Guidelines of how to instrument, measure and evaluate refrigeration systems in supermarkets

PAU GIMÉNEZ GAVARRELL

Master of Science Thesis
Stockholm, Sweden 2011

What is not measured does not exist;
What is measured, improves.

KTH Industrial Engineering and Management

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Abstract

This Master Thesis aims at establishing guidelines of how to instrument refrigeration systems with the objective that will be possible and easy to evaluate its efficiency (Coefficient of Performance).

Also we have put indicators of performance for systems that haven't got extensive instrumentation, how to calculate different system COP's and how to compare systems in field installations. It has been describe the different methods that can be used to calculate system performance.

The project includes the main solutions of supermarket refrigeration systems such as conventional systems, CO2 trans-critical and cascade system.

In previous project aimed at analyzing field measurements of supermarket refrigeration systems including 10 installations, it has been observed that the systems have extensive instrumentation; however, key parameters to perform proper performance analysis are missing. The main purposes of placing the instrumentation on the systems are usually for monitoring and control, but not to perform the system analysis. In some of this systems was necessary an extra instrumentation for the analysis. In other system was used some estimations due to the impossibility to install new measurements for reasons such as distant supermarkets or data acquisition software.

In some projects were studied and compared different cooling systems that had many similarities in terms of refrigerant used and the thermodynamic cycle followed, although it was installed in areas in Sweden with different climatic conditions, and hence was interesting to show the variations in efficiency parameters.

On the other hand has been also studied and compared systems completely different. In this case the key to the problem is to find relations and parameters which homogenize the systems owing to their differences.

In this Thesis is not intended to analyze all the facilities again, nevertheless have been chosen the most representative to base the study. It is considered appropriate to develop general guidelines for the main solutions.

For the analysis of a system, it is necessary to obtain a mathematic model that represents its behavior. The mathematical model is equivalent to a mathematical equation or set of them on the basis of which we know the behavior of a system. The mathematical model will be used to calculate the parameters needed for the system analysis; cooling capacity, COP low temperature and COP medium temperature form power consumption, temperature and pressure sensors.

Acknowledgements

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This thesis is especially dedicated to my parents, Paqui and Juan Ramón, my brothers JuanRa and Franc and my family, Ambrosio, Elena, Adrian, Amparin, Adri and Amparo, who supported me during my studies in all the decision I made.

It is also dedicated to my colleagues in Juvenalia: Enrique, Sara, Ainara and Isabel.

And finally I want to thank you, who followed me around Europe, visiting me several times during this six month, who helped me during the hard time and who makes life so nice. Thanks Sara.

Pau Giménez Gavarrell
Stockholm, September 2011

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Nomenclature

Roman

CC	Cascade refrigeration system
CO_2 or $CO2$	Carbone dioxide
COP	Coefficient of performance [-]
DH	District heating
DX	Direct expansion
$\dot{E}$	Electrical power [kW]
EES	Engineering Equation Solver
FA	Low temperature unit or cabinet
h	Enthalpy [kJ/kg]
$HVAC$	Heating, Ventilating, and Air Conditioning
IHE	Internal heat exchanger
KA	Medium temperature unit or cabinet
$KAFA$	Booster system with low and medium temperature
LR	Load ratio
LT	Low temperature
$\dot{m}$	Mass flow [kg/s]
MT	Medium temperature
n	Rotational speed [rpm]
P	Pressure [bar absolute]
PR	Pressure ratio [-]
$\dot{Q}_c$	Condensation capacity [kW]
$\dot{Q}_o$	Cooling capacity [kW]
SC	Subcooling
SH	Superheat [K]
SHP	Separate heat pump
T	Temperature [°C]
TR	Transcritical refrigeration system
$\dot{V}$	Volume flow [m^3/s]

Greek

Δ	Difference [-]
ρ	Density [kg/m^3]
η_{is}	Isentropic efficiency [-]
η_{v}	Volumetric efficiency [-]
η_{tot}	Total efficiency [-]
v	Specific volume [m^3/kg]

Subscript

abs	Absolute
air	For air
amb	Ambient
app	Approach temperature difference
$booster$	Booster system
$brine$	Brine
cab	Cabinet medium temperature
$chiller$	Chiller
$comp$	Compressor
$cond$	Condenser
el	Electric
$evap$	Evaporation
$freezer$	Freezer
gc	Gas cooler
HE	Heat Exchanger
in	Inlet
$inst$	Instantaneous
is	Isentropic
$losses$	Heat losses
LR	Load ratio

map	Map or design conditions
$medium\_temp$	Medium temperature
new	New or running conditions
out	Outlet
$oil\,cooler$	Oil cooler losses
$pumps$	Pumps
s	Swept
sat	Saturation
$subcool$	Subcool
tot	Total

1. Introduction

Global warming is a worldwide challenge nowadays and industries are following the international and national guidelines to tackle the problem. The industries are being more concerned of wherefrom their energy is coming from and how they can contribute to a more sustainable interaction with the environment.

To keep food products cold or frozen is essential in today's lifestyle. Related to the global warming impact caused by the refrigeration industry, supermarkets are main contributors by two ways:

- Directly contributing by leaking refrigerants
- Indirectly with their high energy consumption, mainly due to the large use of energy to run the refrigeration systems.

In the future, due to the population growth these two effects will increase and will have a negative impact on the environment. From the following Figure it is obvious that for a typical medium size Swedish supermarket the main part of consumption comes from the refrigeration system, representing about 50% of the energy consumption.

