,CO_2} + \dot{E}_{R404A,freezers} + \dot{E}_{pump\_2}} \quad [-] \tag{5.28}$$

The total COP of the system is estimated by equation 5.29:

$$COP_{total} = \frac{\dot{Q}_{CO_2} + \dot{Q}_{cab}}{\dot{E}_{el,CO_2} + \dot{E}_{el,R404A} + \dot{E}_{pumps,total}} \quad [-] \qquad (5.29)$$

6 Results

In this section the result for all supermarkets will be presented as an average per month. There will also be "close-ups" for one summer period and one winter period for those systems where this data is available, to see how the operation of the system is changing during opening-hours and night time.

6.1 Trans-critical system 1

The supermarket has been investigated from January 2008 to February 2009.

6.1.1 Power consumption

In figure 6.1 the power of the compressors in the four units FA1, FA2, KA1 and KA2 are drawn. It is a monthly average.

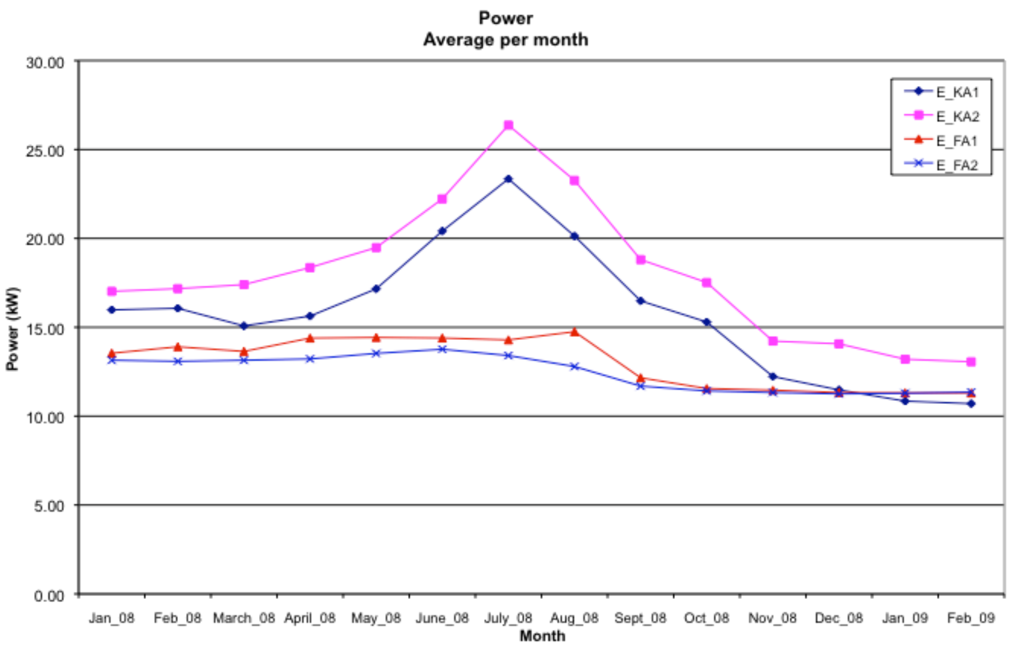


Figure 6.1 - Monthly average of the compressor power for the four units in TR1.

The result shows that the outside temperature affect the power consumption and that is has the largest influence on the medium temperature side. The energy that goes to the FA-compressors is slightly increased during the summer but not as much as the power for the KA-unit and this is due to the fact that there are doors on many of the freezers that retain the cold air. The power of the KA-compressors is varying by using about 10 kW more in summer than in wintertime. This is because the increased need of cooling load during the summer due to the higher humidity rate in the supermarket.

Figure 6.2 shows how the total power consumption varies during 7 days for the compressors in the KA- and FA-units in trans-critical system 1 in January 2008.

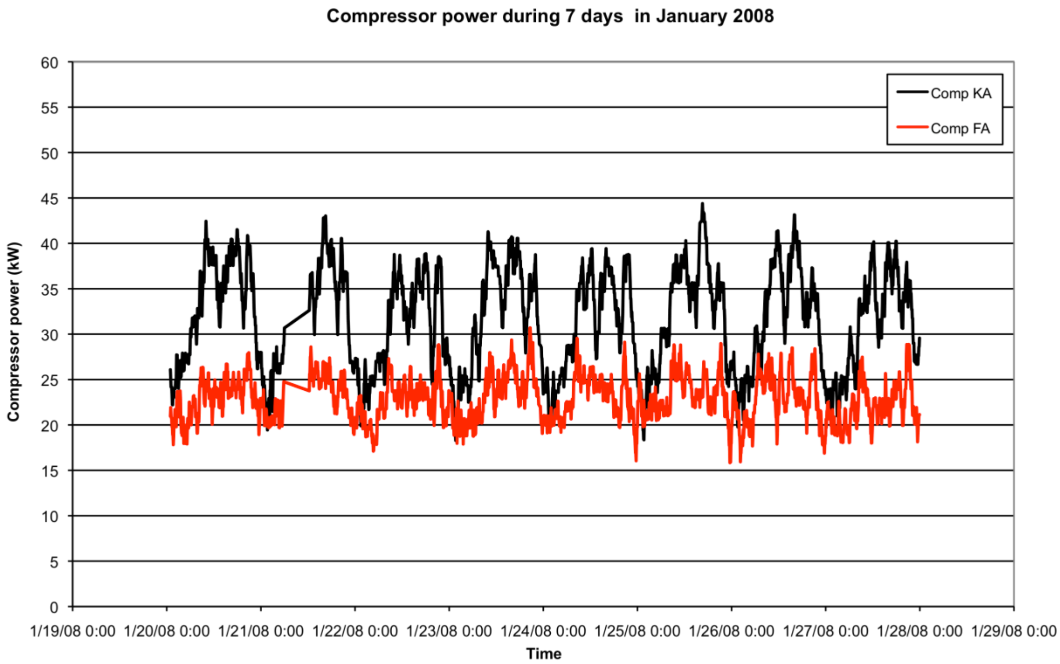


Figure 6.2 - The total compressor power during 7 days in January in TR1

The power consumption is increased during the days when the supermarket is open. It can also be seen that the power consumption is higher for KA-units in the days than for FA-units. The difference between KA-units and FA-units are decreased during night period. The reason for this is because the medium temperature cabinets are covered with curtains during the nights.

Figure 6.3 shows how the total power consumption varies during 7 days for the compressors in the KA-units and FA-units in trans-critical system 1 in July 2008.

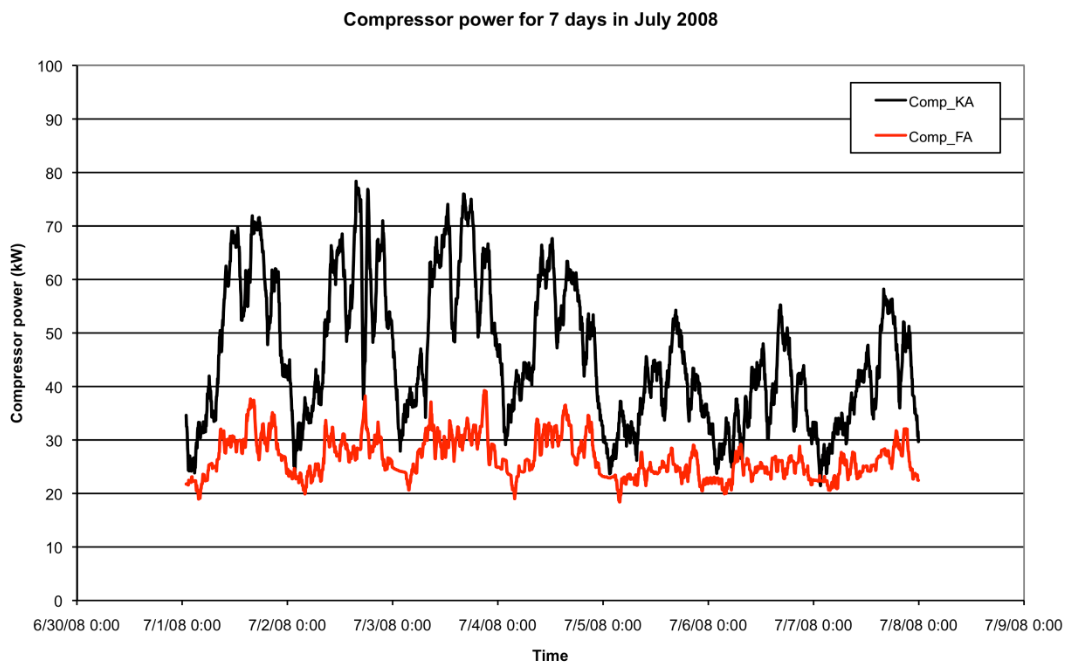


Figure 6.3 - The total compressor power during 7 days in July in TR1

The power consumption is still higher for the KA-units than FA-units. A comparison between the energy consumption in January and in July shows that the consumption is higher in the summer for both units, which is reasonable since the humidity in the air is higher during summer.

Figure 6.4 shows how the compressors in one KA-unit and one FA-unit are working during 24 hours in January.

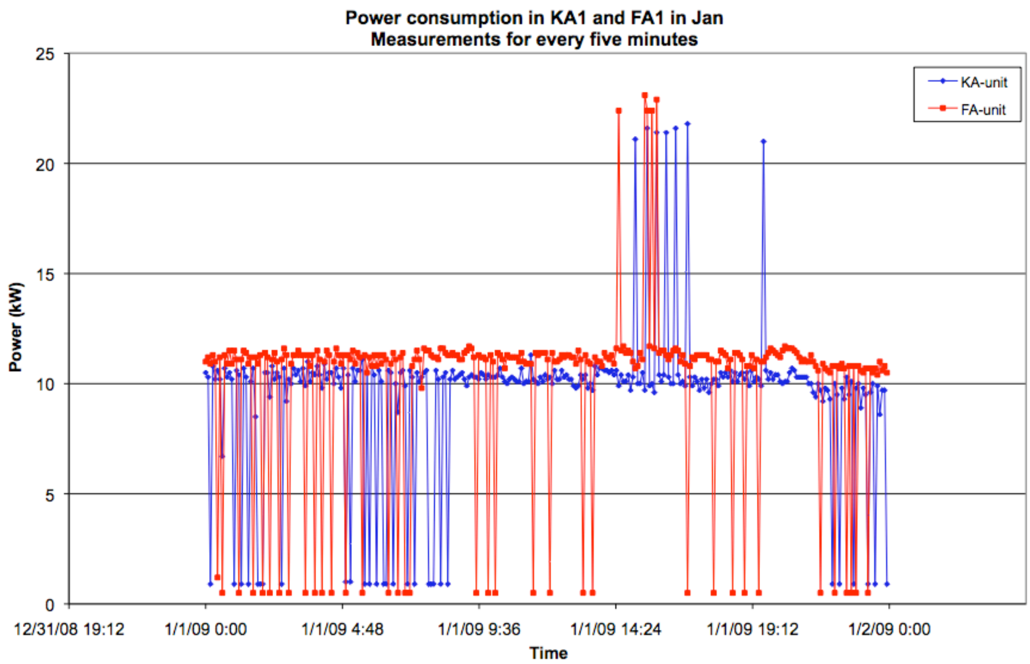


Figure 6.4 - The compressor power for KA1 and FA1 in January in TR1.

Most of the time there is only one compressor running in both units and the consumption is around 10 kW. During the night the power consumption is sometimes zero, which is due to the low need of cooling during the night but also since the cabinets are covered with curtains. In the afternoon sometimes two compressors are running at the same time to supply the higher cooling capacity that is needed.

Figure 6.5 show the power consumption of the compressors of the same units as in figure 6.4 but in July.

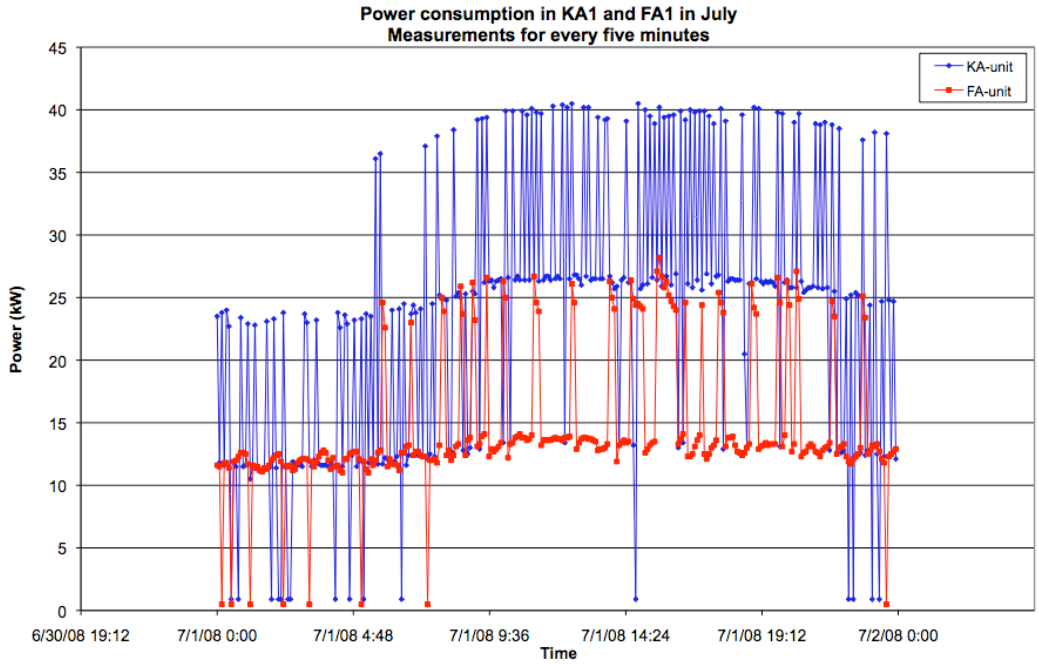


Figure 6.5 - The compressor power for KA1 and FA1 in July in TR1

In this figure it can easily be seen that the compressors in the KA-unit are working harder than the compressors in FA-unit. This was also reflected in figure 6.1 that showed the monthly average. For the KA-unit there are often one or two compressors running during the night and two or three compressors during the day when the supermarket is open. The FA-unit is operating with one compressor most of the day but a second compressor kicks in during the opening hours of the supermarket when necessary.

6.1.2 Cooling capacity and coefficient of performance

Figure 6.6 shows the cooling capacity and COP of the two FA-units for every month during one year.

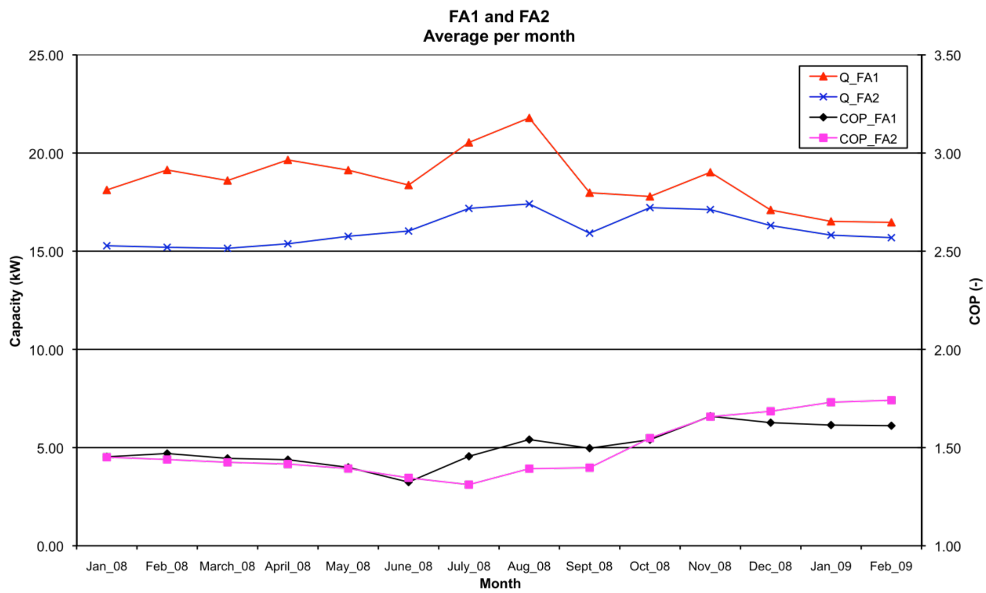


Figure 6.6 - Monthly average of the cooling capacity and COP for the freezing units.

The COP of both units is slightly lower in the summer than in the winter due to the higher load and the increased surrounding temperature and humidity. The COP is higher in January 2009 than in January 2008 and is varying between 1,4 and 1,75. This trend will be recognized in all the graphs for trans-critical system 1 and that is due to a change in operating conditions done in the supermarket, which resulted in lower condensers'/gas coolers' coolant temperature.

Figure 6.7 shows the cooling capacity and COP of the two KA-units for every month during the period.