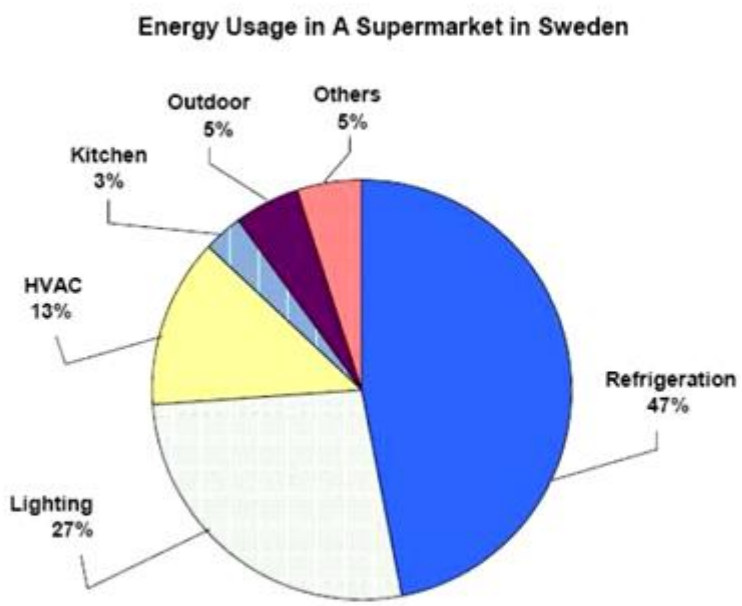

Figure 1: Energy use in a typical medium size Swedish supermarket (1)

This implies an important reason why it is interesting to know the efficiency of the system. All existing supermarkets have different types of food that need different temperatures for its conservation. These products are generally chilled or frozen and present different range of temperature, +3°C and -18°C respectively, depending on the durability of products. These foods are kept in fridges surrounded by air. The air has a temperature between 23°C and 25°C, which is the comfort temperature for the consumers. As a result, there is a net heat flow incoming in the coolers that has to be removed.

On the other hand supermarkets also have heating and cooling needs due to weather conditions and the season. In Sweden the most important part is how to satisfy the heat need. These needs have the highest values in relatively cold climates. The supply temperature to the brine to the heating system, when the heat recovery is used in the refrigeration system, is about 35°C. In order to satisfy them there are different solutions that can be divided in two groups:

- Independent heat supply systems: in this case, the heating needs in the supermarket are covered by district heating (DH) or a separate heat pump system (SHP) which operates independent of the cooling system.

- Systems coupled to the cooling system: These systems aim to increase the overall COP of the complete system by means the use of energy transfer in the condenser of the refrigeration system. These systems increase the discharge pressure of the cooling system's compressor to raise the thermal level of heat (35°C) and use the heat rejected in the condenser as heat source. If the fluid used is CO_2 the thermal level of the compressor outlet is high. Another most common solution is to attach a heat pump to the condenser cooling system, which plays the role of cold source in the heat pump.

This is the function of the refrigeration systems, but among all possible systems, refrigerants and cycles, it has to be chosen to best suit the supermarket requirements. There are many input parameters for the choice of a system, such as weather and location considerations, relationship between cooling capacity for fresh food (3°C) and the cooling capacity for frozen products(-18°C), the need for heat recovery, etc, and then the design is adjusted.

The Department of Energy Technology (KTH) is conducting for several years many different analyses in real Sweden supermarkets. It is obtained the efficiency parameters of each installation, and compared the results with the aim of establishing a classification system to ensure maximum efficiency in future installations.

2. Objectives

The main objective of this master's thesis is to propose general guidelines of how to instrument refrigeration systems with the objective that will be possible and easy to evaluate its performance.

This study describes the different methods that can be used to calculate system performance and the problems found. Different assumptions have been questioned and some modifications have been proposed in the Excel Template for new analysis.

The project includes the main solutions of supermarket refrigeration systems such as conventional, CO2 trans-critical and cascade solutions.

This study also aims at providing some equations in order to know the COP of the installation based on temperature, pressure and electric power consumption, without the necessity of obtaining the thermodynamic properties of the refrigerant.

2.1. Specific Objectives

The specific objectives for the thesis are:

- Analyze the different steps for a new supermarket study
    - Collecting data for a period
    - Creating calculation templates
    - Data processing (normalizing, filtering)
    - Calculating main parameters (cooling capacity and COP)

- Revision of the equations and methods used, solving the problems found in earlier analysis.

- Question the assumptions and results.

- Propose new solutions for implement the calculations.

- Show some recommendations for the analysis of new systems.

3. Parameters and instrumentation

The steps to assess the energy performance of cooling systems start with the necessary information about a system. The refrigeration systems are intensive energy consumers; therefore it is necessary to establish good control of its operation.

All the refrigeration systems consist of one or more closed circuits. Inside is the refrigerant fluid undergoing different thermodynamic transformations in each part of the circuit. The first two important parameters that we need to evaluate the performance of the systems is the useful effect and the involved cost, that is the cooling capacity and the electric power consumption. Using the ratio between these two parameters is calculated the Coefficient of performance (COP) of the system giving an idea about the efficiency and it can be used as a first comparison parameter between systems. In order to compare different cooling systems this is the most used reference parameter, expressed as:

$$COP_{inst.} = \frac{\dot{Q}_o}{\dot{E}_{comp}} = \frac{Cooling \quad capacity}{Electrical \;\; power \; consumption}$$

Equation 1

The COP can be calculated using instantaneous power, or evaluating the energy consumed during each month and gets a monthly average of the efficiency of the system.

The following Figure shows a schematic of the simplest compression refrigeration circuit.

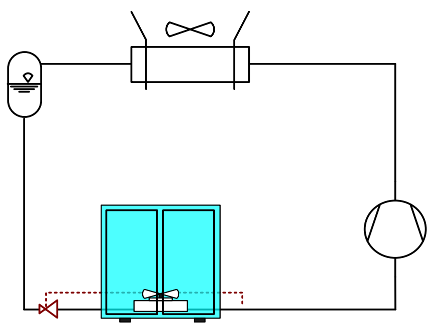

Figure 2: Simple schematic of a basic refrigeration cycle

This system consists of one compressor, keeping two pressures levels. In the evaporator is maintained the low pressure corresponding to the evaporation temperature to achieve the desired cabinet temperature. The pressure of the condenser is pressure needed to reject the heat to the ambient.

In the COP equation each term has to be evaluated. The cooling capacity for the circuit from the refrigerant side is calculate by:

$$\dot{Q}_o = \dot{m} \cdot \Delta h_o$$

Equation 2

The first parameter that it is needed for the COP's evaluation is the mass flow in each closed loop. This parameter can be measured by using a mass flow meter or estimate by other methods. Δh_o is the enthalpy difference over the evaporator. For its evaluation we need to know the state of the refrigerant before and after the evaporator and to install the appropriate instrumentation. Knowing that the state of the refrigerant before the expansion valve is liquid sub-cooled and before the compressor is vapor superheated, for determinate the enthalpy it is necessary to know the pressure and temperature.

“In order to measure the performance of the circuit, it is necessary to monitor the following: mass flow, suction pressure, discharge pressure, superheat (internal and external), sub-cooling and power input.” (2) With the above points we can evaluate the cooling capacity and the COP. In the following Figure the indicated instrumentation has been represented. It is possible to see two pressure and temperature sensors to evaluate the enthalpy in the evaporator inlet and outlet and a flow meter. It will be needed a power meter to measure the electric power absorbed by the compressor. The temperature sensors inside a rectangle are not necessary if it is installed a flow meter. If it is not installed a flow meter theses sensors are necessary to estimate the mass flow using different methods.

Mass flow
Suction pressure
Discharge pressure
Internal superheat
External superheat
Sub-cooling
Power input

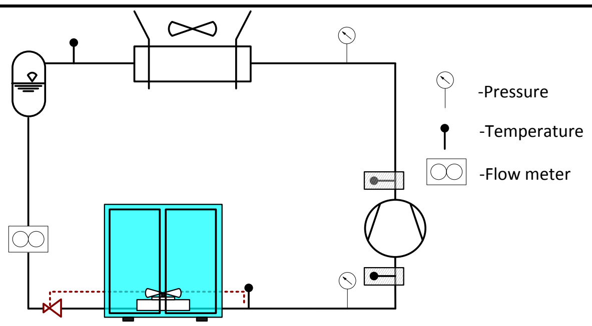

Figure 3: Simple schematic of a basic refrigeration cycle with the instrumentation

The process on working with one supermarket, one month is the next:

1. **Start with determining what data you have and what data you need. If you have all the needed data you can start retrieving it, if not - think how estimate\predict the data you need.**

2. **Collect all the data, check if it is normalized, if no data is missed.**

3. **Calculate the template (Excel and RefProp software)**

4. **Filter the data.**

5. **Check if the results calculated are feasible for given supermarket.**

The important parameters for the supermarket analysis are:

- Temperatures
- Pressures
- Electric power consumption
- Data acquisition software
- Normalization of the data
- Filtering the data

4. Refrigeration System Solutions

In general, two temperature levels are required in supermarkets for chilled and frozen products. Product temperatures of around +3°C and -18°C are commonly maintained. In these applications, with a large difference between evaporating and condensing temperatures, the cascade or other two-stage systems become favorable and are adaptable for the two-temperature level requirements of the supermarket. The following sub-sections describe four refrigeration systems in which we will base our analysis: conventional and the main CO2-based solutions in supermarkets.

The steps that will be followed in the next part are:

- Definition of each system.
- Definition of the Total COP, freezer COP, chiller COP and the cooling capacities. These are the needed parameters to perform the refrigeration system analysis.
- Define the input parameters for the mathematical model. Suggest the instrumentations needed.
- How should the date be handled: normalizing the data intervals if necessary, filtering and averaging.
- How to compare different system solutions

<table>
<tr>
<th>Conventional R404A system</th>
<th colspan="2">CO2 trans-critical systems</th>
<th>Cascade systems with CO2</th>
</tr>
<tr>
<td rowspan="2">The R404A system consists of two separate circuits; At the low temperature unit it uses direct expansion (DX) with brine at the medium temperature level. A heat exchanger is connecting the medium and the low temperature stages of the system to further sub-cool the liquid coming out of the low temperature condenser or sub-cooler.

It is used a single-stage compression with R404A due to the steepness of the isentropic compression lines for R404A two-stage compression with inter-cooling has very little influence on improving the COP of the medium and low temperature levels.

The following figure is a simple schematic of the system.</td>
<td colspan="2">Due to the widespread interest in CO2 as an alternative to synthetic refrigerants in refrigeration systems, the components have been redesigned and have gone down in price to reach available and competitive systems. This made it possible to build CO2 trans-critical systems for supermarkets. The main two arrangements applied in Swedish supermarkets are:
• Booster
• Parallel arrangement</td>
<td rowspan="2">Cascade systems with CO_2 in the low-temperature stage have been applied in several supermarket installations in Sweden.

• R404A-CO2 cascade
In this system arrangement, which exists in several installations in Sweden, the refrigerant in the high-temperature stage is R404A. The medium temperature circuit uses a conventional single phase secondary working fluids. CO_2 is the working fluid in the low-temperature circuit where it rejects the heat to the brine at the medium temperature level. The following plot is a simple schematic of such system.</td>
</tr>
<tr>
<td>Booster system

In this system solution the refrigerant is expanded in two different pressure/temperature levels, medium and low. The low stage compressor (booster) rejects the discharge gas into the suction line of the high stage compressor mixing with the superheated return vapor from the medium temperature level.</td>
<td>Parallel arrangement

In this system two separate parallel CO_2 circuits operating between the ambient temperature on the high side and the intermediate and freezing temperature levels on the other sides. In order to obtain reasonable efficiency, the CO_2 circuit that operates between ambient and freezer temperatures should have two-stage compression with an intercooler. The following figure is a simple schematic of the parallel system solution.</td>
</tr>
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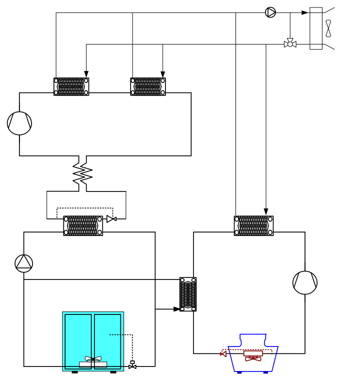


Figure 4: Schematic diagram of a conventional R404A system.

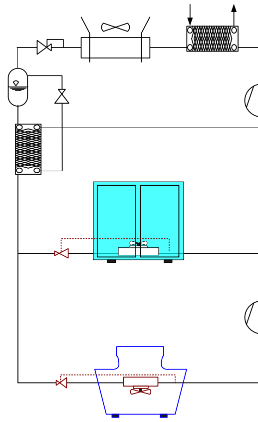


Figure 6: Schematic diagram of CO2 booster system solution with low pressure receiver.

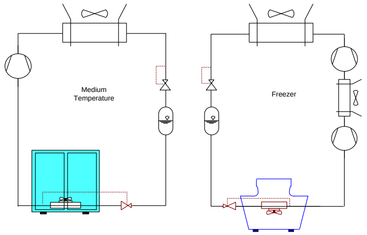


Figure 7: Schematic diagram of CO2 parallel system solution with two-stage compression on the low temperature level.