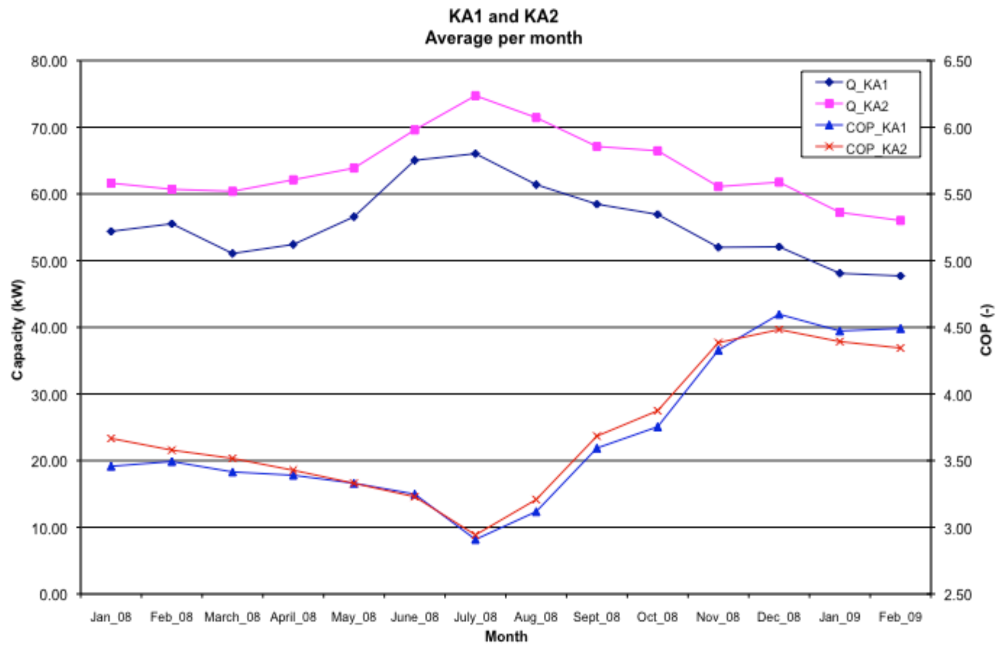


Figure 6.7 - Monthly average of the cooling capacity and COP for the medium temperature units.

The capacity is significantly higher for KA-units compared to the FA-units in figure 6.6 which is due to that there are more chillers than freezers and that there are doors on some of the freezers. The difference of the cooling capacity for the KA-units in winter and summer is about 15 kW. The COP is higher in the winters than in the summer and in this graph it is clearly shown that the COP is higher in winter 2009 than in winter 2008 and that is due to a change in operating conditions that lowered the condensing temperature. The COP is varying between 3 and 4,5 depending on the season.

Figures 6.8 and 6.9 show how the COP, cooling capacity and the power is changing during 24 hours for KA-unit in January and July.

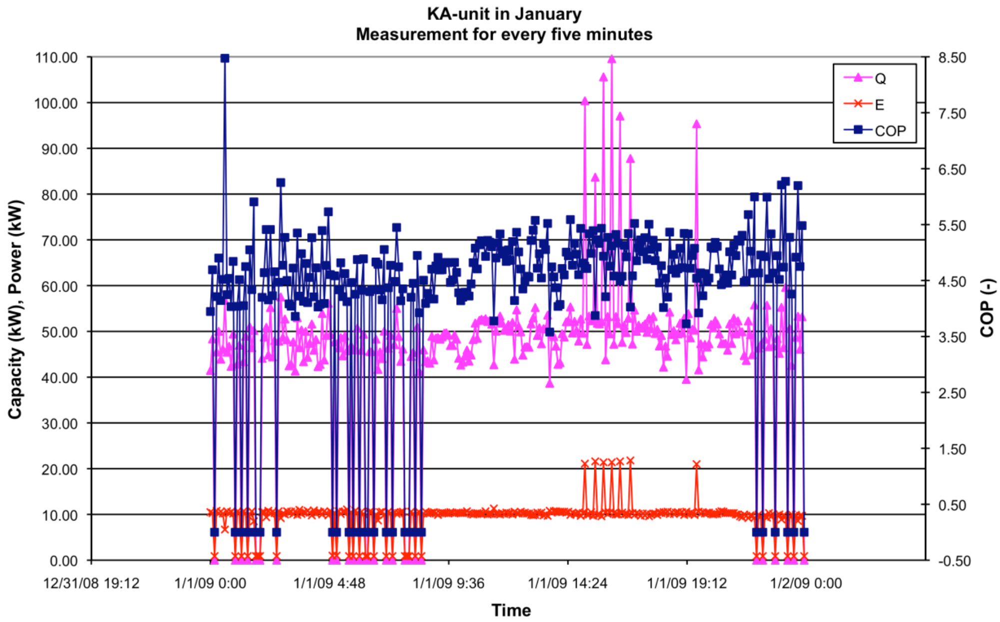


Figure 6.8 - The power, capacity and COP of KA-unit in January in TR1

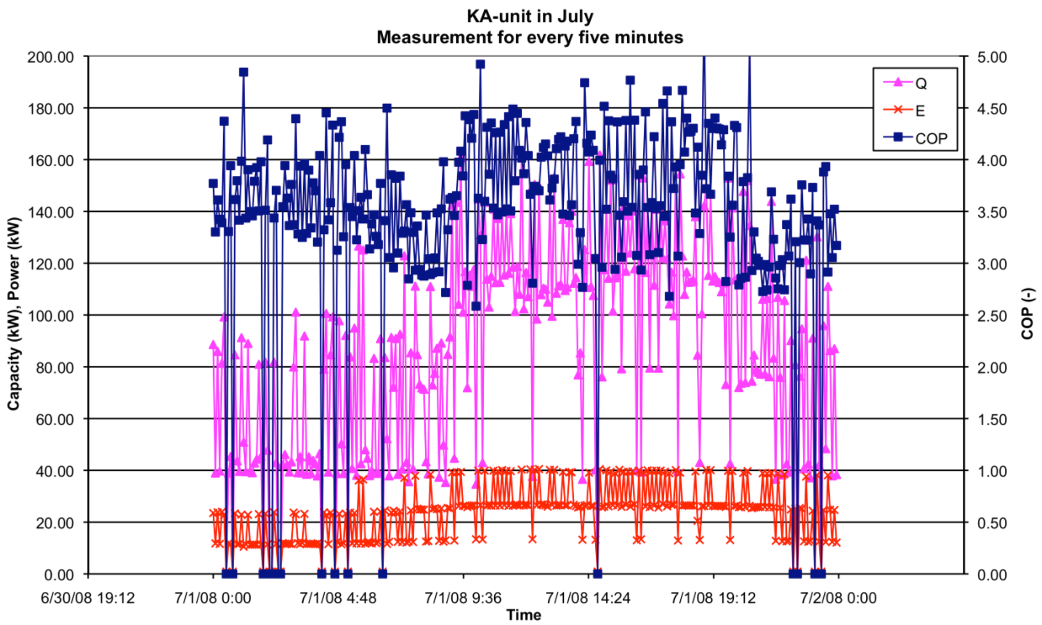


Figure 6.9 - The power, capacity and COP of KA-unit in July in TR1

It can be seen in figure 6.8 that the capacity is a bit higher at some occasions during the day and mostly one compressor is running except from these occasions where a second compressor is required to cut in. In January the variation of the values is fairly low compared to the summer case and this is due to the fact that more compressors are going in and out of operation during summer since the cooling load is higher. The COP is lower in the summer because of the higher condensing temperature. Figure 6.9 clearly shows that the cooling load is higher during summer and higher during the day than in the night. This also shows that it is of importance to use curtain during the nights to lower the cooling load and energy consumption.

Figures 6.10 and 6.11 show the COP, cooling capacity and the power for FA-unit in January and July.

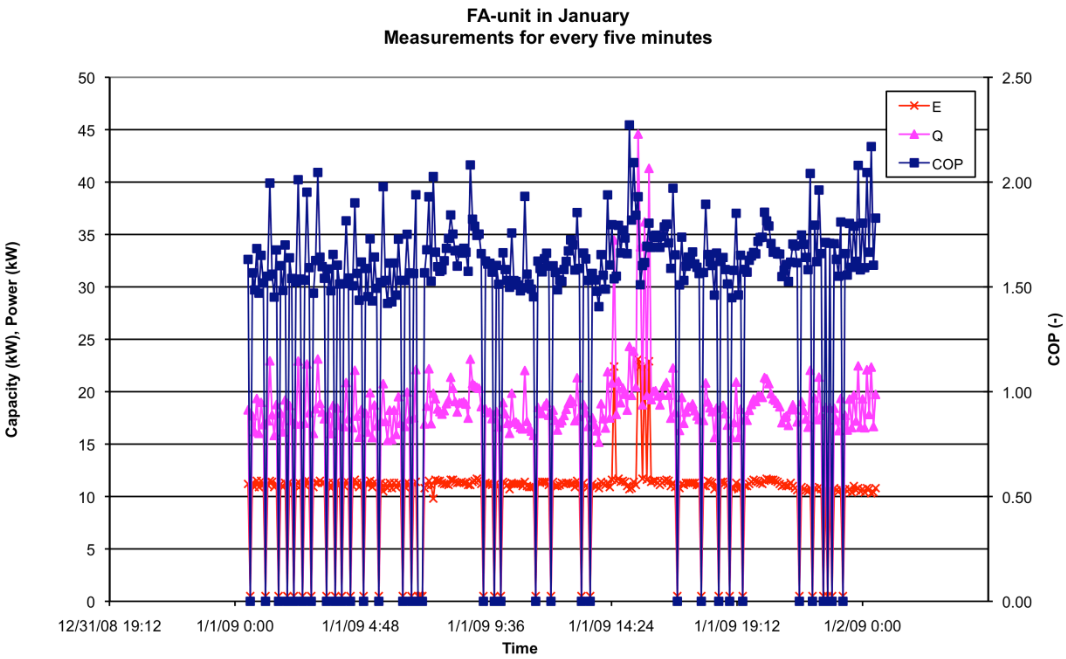


Figure 6.10 - The power, capacity and COP of FA-unit in January in TR1

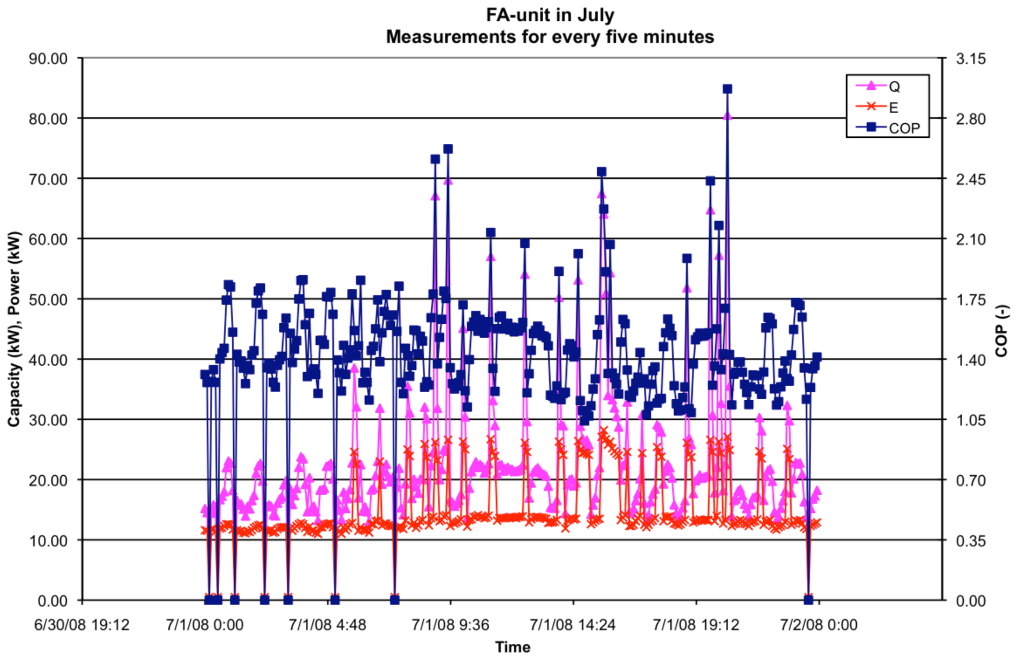


Figure 6.11 - The power, capacity and COP of FA-unit in July in TR1

In this case the values are not varying as much as in the previous case but there is still a larger spread in July than in January. The cooling load is more or less the same during the winter day and summer day and this is due to the doors that are used on many of the freezers.

6.1.3. Temperatures and pressures

Figure 6.12 is showing the monthly average of temperatures in trans-critical system 1.

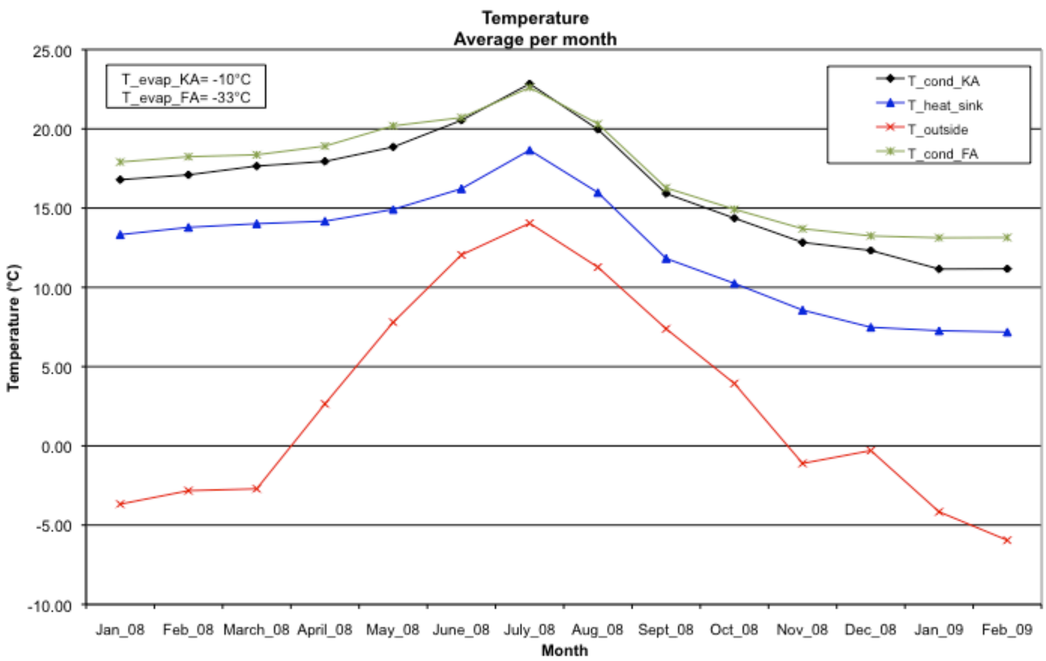


Figure 6.12 - Monthly average of temperatures in TR1.

The condensing temperature for both KA-units and both FA-units is varying between 10°C and 22°C during the period and is following the heat sink curve. The higher ambient temperature during summer forces the system to have higher condensing temperature, which increase the energy use. The evaporating temperature is -10°C for the medium temperature side and -33°C for the low temperature side and is rather constant during the year. What can be seen is that the condensing temperature is lower in winter 2009 than last year which is the reason for the increase in COP that has been seen in the previously figures. In the figure the heat sink temperature is close to the outside temperature in the summer but has a difference of about 20 °C in the winter of 2008 and of 10°C in winter 2009. This is due to a valve controlling the heat sink temperature. In the summer it is fully open and allows all the coolant to heat exchange with outside air before going to the CO_2 condenser/gas cooler. During the colder periods it is partly open, hence allowing some of the brine from the CO_2 condensers/gas coolers to reticulate and mix with the outside air cooled brine and condense/gas cool the CO_2 again. The controlling valve can be seen in figure 6.13.

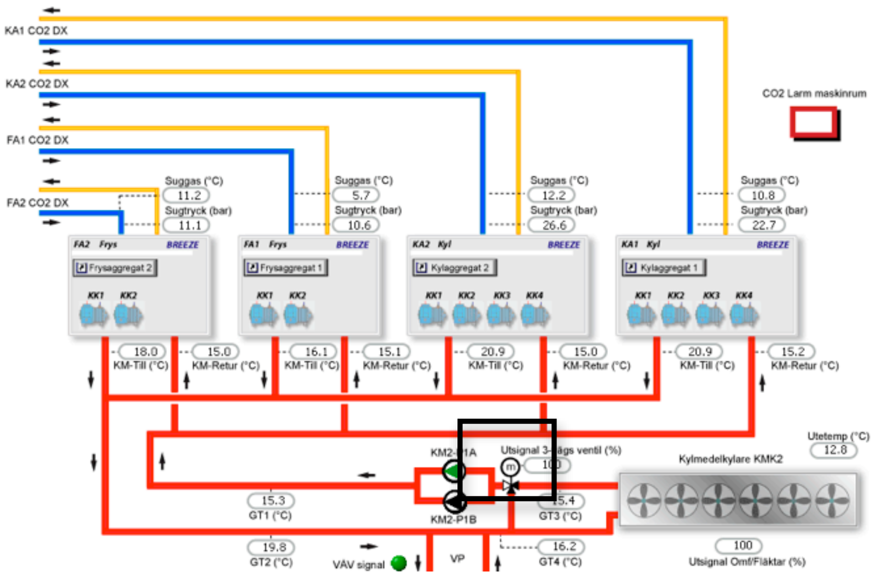


Figure 6.13 - A picture from IWMAC of the machine room in TR1.

Figure 6.14 shows how much the valve is open from mars 2008 to the end of February 2009. 100% means that it is completely open and does not recirculate the brine. The valve is often at 100% during summer and this is why the temperature of the heat sink is close to outside temperature during summer in figure 6.12. The curve (red) is the outside temperature during the period.