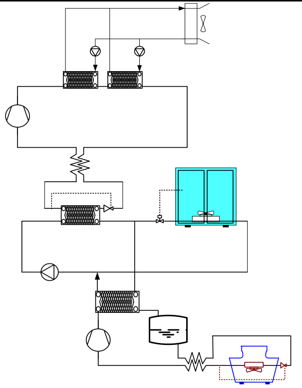


Figure 5: Schematic diagram of R404A-CO2 cascade system with brine at the medium temperature level.

4.1. Conventional R404A system

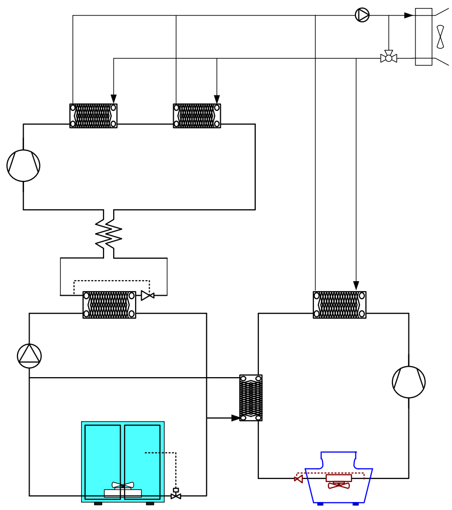


Figure 4: Schematic diagram of a conventional R404A system.

The previous system is described as RS1 in the Manickam Louis Tamilarasan's Master Thesis (3). Consist of a medium temperature stage and a low temperature stage. The secondary circuits on the evaporator and condenser side are connected to a single propylene glycol circuit. The system has been chosen for the study because it is a conventional solution used in supermarkets in Sweden and presents the basic four loops that we can find in refrigeration systems with the necessary connection between them to increase the efficiency with respect to the use of separate systems.

4.1.1.Medium and low temperature units

The low temperature stage use R404A as the refrigerant. The sub-cooler is installed after the condenser. The sub-cooler is a heat exchanger connected to the brine loop. The supply temperatures of the brine are about -8°C. It reduces the enthalpy before the expansion valve. This reduction increases the enthalpy difference between the inlet and outlet in the evaporator, considered isenthalpic expansion valve. This power is transferred to the brine loop, and will be extracted by the chiller loop, which has a higher COP, and for these reason, the extra cost of the sub-cooler in the freezer loop is justified. (3)

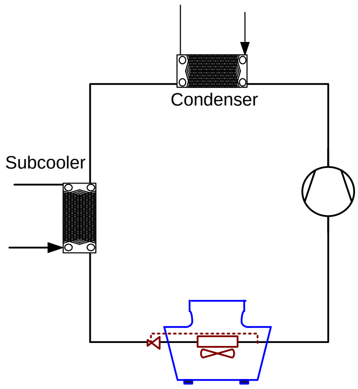


Figure 8: Low temperature unit, freezer

In the freezers, carbon dioxide can be used as brine in a secondary circuit. In this case, the CO2 in the secondary circuit is stored in a large tank in liquid form. CO2 is preferably where the recommended maximum working pressure is about 40 bars, which is said to be higher than normal in a refrigeration system with conventional components. In such indirect system with CO2 for freezer applications the pressure level is about 12 bars. Currently in Sweden there are more than 100 installations with such solution.

The medium temperature stage use R404A as the refrigerant. This stage constitutes of a sub-cooler which is located after the condenser for the purpose of sub-cooling the liquid out of the condenser. An electronic expansion valve is used on this medium temperature stage. Furthermore, an internal heat exchanger is connected to further sub-cool the liquid coming out of the sub-cooler and super-heat in the compressor inlet.

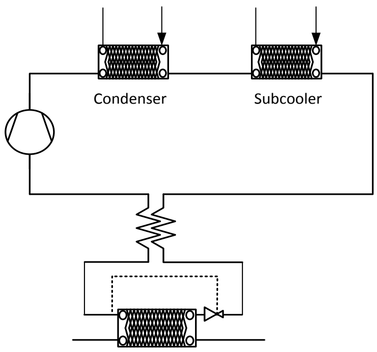


Figure 9: Medium temperature unit, chiller

4.1.2.COP medium and low temperature

The sub-cooling of the liquid has an energy cost and it is considered in the evaluation of the COP. For these reason it is added the energy consumption for the freezer compressor and the electrical cost of sub-cooling when the COP is defined.

$$COP_{freezer} = \frac{\dot{Q}_{o\_freezer}}{\dot{E}_{comp\_freezer} + \dot{Q}_{o\_subcool} / COP_{chiller}}$$ **Equation 3**

This electric cost is introduced because separate systems for each thermal level are not used. There is a coupling between systems and hence the energy used by the chiller compressor to dissipate the $\dot{Q}_{o\_subcool}$ in the chiller condenser should be taken into account in the corresponding COP. Therefore we must evaluate the following terms:

$$\dot{Q}_{o\_freezer}, \dot{E}_{comp\_freezer}, \dot{Q}_{o\_subcool}$$

The cooling capacity of the freezers is using the equation

$$\dot{Q}_{o\_freezer} = \dot{m}_{freezer} \cdot \Delta h_{freezer} = \dot{m}_{freezer} \cdot (h_{out\_cab\_freezer} - h_{in\_cab\_freezer}) \qquad \textbf{Equation 4}$$

This expression is valid for all freezers in each refrigeration system. If there is direct expansion in the freezer, the refrigerant state before the expansion valve is sub-cooled liquid. Considering isenthalpic expansion valve, $h_{in\_cab\_freezer}$ is obtained using a pressure transducer and a thermometer before the expansion valve. On the other hand the state of the cabinet's exit $h_{out\_cab\_freezer}$ is superheated vapor. It's value is obtained with a thermometer and pressure transducer, using the refrigerants thermodynamic properties. It is also needed to evaluate the mass flow through the freezer circuit $\dot{m}_{freezer}$. The easy way is by installing a mass flow meter. It can be coriolis flow meter, gas flow meter or liquid flow meter. The parameters and the instrumentation are listed in the table below.

Parameters	Instrumentation
$\dot{m}_{freezer}$	Mass flow meter
$h_{out\_cab\_freezer}$	Pressure, Temperature
$h_{in\_cab\_freezer}$	Pressure, Temperature

Table 1: Parameters to evaluate the cooling capacity in the low temperature cabinets

For calculate sub-cooling from the medium temperature unit, using the measurements to evaluate $\dot{Q}_{o\_cab\_freezer}$, it is only necessary $h_{in\_HE\_subcool}$.

$$\dot{Q}_{o\_subcool} = \dot{m}_{freezer} \cdot (h_{out\_HE\_subcool} - h_{in\_HE\_subcool}) \qquad \textbf{Equation 5}$$

The state of the refrigerant is slightly sub-cooled liquid, and for this reason we need its temperature, considering constant pressure in the heat exchanger. The parameters and the instrumentation for the cooling capacity in the sub-cooler are listed in the following table.

Parameters	Instrumentation
$\dot{m}_{freezer}$	Mass flow meter (measured)
$h_{out\_HE\_subcool} = h_{in\_cab\_freezer}$	Pressure, Temperature (measured)
$h_{in\_HE\_subcool}$	Pressure, Temperature

Table 2: Parameters to evaluate the sub-cooling in the low temperature unit

Finally the $\dot{E}_{comp\_freezer}$ is measured using a power meter. Once assessed each term, and with the chiller COP which subsequently we calculate, it is completely assessed the $COP_{freezer}$ and the cooling capacity $\dot{Q}_{o\_freezer}$.

We must evaluate the following terms $\dot{Q}_{o\_chiller}$, $\dot{E}_{comp\_chiller}$, $\dot{E}_{pump\_brine}$ for the COP calculation in the medium temperature. Being an indirect system, the useful cooling capacity is not in the medium temperature refrigerant loop, but in the brine loop. The COP for the chiller is calculated with the follow equation:

$$COP_{chiller} = \frac{\dot{Q}_{o\_chiller}}{\dot{E}_{comp\_chiller} + \dot{E}_{pump\_brine}} \qquad \textbf{Equation 6}$$

The power consumption by the compressor and the brine pump can be measured with a power meter. On the other hand $\dot{Q}_{o\_chiller}$ is evaluated as following:

$$\dot{Q}_{o\_chiller} = \dot{m}_{chiller} \cdot \Delta h_{chiller} = \dot{m}_{chiller} \cdot (h_{out\_evap\_chiller} - h_{in\_evap\_chiller}) \qquad \textbf{Equation 7}$$

Parameters	Instrumentation
$\dot{m}_{chiller}$	Mass flow meter
$h_{out\_evap\_chiller}$	Pressure, Temperature
$h_{in\_evap\_chiller}$	Pressure, Temperature

Table 3: Parameters to evaluate the cooling capacity in the medium temperature unit

Although the energy consumption by the pumps in the brine loop should be divided between each COP, is considered only in the chiller COP because this energy consumption is low in comparison with the energy consumption in the compressors. On the other hand, $\dot{Q}_{o\_subcool}$ is smaller than $\dot{Q}_{cab\_chiller}$ and for this reason the energy cost of the brine pumps is considered only in the $COP_{chiller}$.

4.1.3.Brine loop

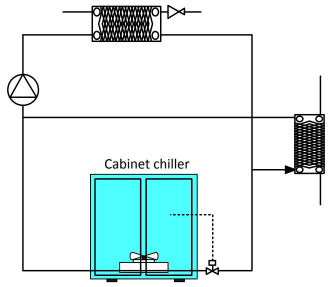


Figure 10: Brine loop

The secondary refrigerant is propylene glycol, although can be used ethylene glycol and other fluids. We could place directly the evaporator inside the cabinets, and extract a refrigerant line, with a by-pass to the evaporator, and connect expanding first, with the freezer heat exchanger. But nowadays the main worry is to reduce the amount of refrigerant as much as possible due to the known effects about climate change; this is why the indirect brine loop is used. From this point of view is preferred a slight reduction in system efficiency due to the need to maintain lower evaporator temperature in the chiller loop than the temperature in systems without brine loop. The electric power from the pump is directly load to the chiller COP for reasons discussed above.

The energy balance in this loop is:

$$\dot{Q}_{o\_chiller} = \dot{Q}_{cab\_chiller} + \dot{Q}_{o\_subcool} + \dot{E}_{pump\_brine} \qquad \textbf{Equation 8}$$

4.1.4.Coolant loop

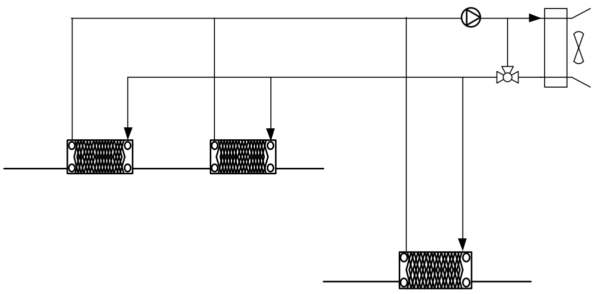


Figure 11: Coolant loop

For the same reason that is used the brine loop despite the efficiency reduction, in order to use less refrigerant, in case of the coolant loop there is also another reason, the heat recovery for the HVAC (heat ventilation and air conditioner). The renovation air for the supermarket can be preheated with the heat rejection from the cooling system.

The above equation did not take into account the electric power by the fans and the pump in the coolant loop. This electric cost should be divided in the two COPs in proportion to the cooling capacity of each unit: $\dot{Q}_{o\_freezer}, \dot{Q}_{cab\_chiller}$

4.1.5.Total COP

The total COP will be calculated with the following equation:

$$COP_{tot} = \frac{\dot{Q}_{o\_freezer} + \dot{Q}_{cab\_chiller}}{\dot{E}_{comp\_freezer} + \dot{E}_{comp\_chiller} + \dot{E}_{pump\_brine}} \qquad \textbf{Equation 9}$$

Moreover, in the same Thesis named before (3) RS2 and RS3 are presented, with similar structures but different variations in the number of units for freezer and chilled products and the refrigerant used. While in RS2 the primary refrigerant is R404A in the freezers, R407C in the chillers and ethylene glycol as secondary refrigerant, in RS3 the primary refrigerant is R404A in the freezers, R407C in one of the chillers and R404A in the other chillers, the secondary refrigerant is propylene glycol. The systems RS3 have in addition a heat pump connected to the coolant loop, to recover part of the heat rejected in winter period.