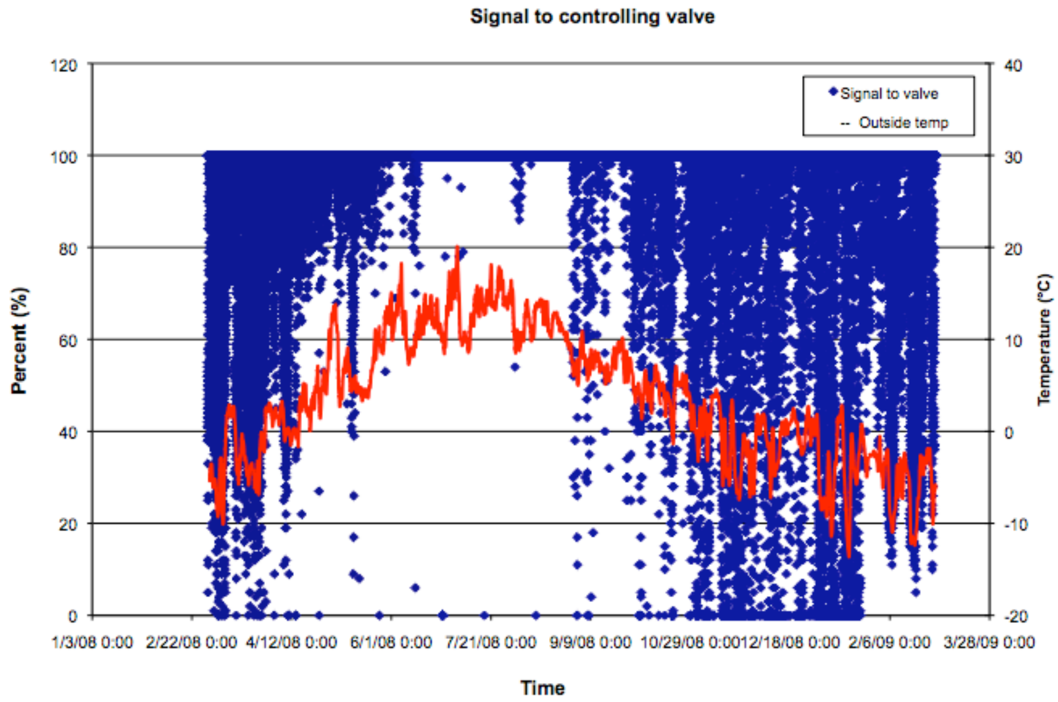


Figure 6.14 - The outside temperature and operation signal of the valve for a year.

The reason for not allowing the CO_2 to condense at too low pressure (temperature) is that the compressors need a certain pressure difference to work properly.

6.2 Trans-critical system 2

The supermarket has been investigated from September 2008 to February 2009 for the three units KA3, KAFA1 and KAFA2.

6.2.1. KA3-unit

The COP, cooling capacity and compressor power for KA3-unit during this period can be found in figure 6.15. It is a monthly average.

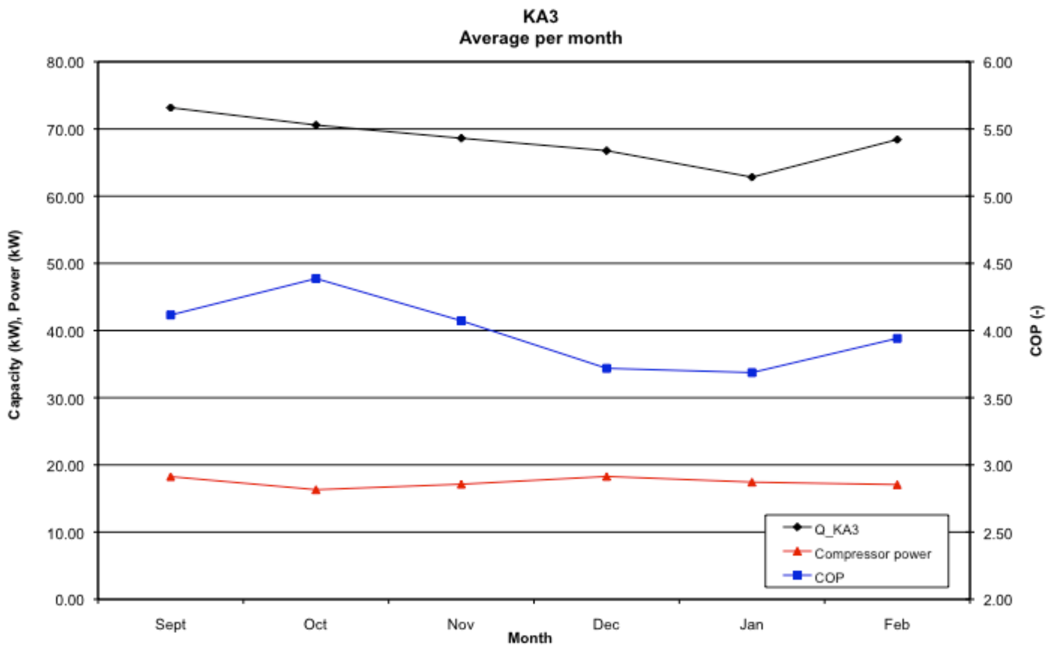


Figure 6.15 - Monthly average of the COP, cooling capacity and compressor power for KA3-unit.

The compressor power is more or less constant during the period and is about 18 kW but should in a general case be lower in the winter. The cooling capacity is decreased from September to January with about 10 kW, which is expected since the load will decrease when the surrounding temperature is colder. The COP is decreasing from October to January, which should normally not be the case since the outside temperature is decreased during this period, see figure 6.18 below, and therefore the COP should be improved during colder periods. The COP is dependent on the cooling capacity but also the power. When the compressor power is constant, as in this case, and the cooling capacity is decreased the COP will be lower. This will be further explained in the end of this section when looking at figure 6.18.

Figures 6.16 and 6.17 show the COP, capacity and power consumption in KA3-unit in September and January. In September there are two or three compressors running during opening-hours. In January there are one and two compressors running.

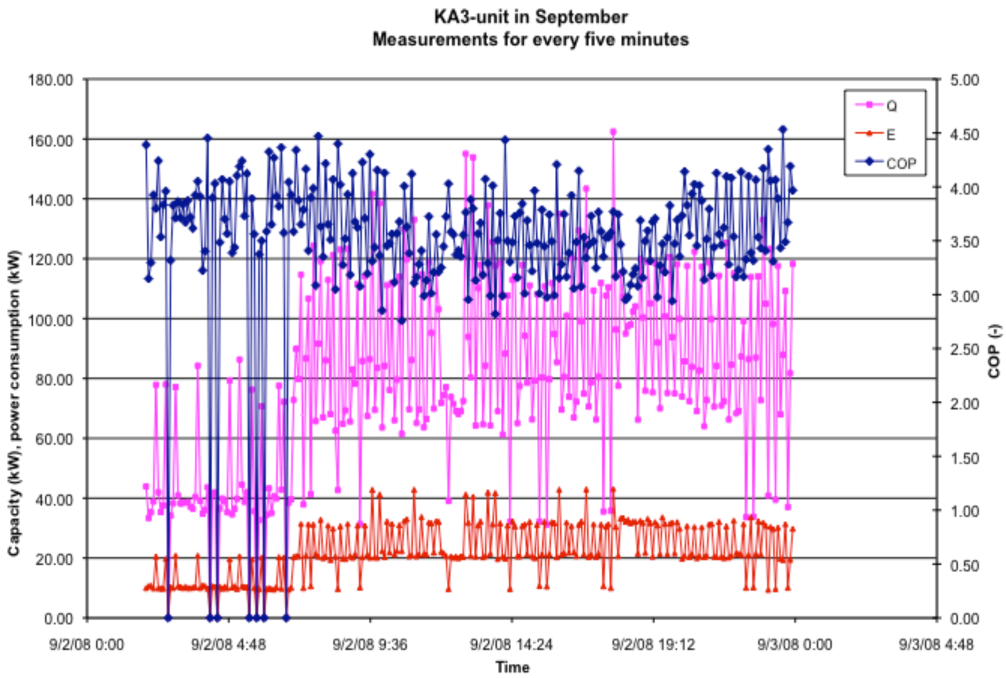


Figure 6.16 - The power, capacity and COP of KA3-unit in September in TR2.

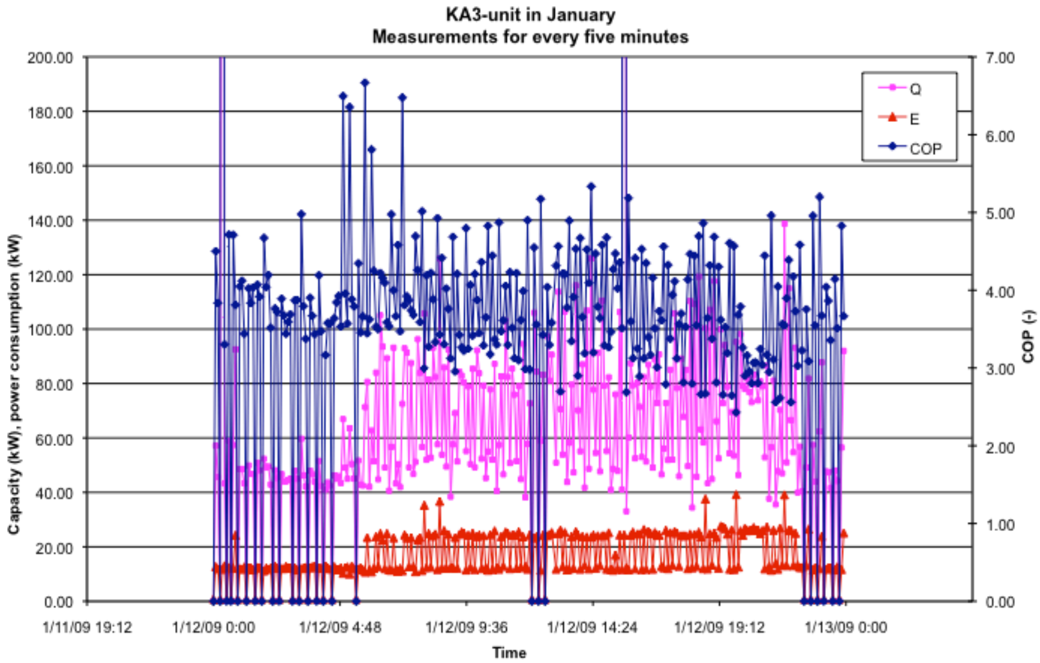


Figure 6.17 - The power, capacity and COP of KA3-unit in January in TR2.

Figure 6.18 shows the monthly average of temperatures in KA3 in trans-critical system 2. T_gascooler_outlet is the temperature out of the gas cooler and T_before_exp_valve is after the sub cooler and before the expansion valve.

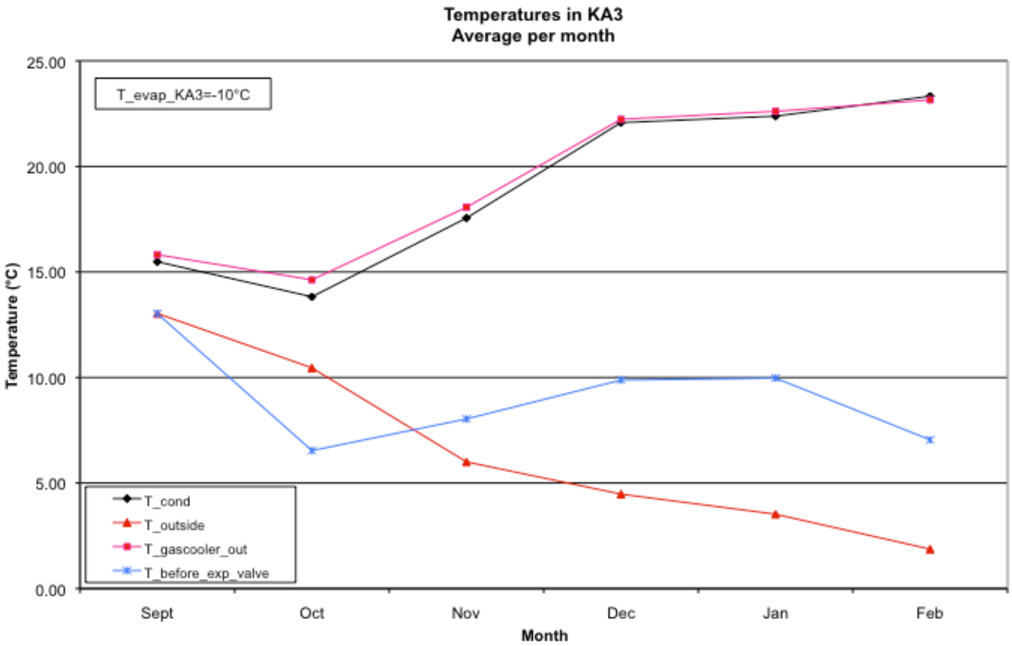


Figure 6.18 - Monthly average of temperatures in KA3-unit in TR2.

The outside air temperature is constantly decreasing during the period and the evaporation temperature is constant of -10°C. Between September and October the condensing temperature and gas cooler outlet temperature is following the same trend. From October the condensing temperature is increased while the outside temperature continue to decrease, the reason for this is to recover some heat from the system in the heat exchanger placed before the gas cooler. The heat recovery system is requiring heat and the refrigeration system must keep the high pressure to be able to deliver the heat that is necessary, the higher the pressure is the more heat the system is able to recover. This means that the colder the outside temperature is the higher the high pressure must be hold to be able to maintain the heat recovery system. This can be seen in the figure since the condensing temperature is increased. To be able to have the high condensing temperatures the compressors requires extra power even though the load is decreasing and therefore the power consumption is more or less constant as it was seen in figure 6.15.

Since the temperature out of the gas cooler is so high there is the extra heat exchanger for the ground heat sink, which subcools the refrigerant and recover heat. The difference between the gas cooler outlet temperature and the temperature before the expansion valve is the extra recovered heat in the sub cooler from the system. The difference is about 15°C in February. This sub cooler makes it possible for the system to have such high pressures in the gas cooler without losing too much of the efficiency of the system.

6.2.2 KAFA-units

Figure 6.19 shows the compressor powers for the KAFA-units in trans-critical system 2. There are two power levels, the high stage compressors (E_KA1, E_KA2) and the booster compressors (E_FA1, E_FA2). It is a monthly average.

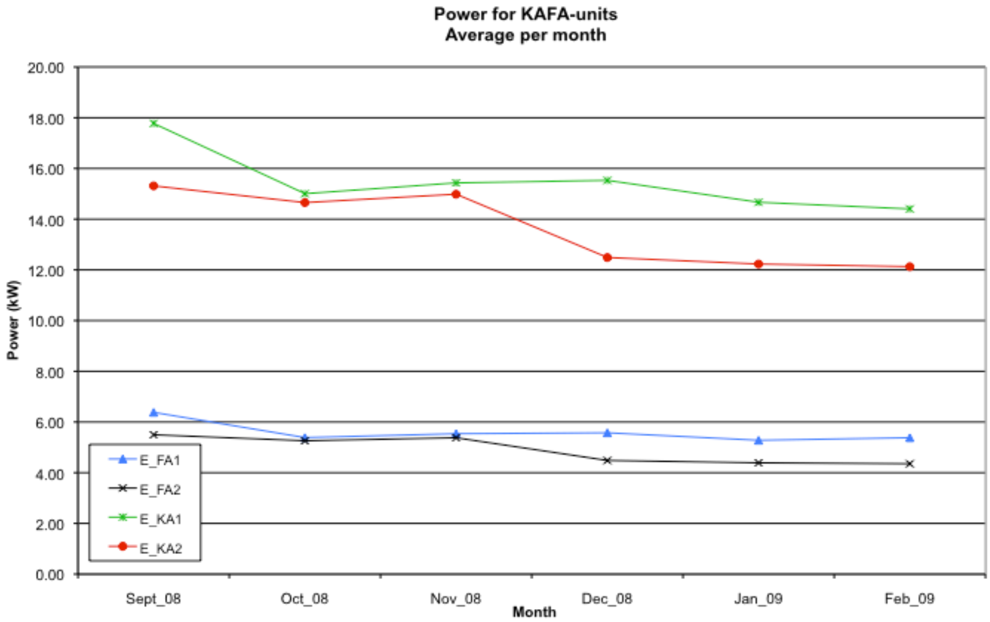


Figure 6.19 - Monthly average of the compressor power in KAFA-units in TR2.

The high stage compressors require more energy than the booster compressors. They are both operating with a pressure ratio of about 2 but the high stage compressors are operating with higher inlet and outlet temperature of the refrigerant than the booster compressors, which requires higher energy input. The mass flow is significantly higher in the high stage compressors than in the booster compressors, which also affect the energy use.

Figure 6.20 shows the cooling capacity for both KAFA-units in trans-critical system 2. The capacity is slightly increased during wintertime which should normally not be the case.

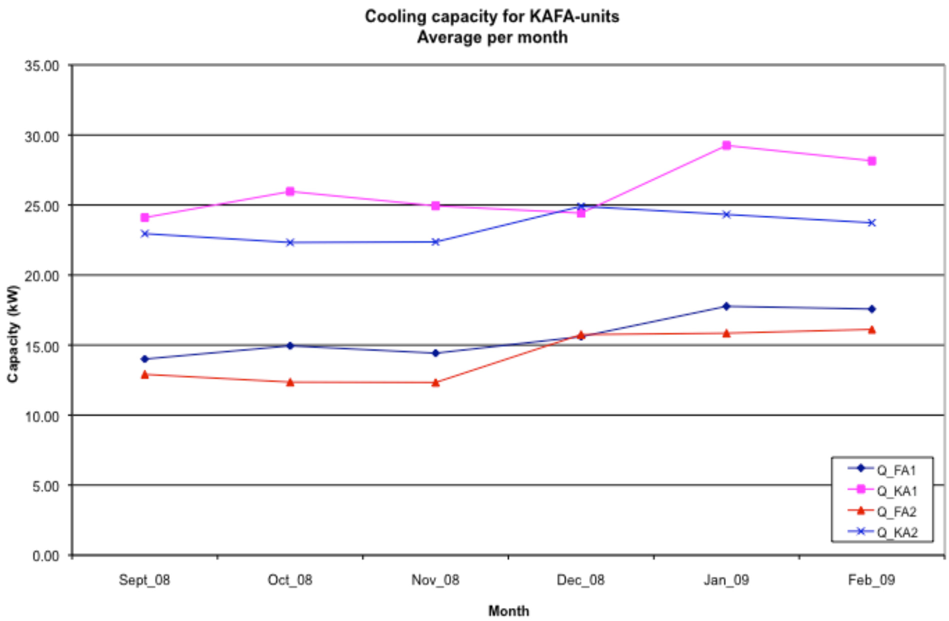


Figure 6.20 - Monthly average of the cooling capacity in KAFA-units in TR2.