The diagram below shows the needed instrumentation to assess the specified parameters. It has been introduced other temperature sensors (inside rectangles) needed to estimate the mass flow if it is not installed mass flow meters..

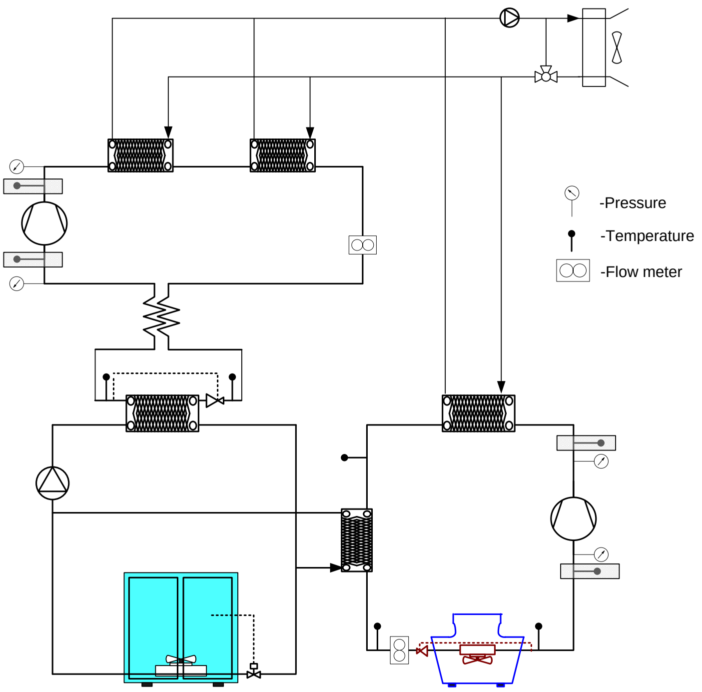


Figure 12: Instrumentation in Conventional R404A system

4.2. CO2 trans-critical systems: Booster system

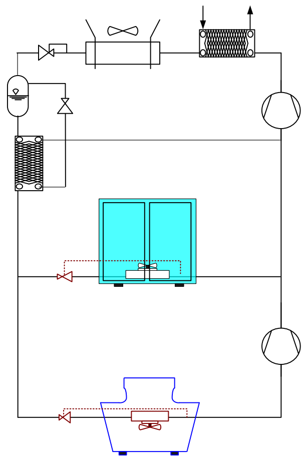


Figure 6: Schematic diagram of CO2 booster system solution with low pressure receiver.

In the supermarket refrigeration systems, as has been mentioned, there are typically two types of refrigeration circuits: medium temperature circuits for the chilled food display cabinets and cold rooms, and low temperature circuits for the frozen food. In the case of booster systems these two temperature circuits are integrated in a circuit with only one condenser, and mixing the state in the intermediate pressure reducing the overheating in the output of the low temperature compressor.

One advantage of the CO2 booster system is that the medium and low temperature levels can be served by one unit and then a single control system package can be used which cuts the cost of the system. Other advantage is the use of a single refrigerant at medium and low temperatures. Because of its thermodynamic properties is more appropriate system for cold climate in Northern countries such as Sweden, Denmark and Germany, due to in a hot climate the annual consumption of CO2 systems is higher than conventional R404A systems (4). One of the reasons is the high heat sink temperature, so the system works in trans-critical regime.

This system consists of two compressors groups, keeping fourth pressures and temperature levels. This fourth pressure level is achieved by expanding to an intermediate pressure between the saturated pressure in the condenser and the saturated pressure in the medium temperature evaporator. The discharge of low pressure compressor is connected to the suction line of high pressure compressor and in the outline of the chiller cabinet, getting with the mix to reduce the superheat in the suction line of the high pressure compressor.

"The receiver vessel has two outlets, one at the bottom for the liquid that is to be introduced to the cabinets and evaporated, and one at the top for vapor extraction to a gas bypass circuit. This vapor is expanded first in two parallel expansion valves to reduce its pressure and temperature. Second, the resulting vapor-liquid mixture in this line and the saturated liquid from the bottom of the receiver tank enter a counter flow heat exchanger where the liquid is sub-cooled and the vapor-liquid mixture is heated and returned to the suction side of the high stage compressors.

After the heat exchanger, the liquid refrigerant flow is divided in two parts, one leading to the medium temperature cabinets and one to the freezers. There, evaporation takes place after the refrigerant has passed through expansion valves. The refrigerant from the freezers is returned to the low stage compressors and mixes with the flow from the chillers at the compressor discharge. It also mixes with the flow from the receiver before being compressed to a heat sink level. " (5)

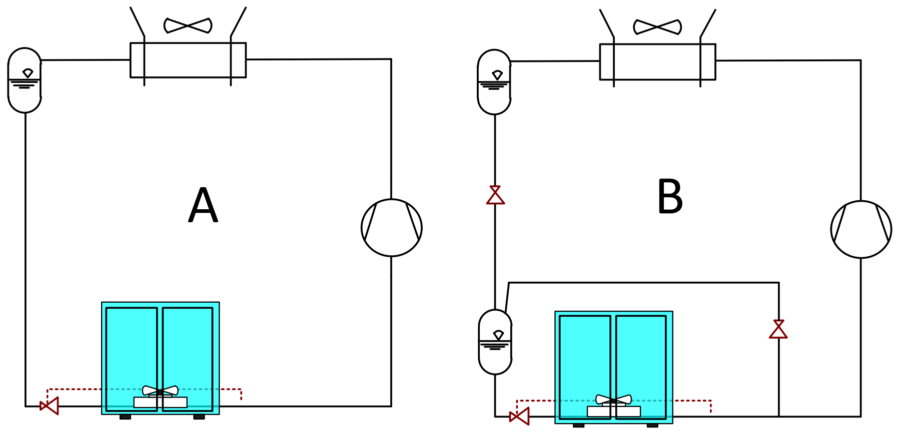


Figure 13: Simple schematic of a basic refrigeration cycle (A) with the introduction of a receiver vessel and vapor by-passl (B)

In the previous image is showed the basic refrigeration system A and it has been installed a bypass for the gas, system B. The system has the same COP, but in a booster system this modification improves the total COP.

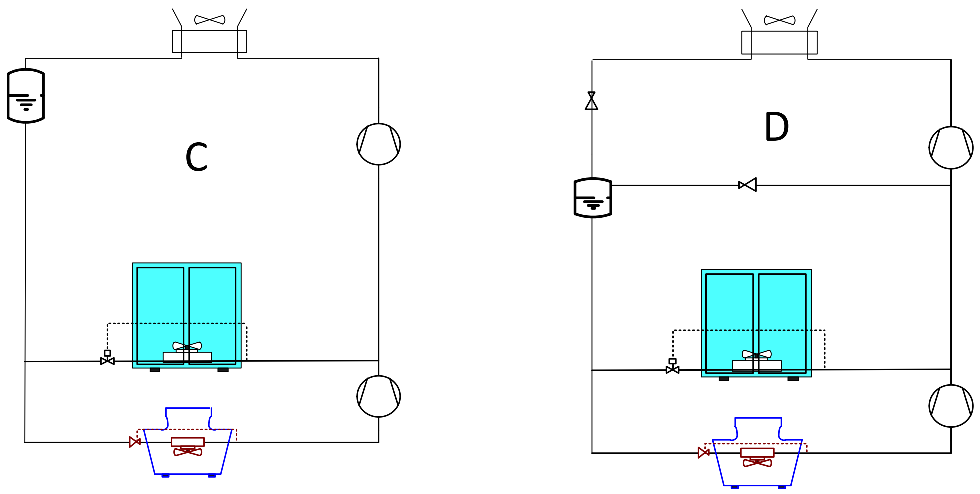


Figure 14: Booster system (C), and booster system with receiver vessel and vapor by-pass (D)

In the previous image we can see the booster system C, and next to this system is showed D with the same modification that the case B. This change is made following the idea to move to the left as much as possible the enthalpy to the liquid state. This idea comes from the objective to reduce the mass flow throw the freezer compressor. In order to reduce this mass flow, and consequently the power consumption in this part of the system we are interested in increase the enthalpy difference throws the evaporator. The higher enthalpy increase is achieved until the saturated pressure for chiller evaporator temperature (-10°C) but the differential pressure in the bypass and the chiller cabinet limits to work with the minimum mass flow due to it simplifies the control of the system.

The installation of a heat exchanger intended to recover the losses caused by this intermediate temperature. "There are two main reasons for using a heat exchanger. First, when expanding the saturated vapor from the receiver to the suction side of the high stage compressors, the result will be a liquid-vapor mixture and there is a risk of liquid droplets entering the compressor causing harmful cavitations. Since this mixture will be mixed with the relatively hot discharge gas from the low pressure compressors, the risk of this happening is probably small but the use of the heat exchanger reduces it further. Due to the slope of the saturated vapor line in the p-h diagram, the lower the receiver pressure is, the higher the vapor quality in the gas bypass will be, which reduces the risk of liquid entering the compressors and increases the COP (6). Second, sub-cooling of the liquid from the receiver reduces the vapor quality at the inlet of the evaporators. This means that a larger region of the evaporators will be filled with liquid which improves the heat transfer. The liquid sub cooling in the heat exchanger turned out to be very small, on average about 1K or even less." (5)

To sum up, all the improvements are oriented to increase the system COP without compromised the safety of the compressor, and without increase the regulation and control cost.

For a booster system we can evaluate the COP total, the COP freezer and the COP chiller. Firstly we can distinguish four different mass flows going through the chiller, the freezer, the condenser, and the line that connect heat exchange with the high pressure compressor. One mass flow is the total mass flow going through the high stage compressor and the condenser. A mass flow balance can be applied in the system, but with one equation and four unknown, we need three more mass flows.

$$\dot{m}_{condenser} = \dot{m}_{chiller} + \dot{m}_{freezer} + \dot{m}_{HE}$$ **Equation 10**

In the booster system we have taken to base our analysis we can distinguish 4 pressure levels and mass flow rates. This mass flow, the pressure and temperature measurements allow calculating the power of each part of the system. For the evaluation of the COP only the capacity in the freezers and chillers is taken into account. On the other hand, as the exit state of the cabinets is overheated vapor, we need again a pressure and temperature sensor for each line to determine the output enthalpy.

The system is designed for a cooling capacity in chiller and freezer cabinets but once the systems is in operation the mass flow rate is determinate by the cooling capacity needed in the cabinets. Therefore, using the mass flow balance, and with the measurement of three mass flow, we can determine the refrigerant flow in the different lines of the circuit. It is used an energy balance in the heat exchanger for calculated the mass flow. Whit two equations and four unknowing variables it is necessary only two mass flow.

The cooling capacity in the freezer and chiller can be calculated by the following equations:

$$\dot{Q}_{o\_freezer} = \dot{m}_{freezer} \cdot \Delta h_{freezer} = \dot{m}_{freezer} \cdot (h_{out\_cab\_freezer} - h_{in\_cab\_freezer})$$ **Equation 11**

$$\dot{Q}_{o\_chiller} = \dot{m}_{chiller} \cdot \Delta h_{chiller} = \dot{m}_{chiller} \cdot (h_{out\_cab\_chiller} - h_{in\_cab\_chiller})$$ **Equation 12**

The instrumentation to evaluate the terms of the above equations is the same indicate in the Table 1 for low temperature unit and Table 3 for medium temperature unit.