Figure 6.21 shows the monthly average of the COP for the KAFA-units in trans-critical system 2.

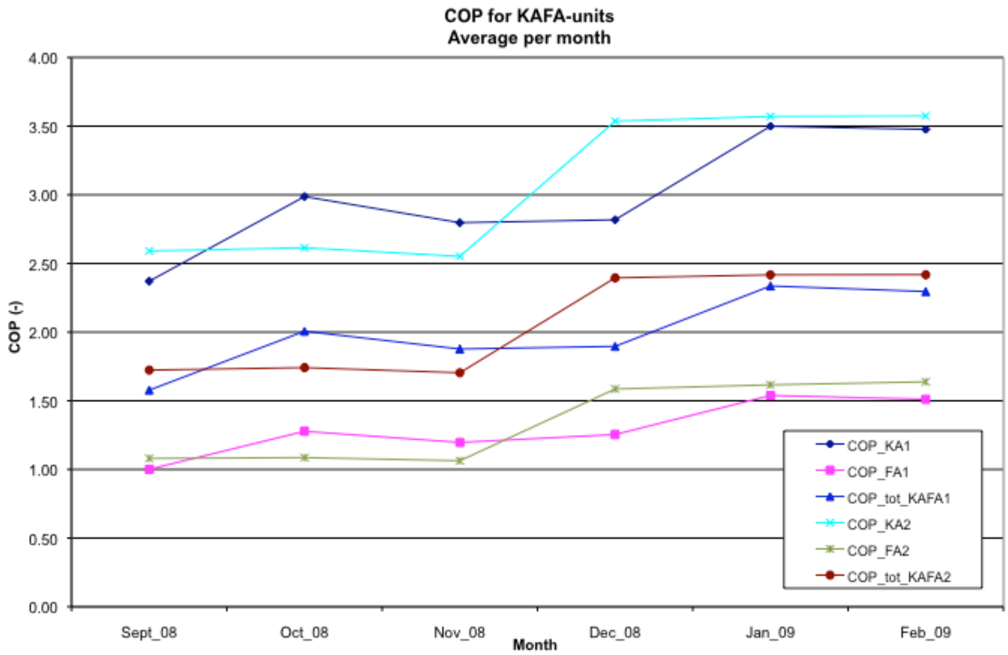


Figure 6.21 - Monthly average of COP for KAFA-units in TR2.

There are three levels of COP, COP of the high stage compressors (COP_KA1, COP_KA2), COP for the freezers (COP_FA1, COP_FA2) and the total COP of the unit (COP_tot_KAFA1, COP_tot_KAFA2). The COP is increased during the period September to February. In KAFA1-unit there is a big step between December and January and this is probably due to the assumption that was made during the calculations, which was explained in the evaluation method in section 5.2.1. In KAFA2-unit there is a big step between November and December, which can be explained by looking at the temperatures in figure 6.23 further below.

Figure 6.22 shows the monthly average of temperatures in KAFA1-unit in trans-critical system 2 from September 2008 to February 2009.

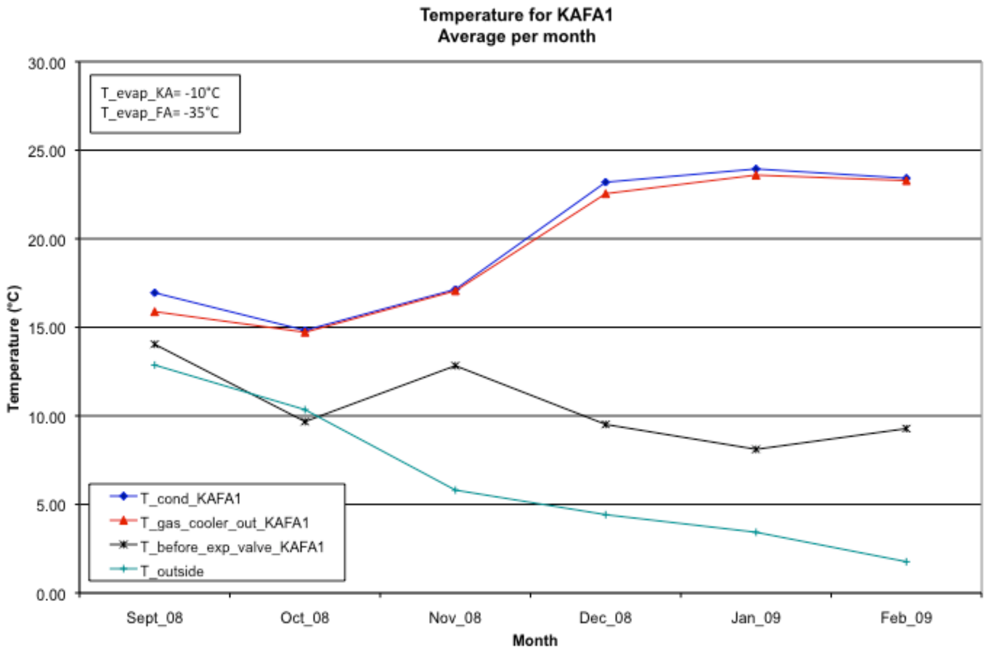


Figure 6.22 - Monthly average of temperatures for KAFA1 in TR2.

The outside temperature is decreased during the period. The condensing and the gas cooler outlet temperatures are following the curve for the temperature before the expansion valve in September to November and there is only a small temperature difference. From December the temperature difference between the gas cooler outlet and temperature before expansion valve is increased and more heat is recovered from the system. The temperature difference in February is almost 15 °C. The reason for increasing the condensing temperature is the same as for KA3-unit related to heat recovery.

Figure 6.23 shows the monthly average of temperatures in KAFA2-unit in trans-critical system 2 from September to February.

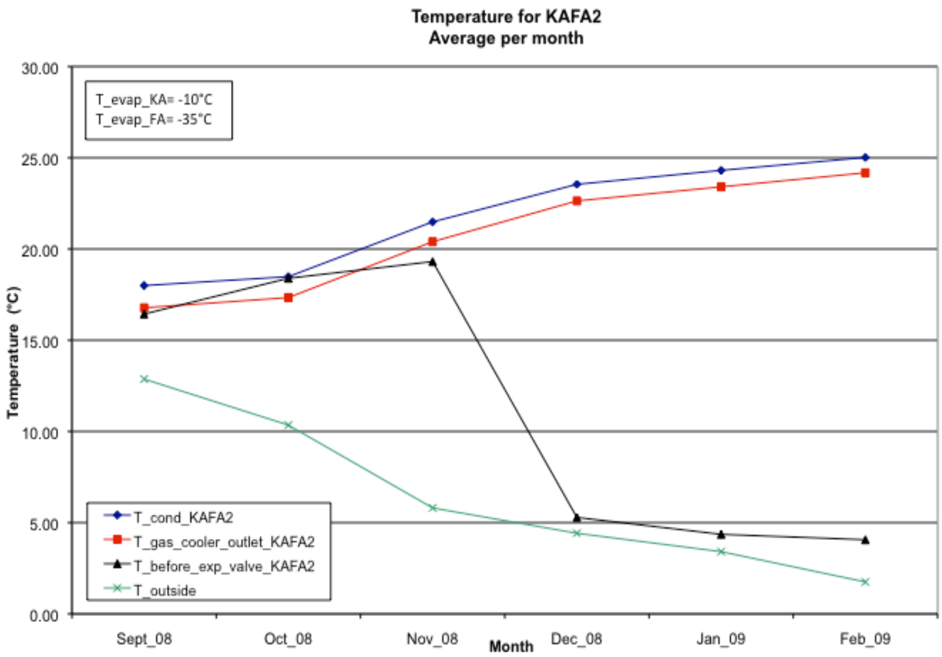


Figure 6.23 - Monthly average of temperature in KAFA2-unit in TR2.

The difference in this unit compared to KAFA1 is that the temperature out of the gas cooler is close to the temperature before the expansion valve and therefore no heat recovery exists in September to November for this unit. Between November and December the heat recovery starts and in December there is a temperature difference of 20°C, higher than in KAFA1-unit. This heat recovery step between November and December is reflected in the COP and can be seen in figure 6.21 above. The COP is increased when the refrigerant is more subcooled and this shows that the sub cooler is necessary for the system to get a good efficiency.

Figures 6.24 and 6.25 show the power, COP and capacity in KAFA-unit in September and January.

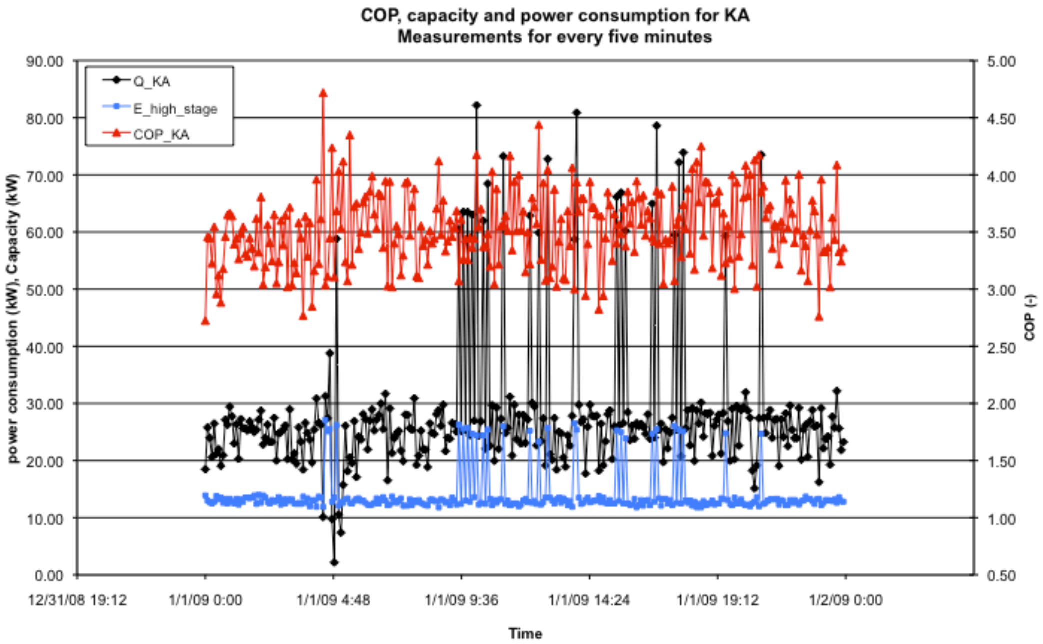


Figure 6.24 - The power, capacity and COP for the chillers in KAFA-unit January in TR2.

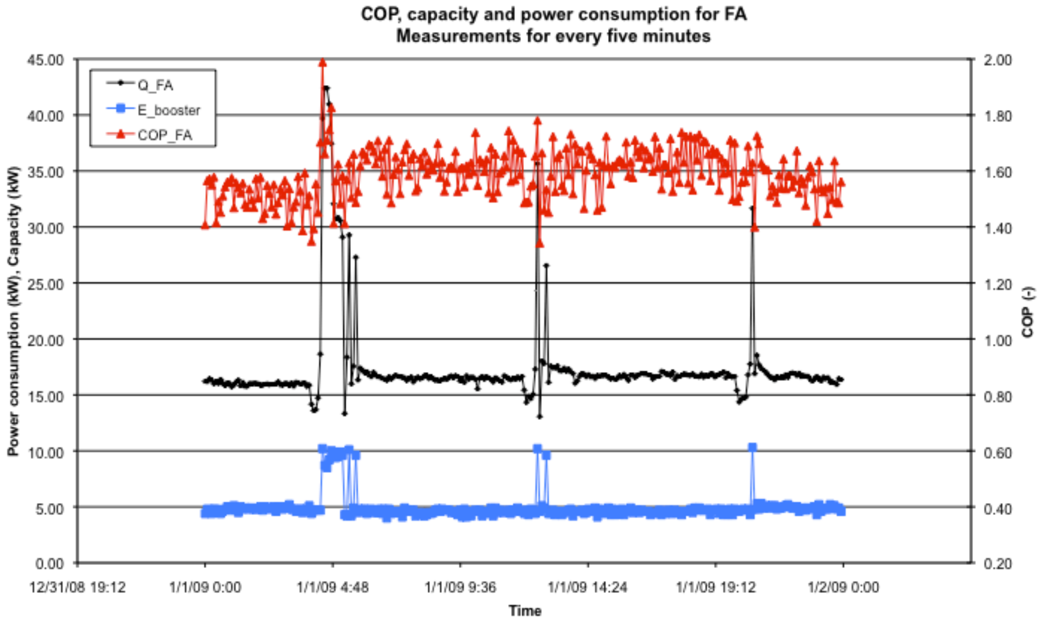


Figure 6.25 - The power, capacity and COP for the freezers in KAFA-unit January in TR2.

6.3 Cascade system 1

The supermarket has been investigated for almost three month, from the middle of December 2008 until February 2009. Since the period is short the result will be presented in table 6.3.1.

Month	Total Power (kW)	Q_KS5 (kW)	Q_KS6 (kW)	Q_cab (kW)	COP_ medium	COP_ freezer	COP_tot
Dec_08	69.44	24.52	22.12	118.95	3.07	1.29	2.20
Jan_09	65.36	24.40	22.11	117.58	3.21	1.32	2.27
Feb_09	62.89	23.47	21.30	107.07	3.13	1.27	2.16

Table 6.3.1 – COP, capacity and total compressor power for CC1.

The table shows the cooling capacity for the two CO_2-units (KS5 and KS6) and the cooling capacity for the medium temperature side supplied by the brine. The COP for the high stage compressor is around 3 in the winter and the COP for the freezers is about 1,3, which includes the power consumption of the R404A compressors that covers the freezing capacity (see equations 5.27-5.28). The total COP for the system is about 2,20.

Table 6.3.2 shows the temperature of the coolant that condenses the R404A, the evaporating and condensing temperatures and the pressure ratio in the system. Table 6.3.3 summarize the same parameters but for the CO_2-units.

R404A				
Month	**T_heat_sink (°C)**	**T_evap (°C)**	**T_cond (°C)**	**Pressure ratio**
Dec_08	17.456	-11.62	32.29	3.73
Jan_09	15.752	-11.62	31.83	3.67
Feb_09	15.919	-12.30	30.48	3.64

Table 6.3.2 – Temperatures and pressure ratio for R404A

CO_2				
Month	**T_heat_sink (°C)**	**T_evap (°C)**	**T_cond (°C)**	**Pressure ratio**
Dec_08	-6.233	-35.50	-4.99	2.58
Jan_09	-5.94	-35.56	-5.13	2.58
Feb_09	-6.267	-36.54	-5.95	2.61

Table 6.3.3 – Temperatures and pressure ratio for CO_2

Figure 6.26 shows the total COP for the freezers and the chillers and the total capacity for freezers and chillers during one day.

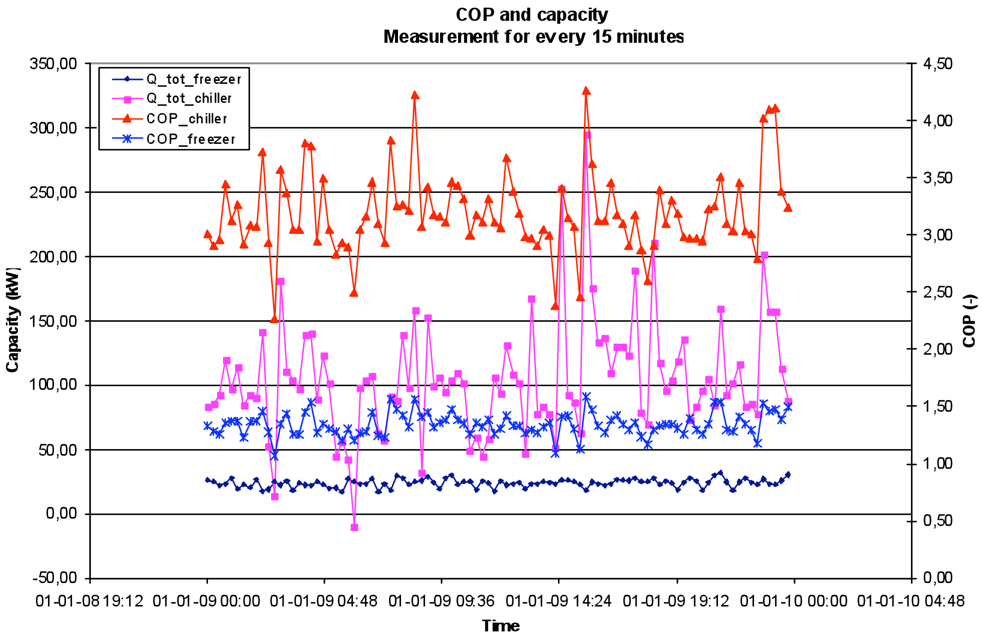


Figure 6.26 - COP and capacity during 24 hours in January in CC1

Figure 6.27 shows the temperatures for one day in January in cascade system 1.

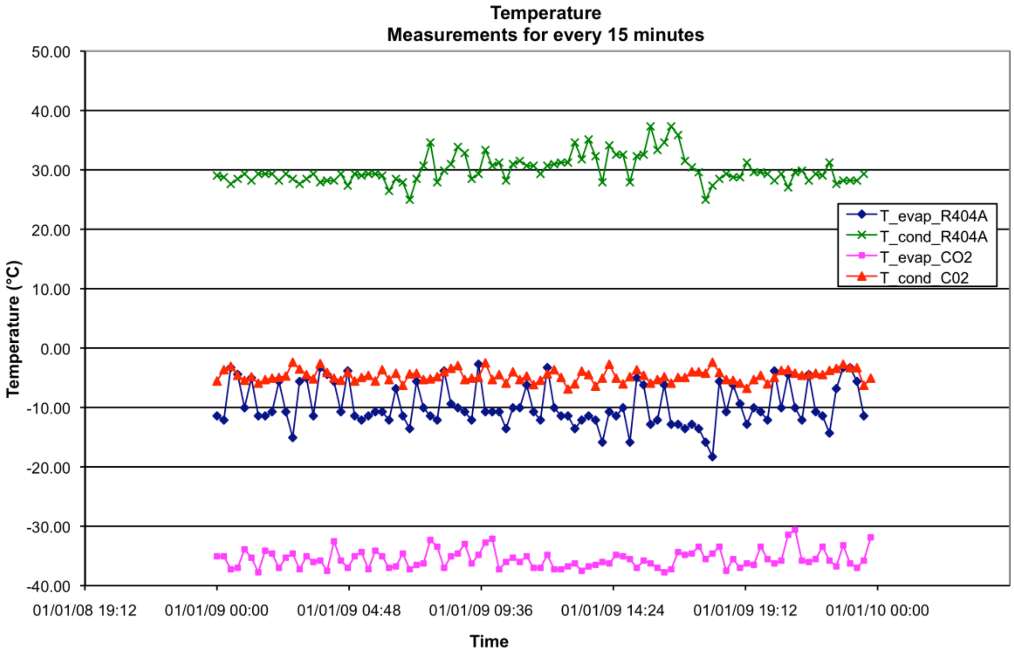


Figure 6.27 - Temperature during 24 hours in January in CC1.

7 Comparison between the systems

This section compares the results of the three system solutions. The figures are based on the monthly averages that were presented in the results section. Trans-critical system 1 will at some comparisons be divided into two periods. One period is between January 2008 and July 2008. The other period is between August 2008 and February 2009. The reason for this division is to make the plots easier to read and to distinguish between two modes of operation. The system was controlled in a way to allow the condensing pressure to float during the second period (autumn and winter of 2009).

Figure 7.1 shows the outside temperature for the supermarkets and figure 7.2 shows the condensing temperatures of the three systems.

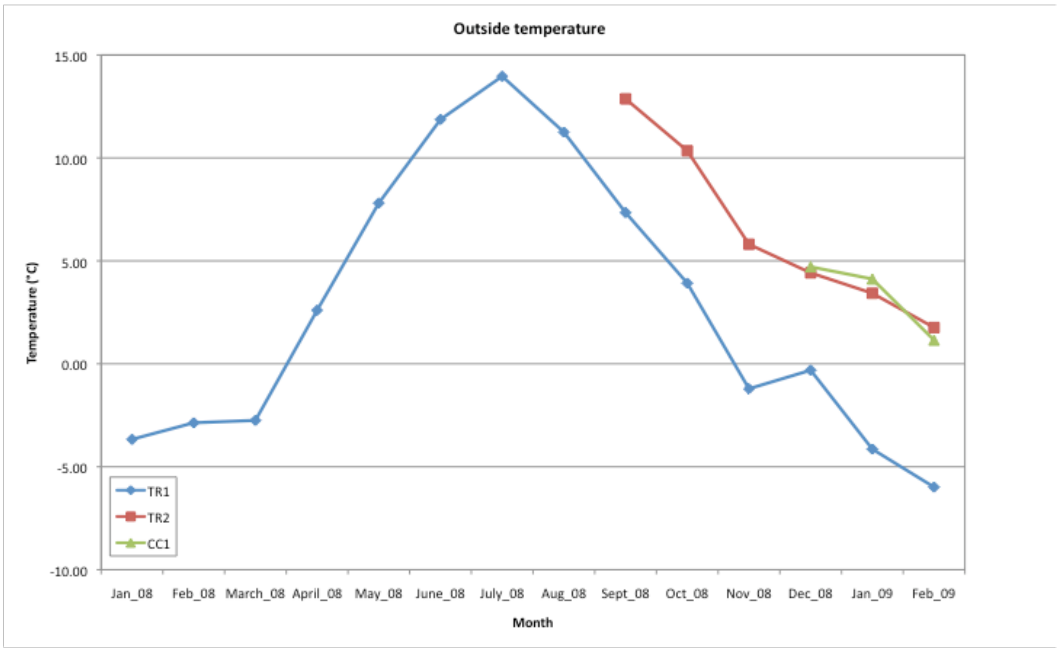


Figure 7.1 - The outside temperature for the supermarkets

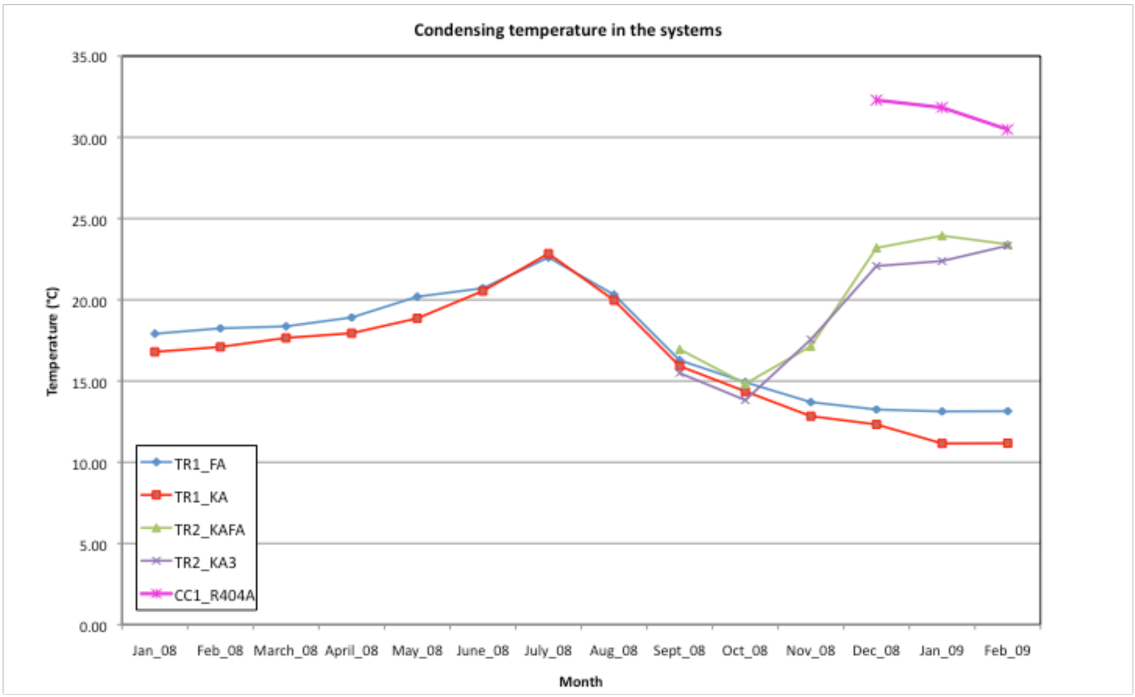


Figure 7.2 - The condensing temperature in the systems

Evaporating temperatures are not varying much during the year and are similar in all systems. The condensing temperature for trans-critical system 2 is much higher than for trans-critical system 1, which was explained in section 6 and is related to the heat recovery system in TR2. The condensing temperature for R404A is the highest in the plot and that is also due to the heat recovery system that exists in that supermarket.

Figure 7.3 shows the relation between the condensing temperature and the outside temperature.

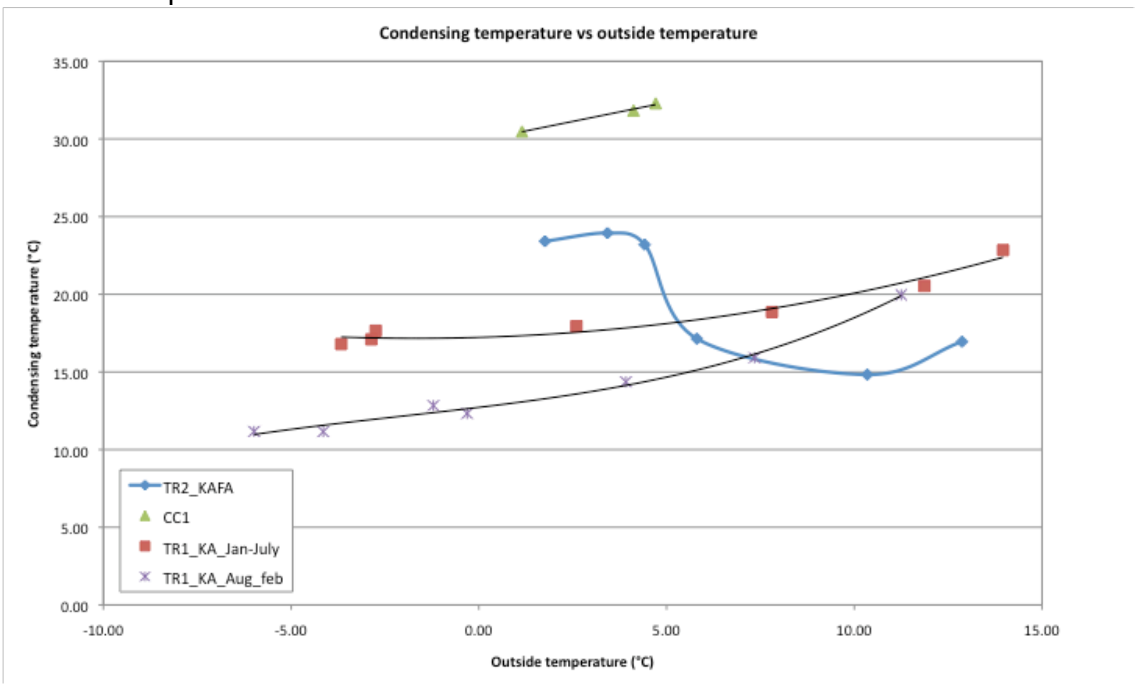


Figure 7.3 – The relation between the condensing temperature and the outside temperature

As it can be seen in the figure the condensing temperature increases with the ambient temperature. This graph clearly shows that the condensing temperature is higher in TR1 in the first period (January-July) than in the second period (August-February). During the second period the system it is allowed to condense at lower temperature, which was shown in previous graphs in section 6. When the ambient temperature is low the TR2 system is controlled to produce more heat to the heating system, this is done by increasing the condensing temperature/pressure. In this graph the change can be seen when the outside temperature is about 5 °C.

Figure 7.4 shows the total power and the total capacity for the supermarkets, which is the sum of the capacities at the low and medium temperature level.

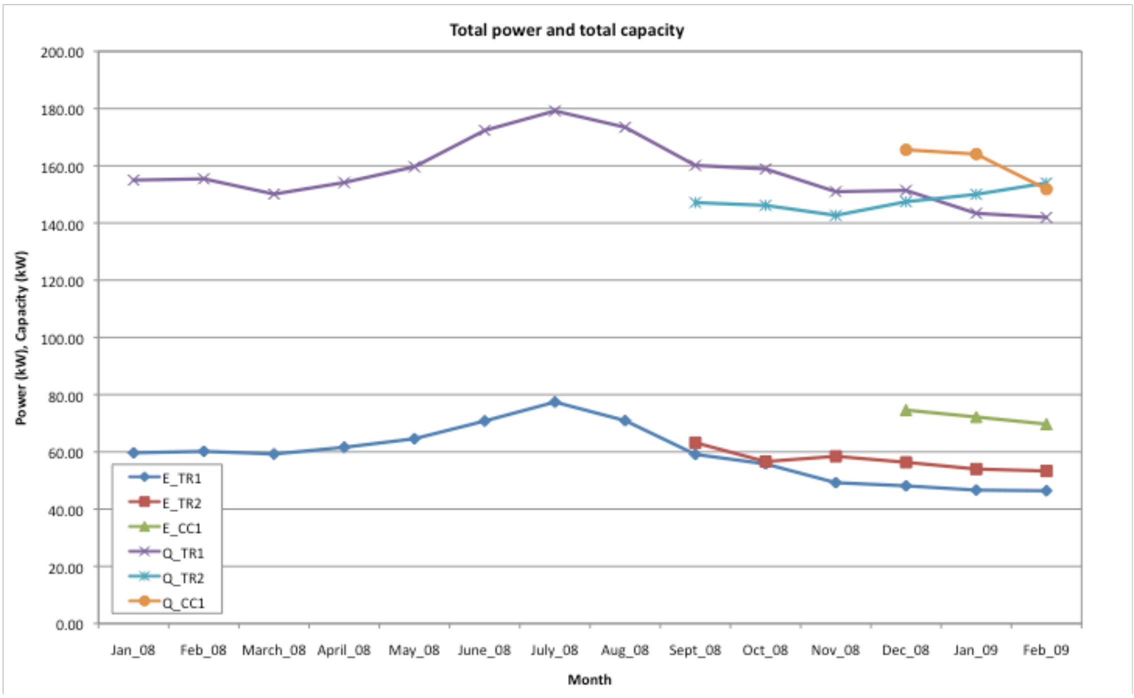


Figure 7.4 - Total capacity and total compressor power consumption for all three systems

Total cooling capacities for the three supermarkets are comparable. The power consumption is a bit higher for the cascade solution. One explanation to this might be due to the extra power that is needed for the pumps circulating the brine. Another reason is the high condensing temperature of R404A, due to heat recovery needs, which increase the power consumption of the R404A compressors.

Figure 7.5 shows the overall COP of the systems, which includes both chillers and freezers. It is the total capacity divided by the total power consumption of all compressors in the system.

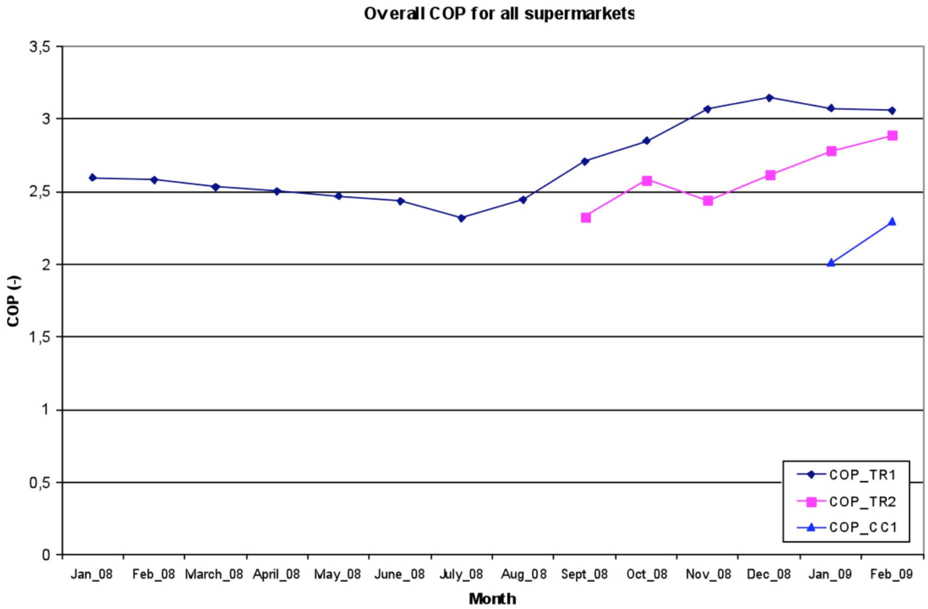


Figure 7.5 - Overall COP for all three systems

Trans-critical system 1 is the system that has the highest overall COP and that is due the changes that lowered the condensing temperature in the system. The cascade solution has the lowest overall COP. This can be clearly seen in the figure and is due to the high power consumption shown in figure 7.4.

Since the systems operate in different locations, the ambient temperatures will be different, as can be seen in figure 7.1, and it is hard to relate the performance of the systems based on the months plot. Therefore, the overall COP is plotted against the ambient temperature in Figure 7.6.

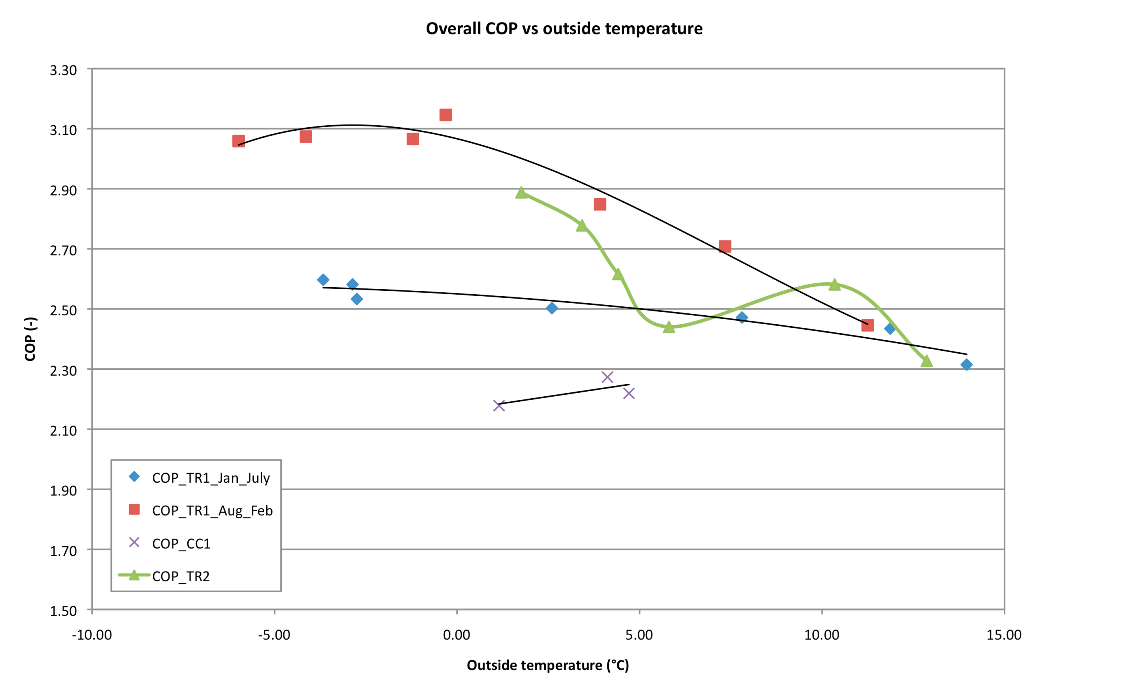


Figure 7.6 – The relation between the overall COP and the outside temperature

The figure shows that the COP is higher for the second period in trans-critical system 1 than for the first period. It can also be seen that the inclination is larger for the second period due to that the system is controlled differently than in the first period. In the second period the system was not controlled to recover heat. The COP of TR2 is not following the trend to decrease with lower ambient temperatures. This is because the changing condensing temperature due to heat recovery requirements at low ambient temperatures. The borehole operated to sub-cool the system during the period where heat recovery is needed is also affecting the COP. The behavior of the COP is a combination of the borehole influence and the heat recovery requirement.

The overall COP of the cascade solution is rather constant since the condensing temperature in the system is not changing much,

Figure 7.7 shows the load ratio for the systems. The capacity ratio is calculated by equation 7.1.

$$Load\ ratio = \frac{\dot{Q}_{total,chillers}}{\dot{Q}_{total,freezers}} \quad [-] \tag{7.1}$$

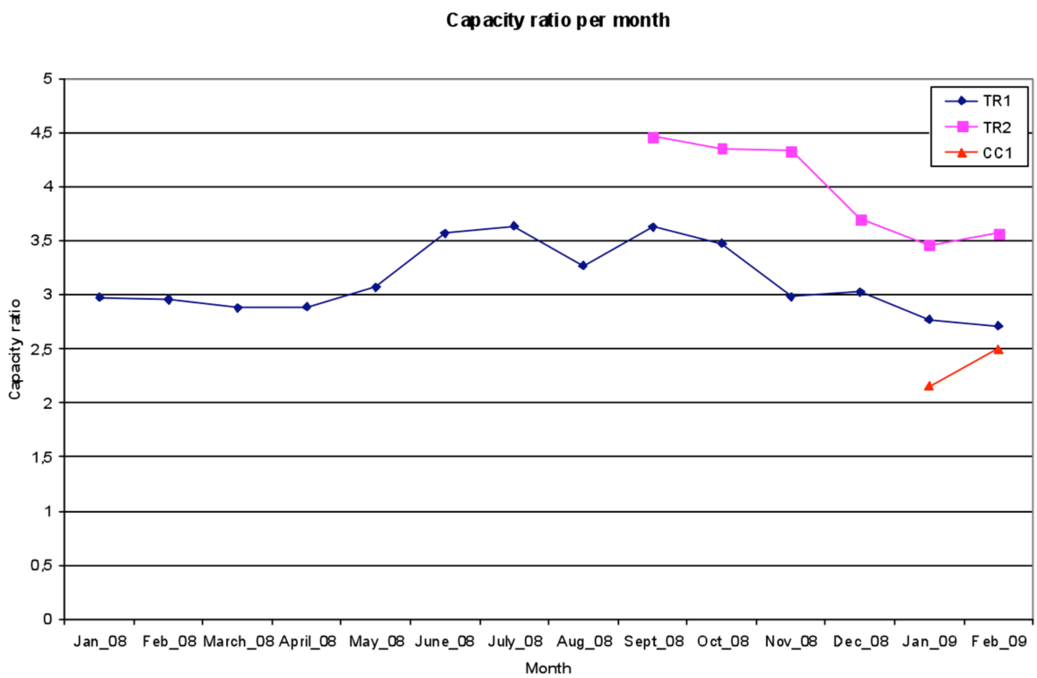


Figure 7.7 - Capacity ratio for the three systems

Figure 7.7 shows that the load is differently distributed between freezers and chillers in the different supermarkets which make it harder to compare the overall COP in figure 7.5. The COP is always higher on the medium temperature side than on the low temperature side. A high load on the medium temperature side compared to the other systems, is then positively affecting the overall COP.

Figure 7.8 and figure 7.9 shows the COP for the medium temperature level and low temperature level.

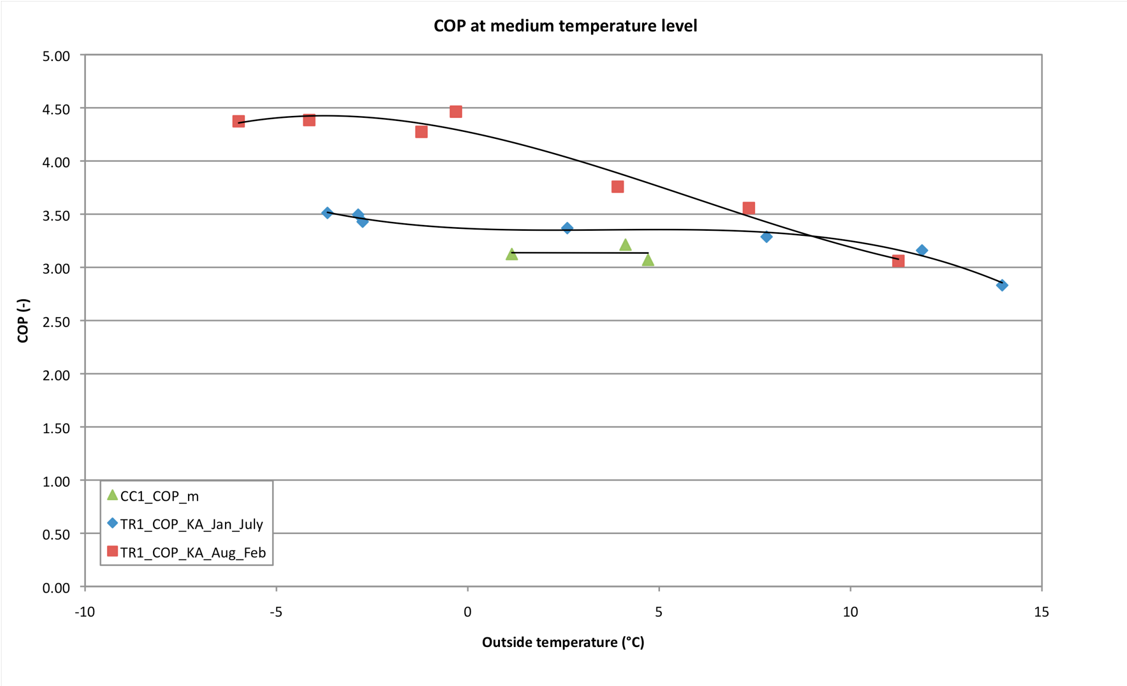


Figure 7.8 - The COP at medium temperature level

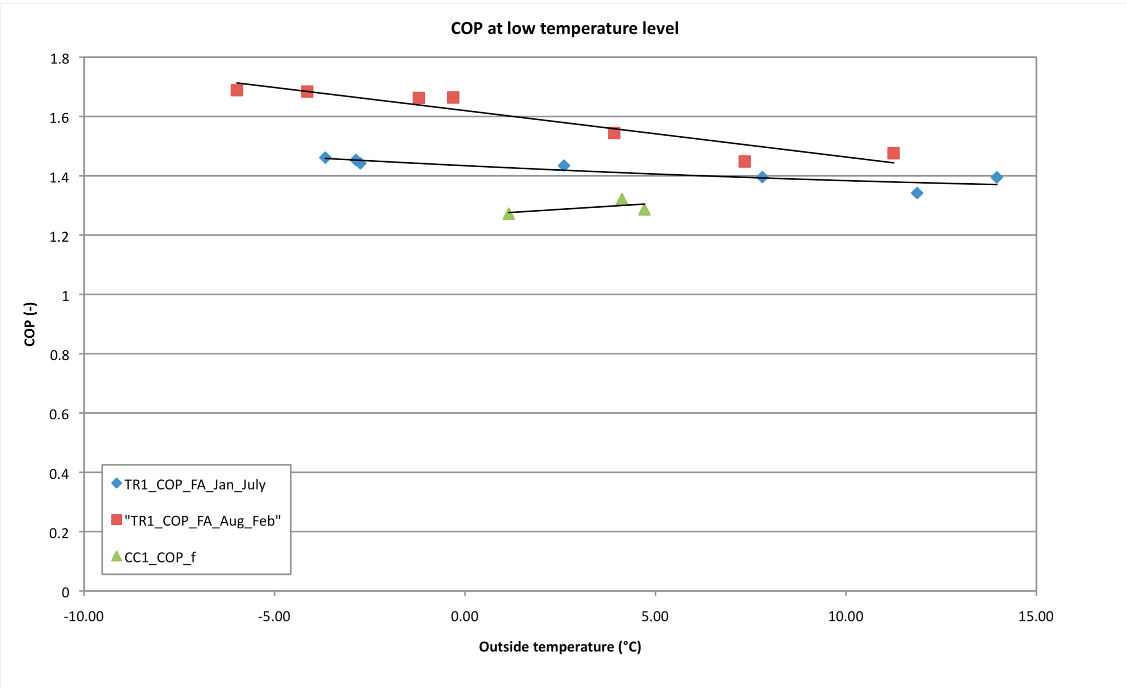


Figure 7.9 – The COP at low temperature level

Figure 7.10 shows how the overall COP for the three systems would look like if the supermarkets have the same ambient temperature and the same capacity conditions on both freezers and chillers. The location that was used in these calculations is for TR2. For calculations see appendix 5.

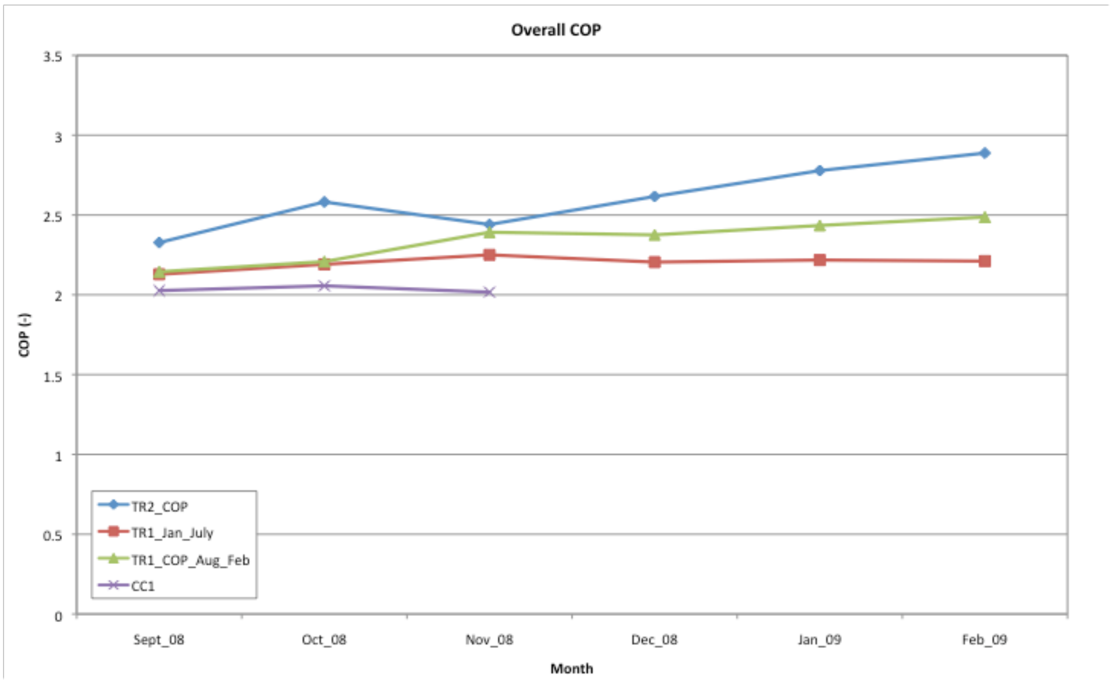


Figure 7.10 – the overall COP if the supermarkets would have the same capacity and outside temperature conditions

These calculations are based on the monthly average that was presented in section 6. This graph is shown to give an indication of how the systems are operating compared to each other at the same capacity and ambient conditions. TR1 is separated into two periods and as it has been shown before the second period has the highest COP.

The cascade system has in this case the lowest COP and one reason is the high condensing temperature that forces the compressors to work harder, which is related to heat recovery requirements in the system. At some periods, the TR1 solution has higher condensing temperature than the TR2 system, which depends on the heat requirements mainly in the TR2 system. Also the operation of the borehole in TR2 improves the COP of the system even it is operating at higher condensing temperature than TR1. However; the borehole connection to the refrigeration system is not a conventional solution and will improve the COP of TR1 and CC if applied. From this analysis the difficulties in comparing real installations can be clearly seen, each system has different operating conditions depending on the location, system solution and requirements in the supermarket. This shows the need and importance of combining the experimental work with computer simulations where the influence of different system parameters can be investigated separately.

8 Discussion

The interest in using carbon dioxide as refrigerant in supermarket refrigeration systems has increased during the past years and CO_2 has already been applied in many supermarkets around the country. This thesis is the first step in a project where CO_2 refrigeration systems will be evaluated in real field installations and compared to conventional installations.

The oil cooler capacity has, as mentioned earlier, been one of the major challenges in this project and further analysis of the behavior of the oil cooler capacity is needed. There should also be further investigation if there is another way to estimate the oil cooler capacity with a more reliable result. One way could be to estimate the oil cooler capacity from the coolant side in trans-critical system 1 by knowing the properties and inlet and outlet temperatures of the coolant.

From the results it can be seen that the changes that has been done in trans-critical system 1 is very favorable for the system since the COP is higher in winter 2009 than in year 2008. When comparing one month to the same month the year before with the same outside temperature, it could be seen that the condensing temperature was lowered with about 5°C and the COP increased from 3,5 to 4,5. By letting the condensing temperature decrease and keeping the level of the evaporation temperature the compressor work can be reduced since the pressure ratio is decreased.

When using CO_2 as refrigerant the temperature after the compressor is rather high and therefore increasing the possibilities for heat recovery. This solution is done in trans-critical system 2 where the pressure is kept high to be able to recover the necessary heat. TR2 system is operated with floating condensing where the system runs with the lowest condensing pressure possible, following the ambient temperature, however; when the outdoor temperature reaches a level where heating is need in the supermarket the condensing pressure/temperature is increased so more heat can be recovered from the system. Since the pressure is kept high the compressors have to work harder than they “normally” should do. The sub cooler that recovers heat to the ground heat sink is of importance for the efficiency of the system since sub cooling increase the capacity of the system.

The assumption that was done for the KAFA-units in trans-critical system 2 was that the energy that goes to the booster compressors is a constant percentage of the total energy. It could be seen in the result section that this assumption has an impact on the result. When more months have passed and more data is available the assumption should be analyzed and maybe corrected.

The benefits from using only CO_2 is that there is no need for pumps to circulate a brine as there is in cascade system 1. The additional pumps require energy and have an adverse effect on the performance of the system. A cascade solution gives extra temperature differences meaning that there is a temperature difference between the evaporating temperature of R404A and the brine temperature. There is another temperature difference between the brine and the CO_2. These temperature losses are negatively affecting the efficiency of the system.

From the comparison it can be seen that the cascade solution has about the same capacity as the trans-critical systems but higher power consumption. This is mainly due to the additional pumps and the high condensing temperature. It is negatively affecting the overall COP, which was shown in section 7.

Figure 7.10 shows that the cascade solution has the lowest COP compared to the other systems at the same capacity and ambient conditions. It is important to mention that this lower COP is due to the operating conditions and not necessarily due to the system solution. If the cascade system would operate in these conditions in reality the condensing temperature, etc., would probably be different, which would affect the overall COP. The ground heat sink in TR2 is of big importance for the overall COP in the system. If CC1 or TR1 would have the same solution the relative COP results might be different.

Further on in this project the supermarkets will be modeled in EES. This step will be of big importance for the final results. Then it will be possible to see how the systems actually would operate at different conditions and how that would affect the condensing temperature, etc. It will also be possible to see how a system would operate without heat recovery and borehole sub-cooling, etc.

As mentioned, this thesis is the first step in this project. It will be interesting to see the studied supermarkets compared to more conventional solutions, which would show how well the CO_2 systems are operating. Since all the systems have different solutions and operating conditions key parameters that are easy to compare must be found.

9 Conclusions

After the investigations that have been done in this project so far, it can clearly be seen that CO_2 has high potential to work as refrigerant in supermarket refrigeration. The COP is about 3-5 for medium temperature side and 1,2-1,8 for the low temperature side depending on the season, shown in figure 7.8 and 7.9. The total cooling capacities is comparable between the three systems and is between 140-180 kW depending on the season. TR2 has the highest capacity ratio.

The analysis in this thesis has shown that it has been difficult to estimate the oil cooler capacity in these supermarkets. It has also shown that the capacity is higher during the winter period than in the summer. It is of importance to estimate the oil cooler capacity correct in order to apply an energy balance over the compressors.

The comparison between the systems is dependent on the operating conditions and system requirements. It is hard to compare field installations therefore computer simulation modeling is an essential tool to be combined with the experimental measurements.

The benefits of CO_2 are all indicating that CO_2 is a good solution but more supermarkets must be investigated and finally compared to conventional supermarkets. After that it will be shown how the CO_2 systems are operating.

9.1 Future work in the project

The oil cooler capacity must be further analysed and other methods of estimating the capacity should be developed and investigated to see if there is better methods.

Six more supermarkets with carbon dioxide will be investigated. Conventional supermarkets with R404A and without CO_2 will also be evaluated and in the end of this project these supermarkets can be compared to see if CO_2 is suitable to use as a refrigerant.

Computer simulation models of the refrigeration systems in all the investigated supermarkets will be developed in order to properly compare the systems and investigate the influence of different parameters on each system's efficiency.

10 References

Arias J., Vahlén P., Westman O. (2005), ”Utvärdering av två butiker med transkritiska kylsystem med CO_2 som köldmedium”, Report, Stockholm, Sweden

Axell M. (2001), “Butikskyla” Report EFFEKTIV 2001:05

Axell M., Lindberg U., Lidbom P. (2004), “Energieffektivisering i livsmedelsbutiker med målsättning att förbättra klimatet för kunder, varor och personal – en fältstudie i tre butiker”, SP Swedish National Testing and Research Institute, Borås, Sweden

Axima refrigeration (2008), “CO_2 in refrigeration”, available at: http://www.aximaref.com/english/co2/refrigeration/index_refrigeration.php, as accessed 2008-12-13

Butler A, (2007), “CO_2 Transcritical Cooling with Heat Reclaim”, available at: http://www.r744.com/conferences/files/07_rac_space_engineering.pdf?PHPSESSID=48df58c635ddee0c6777181e48714119 , Paper presented at RAC Cooling with Carbon Dioxide Conference, March 28, 2007

Campbell, A., Maidment, G.G., Missenden, J.F. (2006), “A natural refrigeration system for supermarkets using CO_2 as a refrigerant”, available at: http://www.cibse.org/pdfs/4a%20A%20Campbell.pdf, paper presented at CIBSE National Conference, March 21-22, 2006

Campbell A., Maidment G.G., Missenden J.F. (2007), “Carbon dioxide for supermarkets”, available at: http://www.r744.com/papers/pdf/pdf_185.pdf, Paper presented before the Institute of Refrigeration at the Institute of Marine Engineering, London, Great Britain

Danfoss (2008), “Transcritical refrigeration systems with Carbon dioxide (CO_2)”, available at: http://www.ra.danfoss.com/TechnicalInfo/Approvals/Files/RAPIDFiles/01/Article/TranscriticalArticle/PZ000F102_ARTICLE_Transcritical%20Refrigeration%20Systems%20with%20Carbon%20Dioxide%20(CO2).pdf, Report, Danfoss

Dorin (2009), “The widest CO_2 compressor range, carbon dioxide for all your needs”, available at: http://www.dorin.com/documents/Download/19/CO2_-0809a.pdf, product information sheet

Granryd E., Ekroth I., Lundqvist P., Melinder Å., Palm B., Rohlin P. (2005), ”Refrigerating Engineering”, Department of Energy Technology, KTH, Stockholm, Sweden

GTZ Proklima (2008), “Natural refrigerants, sustainable ozone- and climate-friendly alternatives to HCFCs”, Eschborn, Germany.

Heatpumpcentre.org (2008), “Annex 31 - Advanced modelling and tools for analysis of energy use in supermarket systems” available at: http://www.heatpumpcentre.org/Projects/Annex_31.asp as accessed 2008-12-28.

International institute of refrigeration (2000), “Carbon dioxide as a refrigerant”, available at: http://www.iifiir.org/en/doc/1013.pdf, Informatory note, Paris, France

Irhace (2008), “Alternative refrigerants, future and current options – part 2”, Two-part editorial article, Sydney, Australia

Iwmac (2009), “Centralised operation and surveillance, by use of WEB technology”, available at: http://www.iwmac.no/english/, last accessed 2009-05-12

Kharazi A.A., Akbari P., Müller N. (2005), “ Preliminary Study of a Novel R718 Compression Refrigeration Cycle Using a Three-Port Condensing Wave Rotor”, available at: http://www.egr.msu.edu/mueller/NMReferences/KharaziAkbariMueller_JGTP2005_R718RefrigerationCycleUsing3PortCondensingWaveRotor.pdf, Journal of Engineering for Gas Turbines and Power, Department of Mechanical Engineering, Michigan State University, USA

Kim M.-H., Pettersen J., Bullard C.W. (2003), ”Fundamental Process and System Design Issues in CO_2 Vapor Compression Systems”. Progress in Energy and Combustion Science 30 (2) 119–174

Naturvårdsverket (2003), “Naturliga köldmedier, exempel på lyckade installationer”, Report 5326, Sweden

Nekså, P. (2004), “CO_2 as refrigerant for systems in transcritical operation, priciples and technology status”, Part of a paper presented at AIRAH’s 2004 natural refrigerants conference, July 28, 2004, Sydney, Australia

Pearson S.F. (2003), “New, natural and alternative refrigerants”, available at: http://www.cibse.org/pdfs/6cpearson.pdf, Paper presented at CIBSE/ASHRAE conference, Edinburgh, Great Britain

R774.com (2008), available at: http://www.r744.com/faq.php, as accessed 2009-01-22

Sawalha S. (2005), “Using CO_2 in Supermarket Refrigeration”, ASHRAE Journal

Sawalha S. (2008), “Carbon dioxide in supermarket refrigeration”, Doctoral thesis, KTH, Stockholm, Sweden

SWEP (2008), "Indirect refrigerant systems", available at: http://www.swep.net/index.php?tpl=page0&lang=en&id=314 as accessed 2008-12-13

Topping D. (2004), "Carbon dioxide refrigerant makes a comeback", available at http://www.appliancemagazine.com/editorial.php?article=567&zone=1&first=1, Article in Appliance magazine, as accessed 2008-11-12

Appendix 1: Calculations of the volumetric efficiency in TR1 and TR2

Data from compressor manufacturer:

Dorin TCS373:

- KA-units in Tr1
- KA3-unit in TR2
- High stage in KAFA-units in TR2

t_evap	p_suc	tgc_out	p_dis	beta	Q (kW)	P (kW)	Swept volume (m3/hr)	Pressure ratio	RPM
-20	19.72	15	75	3.803	26	11.8	3.5	3.803245436	2900
		25	75	3.803	22.3	11.8	3.5	3.803245436	2900
		35	90	4.564	16.3	12.2	3.5	4.563894523	2900
		40	110	5.578	13.1	13.5	3.5	5.578093306	2900
-15	22.93	15	75	3.271	32	12.4	3.5	3.270824248	2900
		25	75	3.271	27.4	12.4	3.5	3.270824248	2900
		35	90	3.925	20.9	13.6	3.5	3.924989097	2900
		40	110	4.797	16.8	14.9	3.5	4.797208897	2900
-10	26.5	15	75	2.83	39.5	12.5	3.5	2.830188679	2900
		25	75	2.83	33.8	12.5	3.5	2.830188679	2900
		35	90	3.396	25.9	13.9	3.5	3.396226415	2900
		40	110	4.151	23.8	16.5	3.5	4.150943396	2900
0	34.86	15	75	2.151	56.4	14.5	3.5	2.151462995	2900
		25	75	2.151	48.5	14.5	3.5	2.151462995	2900
		35	90	2.582	37.1	16.8	3.5	2.581755594	2900
		15	100	2.869	52.7	17.9	3.5	2.868617326	2900
		25	100	2.869	46.1	17.9	3.5	2.868617326	2900
		40	110	3.155	34.2	19.5	3.5	3.155479059	2900
5	39.69	15	75	1.89	66.6	14.7	3.5	1.889644747	2900
		25	75	1.89	57.1	14.7	3.5	1.889644747	2900
		35	90	2.268	44.5	17.7	3.5	2.267573696	2900

Dorin TCDH372:

- FA-units in TR1

t_evap	p_suc	tgc_out	p_dis	beta	Q (kW)	P (kW)	Pressure ratio
-50	6.84	15	75	11	10.3	10.8	10.96491228
		25	75	11	8.7	10.8	10.96491228
-45	8.34	15	75	11	12.9	11.6	8.992805755
		25	75	11	10.9	11.6	8.992805755
-40	10.07	15	75	7.4	15.9	12.4	7.447864945
		25	75	7.4	13.4	12.4	7.447864945
		25	90	8.9	13.7	13.9	8.937437934
		35	90	8.9	10.6	13.9	8.937437934
-35	12.05	15	75	6.2	19.2	13.3	6.22406639
		25	75	6.2	16.3	13.3	6.22406639
		25	90	7.5	16.7	14.8	7.468879668
		35	90	7.5	12.9	14.8	7.468879668
		25	100	8.3	16.8	15.9	8.298755187
		35	100	8.3	13.8	15.9	8.298755187
-30	14.3	15	75	6.2	23	14	5.244755245
		25	75	6.2	19.5	14	5.244755245
		25	90	7.5	20	15.8	6.293706294
		35	90	7.5	15.6	15.8	6.293706294
		35	110	7.7	17.1	17.9	7.692307692
		45	110	7.7	12.6	17.9	7.692307692
-25	16.85	15	75	6.2	27.4	14.7	4.451038576
		25	75	6.2	23.2	14.7	4.451038576
		25	90	7.5	24	16.8	5.341246291
		35	90	7.5	18.6	16.8	5.341246291
		35	100	5.9	19.8	18	5.934718101
		35	120	7.1	20.9	20	7.121661721
		45	120	7.1	16.3	20	7.121661721
-20	19.72	15	75	6.2	32.4	15.3	3.803245436
		25	75	6.2	27.5	15.3	3.803245436
		25	90	7.5	28.4	17.7	4.563894523
		35	90	7.5	22.1	17.7	4.563894523
		35	100	5.1	23.5	19	5.070993915
		35	120	6.1	24.7	21.4	6.085192698
		45	120	6.1	19.4	21.4	6.085192698

Dorin TCS373-D:

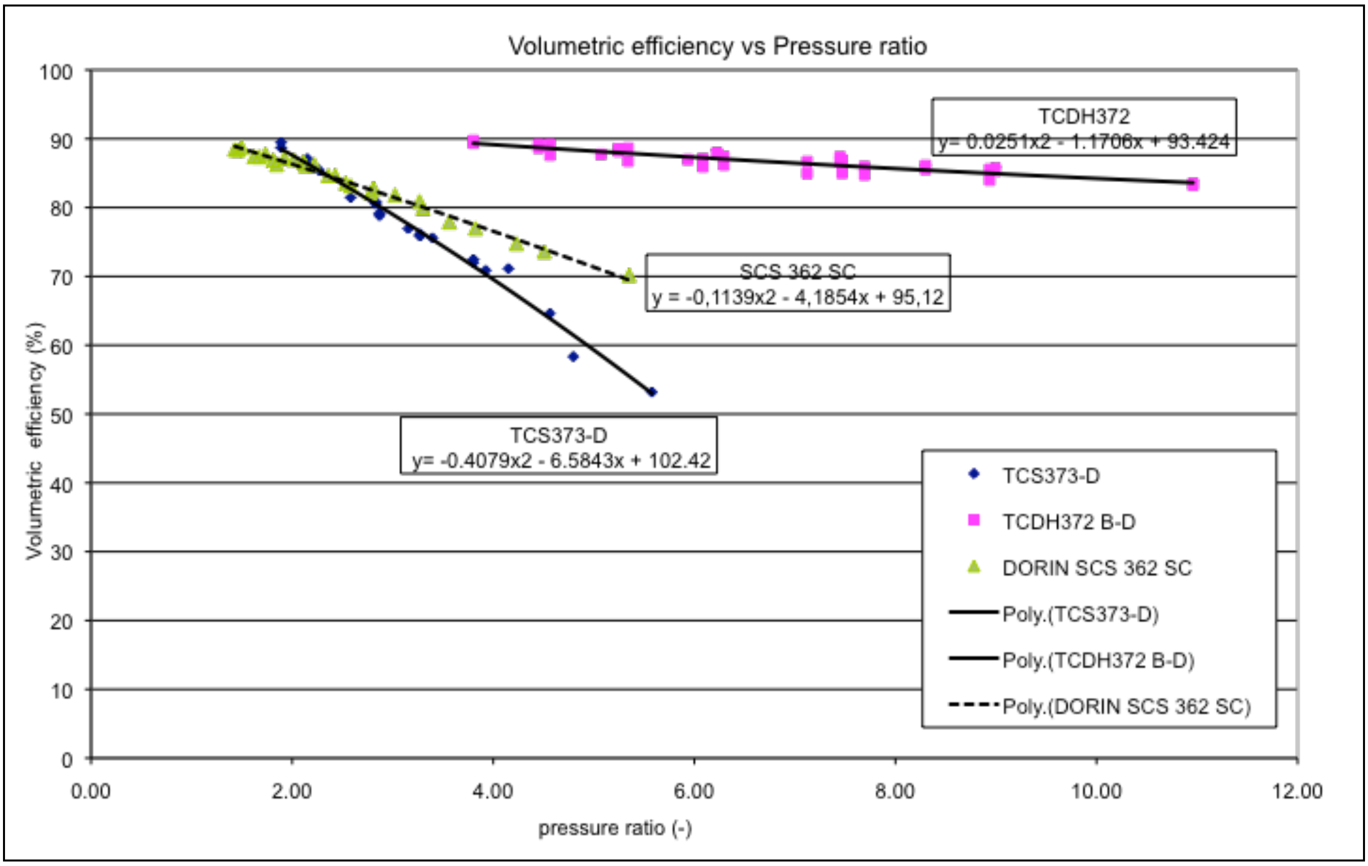
- KA-units in TR1
- KA3-unit in TR2
- High stage in KAFA-units in TR2

Dorin SCS 362:

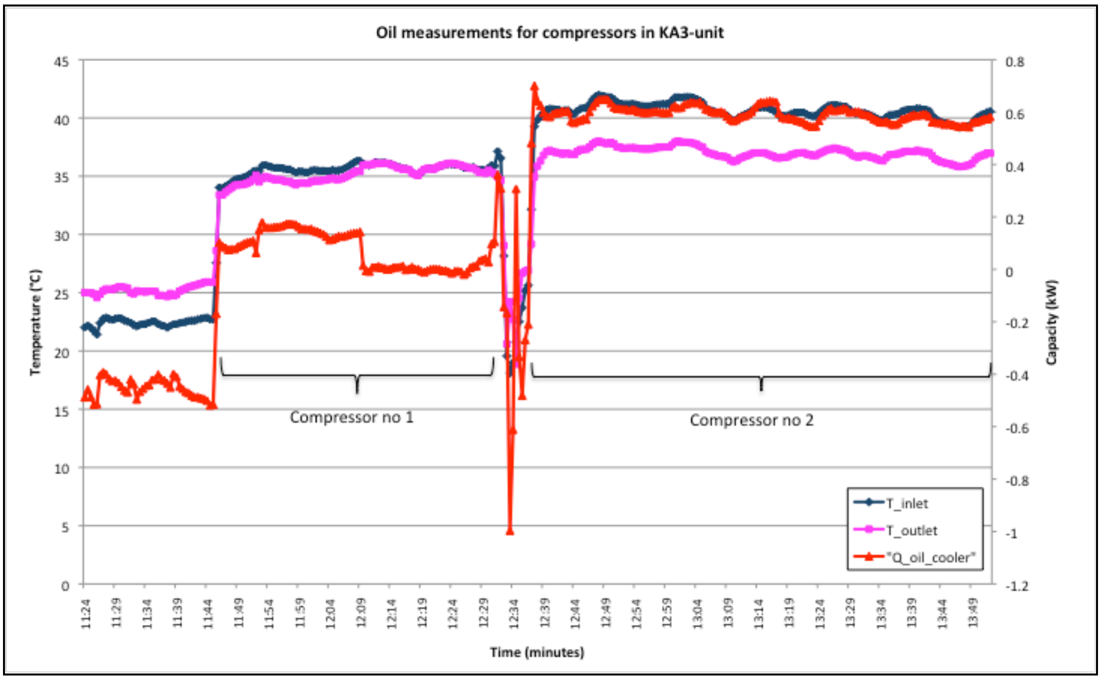
- Booster compressors in KAFA-units in TR2

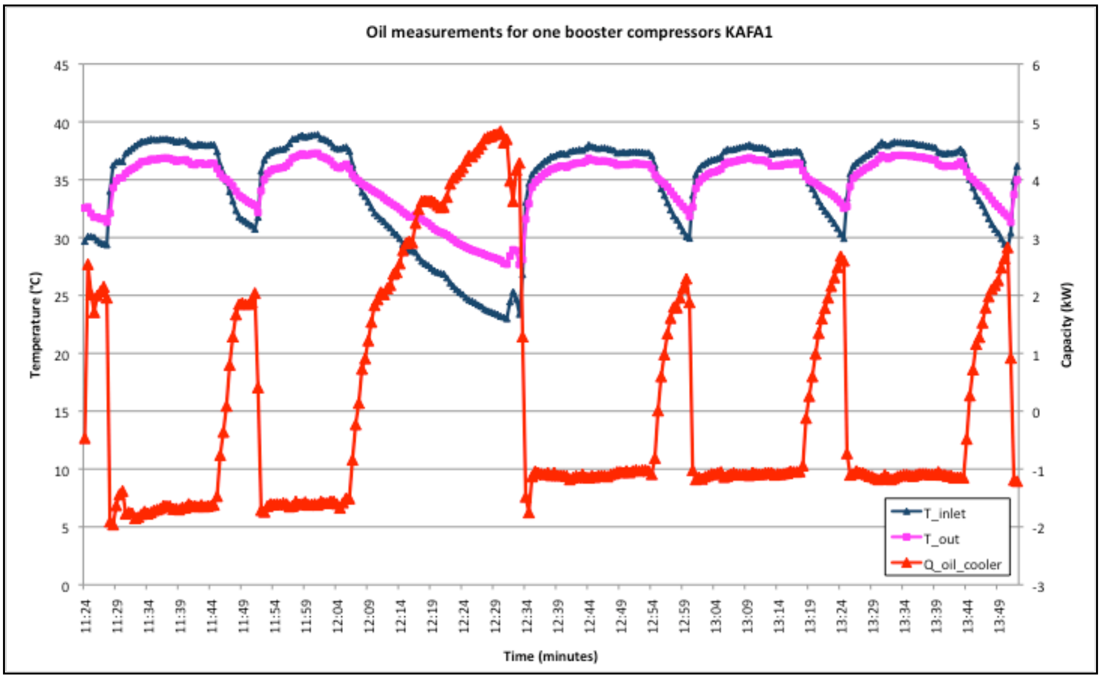
Dorin TCDH372:

- FA-units in TR1



Appendix 2: Oil temperature measurements





Appendix 3: Calculations of volumetric efficiency of compressors in CC 1

Procedure:

- Data from compressor manufacturer is given
- EES code to calculate vol. efficiency etc
- Graphs for curve fit

CO_2 compressor in cascade system 1

P_evap (Bar)	P_cond (Bar)	Pressure Ratio	Q (kW)	P (kW)	Q_evap (kW)	COP	m (kg/hr)	Disch. Temp ©	Eta_tot (%)	Eta_v (%)
10	33	3.3	4.86	2.07	4.52	2.18	62.0	131	52	70
	30	3	5.15	1.92	4.8	2.5	64.3	117	53	72
	27	2.7	5.62	1.68	5.25	3.13	67.6	97	57	76
12	33	2.8	6.32	2.12	5.87	2.77	80.1	108	55	76
	30	2.5	6.65	1.94	6.19	3.19	82.4	97	55	78
	27	2.3	7.19	1.65	6.71	4.07	85.9	79	59	82
14	33	2.4	7.99	2.10	7.40	3.52	100.7	91	58	82
	30	2.1	8.37	1.89	7.77	4.11	103.0	80	58	84
	27	1.9	8.98	1.56	8.36	5.36	106.6	64	63	87

ΔT total superheat = 30 K
ΔT usefull superheat = 10 K
ΔT subcooling = 5 K
Displacement at 1450 RPM= 4,06 m3/hr

Cells in this colour are input value to the EES model

R404A compressor in cascade system 1

P_evap (Bar)	P_cond (Bar)	Pressure Ratio	Q (kW)	P (kW)	Q_evap (kW)	COP	m (kg/hr)	Disch. Temp ©	Eta_tot (%)	Eta_v (%)
3	18	6.0	30.50	13.78	27.70	2.01	799.0	84	65	79
	16	5.3	33.00	13.23	30.20	2.28	824.0	78	65	81
	14	4.7	35.9	12.57	33	2.6253	850	71	65	84
	12	4.0	39.20	11.74	36.20	3.08	879.0	64	65	87
4	18	4.5	43.60	16.13	39.70	2.46	1104.0	80	64	83
	16	4.0	46.90	15.31	42.90	2.80	1131.0	74	64	85
	14	3.5	50.70	14.33	46.50	3.24	1159.0	68	63	87
	12	3.0	54.90	13.14	50.60	3.85	1192.0	61	62	89
5	18	3.6	57.20	17.92	52.00	2.90	1409.0	77	63	85
	16	3.2	61.30	16.83	56.00	3.33	1438.0	72	62	87
	14	2.8	65.80	15.56	60.40	3.88	1469.0	66	60	89
	12	2.4	71.00	14.03	65.50	4.67	1505.0	59	58	91

ΔT total superheat = 20 K
ΔT usefull superheat = 6 K
ΔT subcooling = 15 K
Displacement at 1450 RPM= 73,6 m3/hr

Cells in this colour are input value to the EES model

EES code (for R404A compressors)

$$dT_{sh;tot} = 20$$

$$dT_{sh;int} = 6$$

$$dT_{sc} = 15$$

$$P_{evap} = 5$$

$$P_{cond} = 18$$

$$T_{comp;out} = 77$$

$$\dot{V}_s = \frac{73{,}6}{3600}$$

$$\dot{Q}_{evap} = 52$$

$$T_{evap} = \mathbf{T}\left(\text{'R404A'} ; P = P_{evap} ; x = 1 \right)$$

$$T_{cond} = \mathbf{T}\left(\text{'R404A'} ; P = P_{cond} ; x = 0 \right)$$

$$T_{cond;out} = T_{cond} - dT_{sc}$$

$$T_{evap;out} = T_{evap} + dT_{sh;int}$$

$$T_{comp;in} = T_{evap} + dT_{sh;tot}$$

$$h_{evap;out} = \mathbf{h}\left(\text{'R404A'} ; T = T_{evap;out} ; P = P_{evap} \right)$$

$$h_{evap;in} = \mathbf{h}\left(\text{'R404A'} ; T = T_{cond;out} ; P = P_{cond} \right)$$

$$h_{comp;out;is} = \mathbf{h}\left(\text{'R404A'} ; s = s_{comp;in} ; P = P_{cond} \right)$$

$$h_{comp;in} = \mathbf{h}\left(\text{'R404A'} ; T = T_{comp;in} ; P = P_{evap} \right)$$

$$h_{comp;out} = \mathbf{h}\left(\text{'R404A'} ; T = T_{comp;out} ; P = P_{cond} \right)$$

$$s_{comp;in} = \mathbf{s}\left(\text{'R404A'} ; T = T_{comp;in} ; P = P_{evap} \right)$$

$$h_{tot} = \frac{h_{comp;out;is} - h_{comp;in}}{h_{comp;out} - h_{comp;in}}$$

$$\dot{Q}_{evap} = \dot{m}_r \cdot \left(h_{evap;out} - h_{evap;in} \right)$$

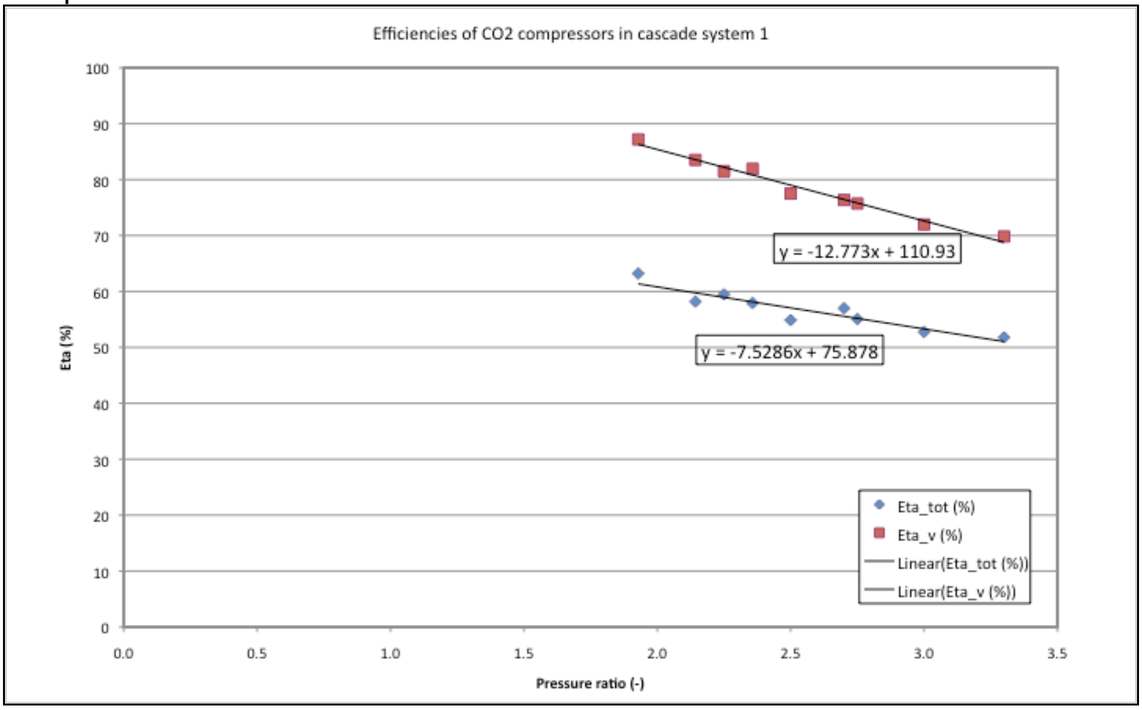
$$\dot{Q}_{comp} = \dot{m}_r \cdot \left(h_{comp;in} - h_{evap;in} \right)$$

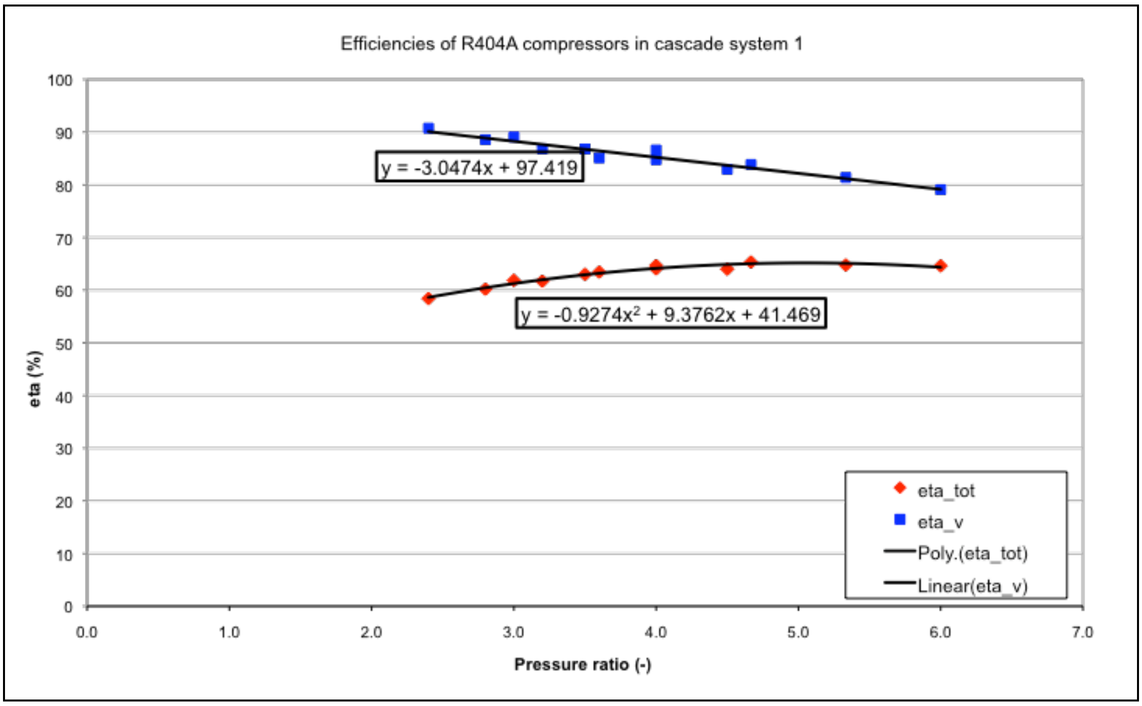
$$\dot{E}_{comp} = \dot{m}_r \cdot \left(h_{comp;out} - h_{comp;in} \right)$$

$$\dot{m}_r = h_v \cdot \dot{V}_s \cdot r_{comp;in}$$

$$r_{comp;in} = r\ (\text{'R404A'}\ ; T = T_{comp;in}\ ; P = P_{evap})$$

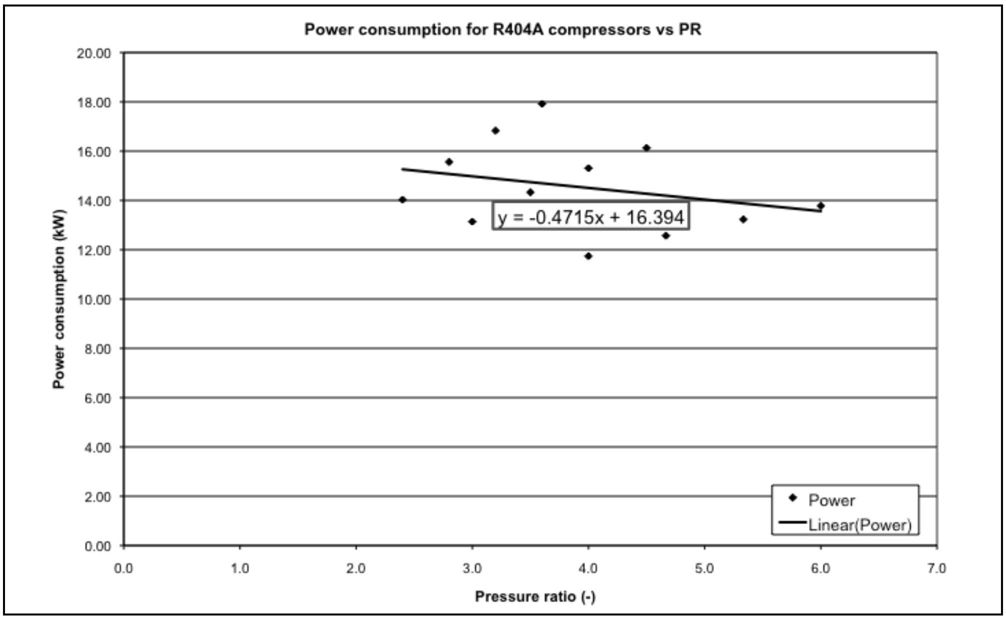
Curve fit for total efficiency and volumetric efficiency for both CO_2 and R404A compressors:

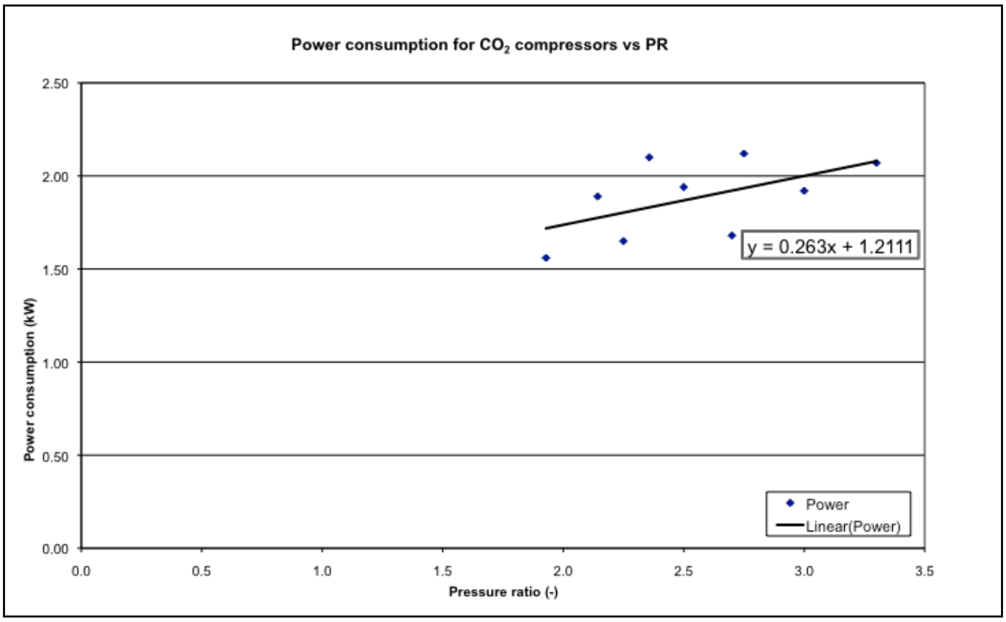




Appendix 4: Curve fit of power consumption vs PR in CC1

Curve fit for R404A compressors and CO_2 compressors from data tables in appendix 3.





Appendix 5: Calculations for the overall COP

T_amb, Q_m and Q_f are taken from the conditions for TR2. COP_m and COP_f are estimated from figure 7.8 and figure 7.9 at the specific conditions. TR1 is separated into two periods (January-July and August-February).

$$\dot{E}_x = \frac{\dot{Q}_x}{COP_x} \qquad [kW]$$

$$COP_{tot} = \frac{\dot{Q}_m + \dot{Q}_f}{\dot{E}_{tot\_m} + \dot{E}_{tot\_f}} \qquad [-]$$

Trans-critical system 1:

T_amb	Q_m	Q_f	COP m_1	COP m_2	COP f_1	COP f_2	E tot_1_m	E tot_2_m	E tot_1_f	E tot_2_f	COP tot_1	COP tot_2
12.87	47.06	26.91	3.14	2.90	1.36	1.47	14.99	16.23	19.76	18.27	2.13	2.14
10.35	48.29	27.30	3.35	3.19	1.36	1.43	14.41	15.14	20.09	19.10	2.19	2.21
5.81	47.30	26.75	3.43	3.66	1.40	1.48	13.78	12.92	19.13	18.04	2.25	2.39
4.42	49.32	31.35	3.42	3.70	1.42	1.52	14.44	13.33	22.15	20.63	2.20	2.38
3.43	53.57	33.61	3.40	3.80	1.43	1.55	15.76	14.10	23.56	21.72	2.22	2.43
1.76	51.89	33.69	3.38	3.90	1.44	1.60	15.37	13.30	23.33	21.11	2.21	2.49

Cascade system 1:

T_amb	Q_m	Q_f	COP_m	COP_f	E_tot_m	E__tot_f	COP_tot
4.42	49.32	31.35	3.18	1.29	15.51	24.30	2.03
3.43	53.57	33.61	3.20	1.31	16.74	25.66	2.06
1.76	51.89	33.69	3.18	1.29	16.32	26.12	2.02