4.2.1.COP medium and low temperature

The compressor power consumption can be evaluated with a wattmeter directly. But to determinate each COP we need $\dot{E}_{comp\_chiller\_for\_freezer}$. We need to distribute the power consumption by the high pressure compressor due to its function is to reject the heat absorbed in the medium and low temperature cabinets. For these reason we need to know which part of the electric power is absorbed by the low pressure compressor really gets to the refrigerant because this power should be rejected by the condenser.

$$COP_{freezer} = \frac{\dot{Q}_{o\_freezer}}{\dot{E}_{comp\_freezer} + \dot{E}_{comp\_chiller\_for\_freezer}}$$ **Equation 13**

$$COP_{chiller} = \frac{\dot{Q}_{o\_chiller}}{\dot{E}_{comp\_chiller} - \dot{E}_{comp\_chiller\_for\_freezer}}$$ **Equation 14**

It is necessary to measure the enthalpy outlet of the low pressure compressor. The electric power in the high stage compressor due to the freezer is calculated as follow:

$$\dot{Q}_{cond\_freezer} = \dot{m}_{freezer} \cdot \left(h_{out\_comp\_freezer} - h_{in\_cab\_freezer} \right)$$ **Equation 15**

$$\dot{E}_{comp\_chiller\_for\_freezer} = \dot{E}_{comp\_chiller} \cdot \frac{\dot{Q}_{cond\_freezer}}{\dot{Q}_{cond\_freezer} + \dot{Q}_{o\_chiller}}$$ **Equation 16**

4.2.2. Total COP

To assess the total COP each term has been calculated.

$$COP_{tot\_booster} = \frac{\dot{Q}_{o\_freezer} + \dot{Q}_{o\_chiller}}{\dot{E}_{comp\_freezer} + \dot{E}_{comp\_chiller}}$$

Equation 17

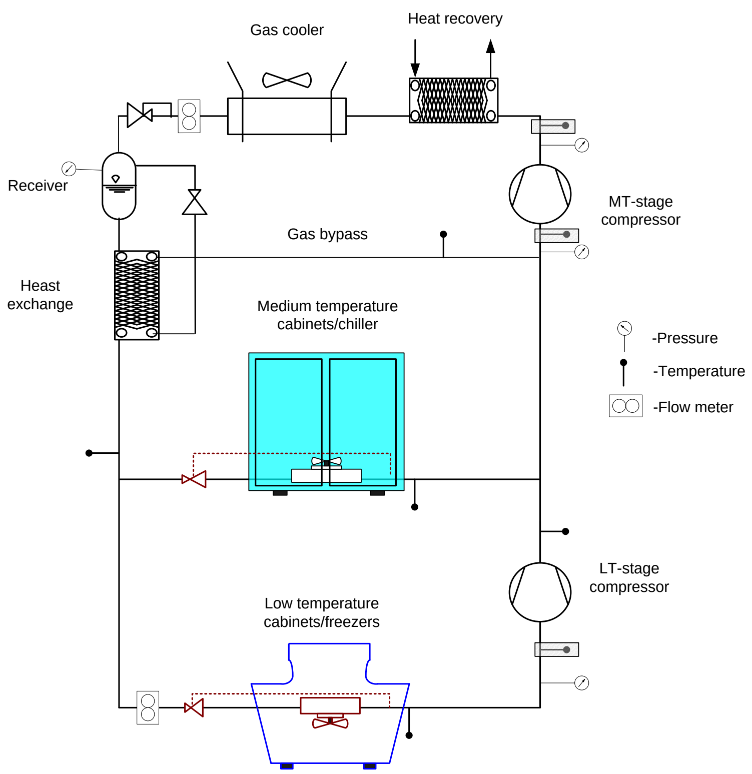


Figure 15: Instrumentation in CO2 Booster system

In the schematic system has been located the above sensors. Two flow meters have been located in the liquid part due to the high density and consequently less size and price of the sensors. The mass flow in the by-pass is calculated using an energy balance. Temperature sensors have been located in the outlet of the compressor as well as in the evaporators outlet directly measuring the useful superheat in the cabinets. The pressure levels of the system are also measured. The temperature in the vapor bypass in the intermediate receiver is measured too, necessary for the heat balance in the heat exchanger, to determinate the mass flow in the by-pass.

"A p-h diagram for the complete cycle is shown in Figure 16, including explanations. It shows a simplified pressure-enthalpy diagram with explanations that also relates to measurement points in image showed before. The distance between some of the parameters and the vapor-liquid saturation lines has been exaggerated to clarify how the systems operate. The figure includes:

1) Cooling of the refrigerant in the condenser [e-f]
2) Expansion of the refrigerant [f-g]
3) The refrigerant entering the receiver where liquid and vapor separation takes place [g-h] and [g-l] respectively.
4) Expansion of the vapor in two parallel expansion valves [l-m].
5) The vapor-liquid mixture and the liquid from the receiver entering the heat exchanger. The vapor-liquid mixture is heated up and the liquid is sub-cooled [m-n] and [h-i] respectively.
6) Expansion of the refrigerant before the cooling cabinets [i-j].
7) Evaporation of the refrigerant in the medium temperature cabinets [j-(m)]
8) Expansion of the refrigerant before the freezers [j-k].
9) Evaporation of the refrigerant in the freezers [k-a] including external super heat.
10) Subcritical compression of the refrigerant [a-b]
11) The refrigerant from the 1:st stage compressor discharge being mixed with the flow from the medium temperature cabinets and the vapor from the heat exchanger [b+n+MT=c].
12) The refrigerant being compressed before entering the heat recovery system [c-d].
13) The refrigerant rejecting heat to the ventilation air in the heat recovery system via a heat exchanger with a water-glycol loop on the heat sink side [d-e]. " (5)

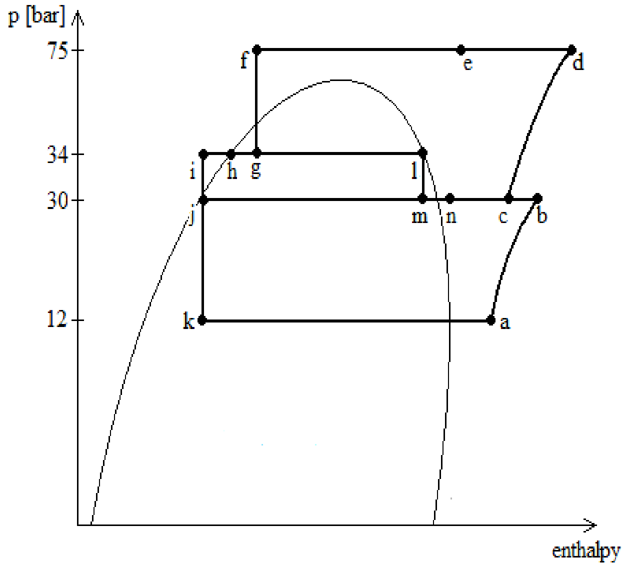


Figure 16: Simplified P-h diagram for a CO2 booster system during trans-critical operation. (5)

4.3. CO2 trans-critical systems: Parallel arrangement

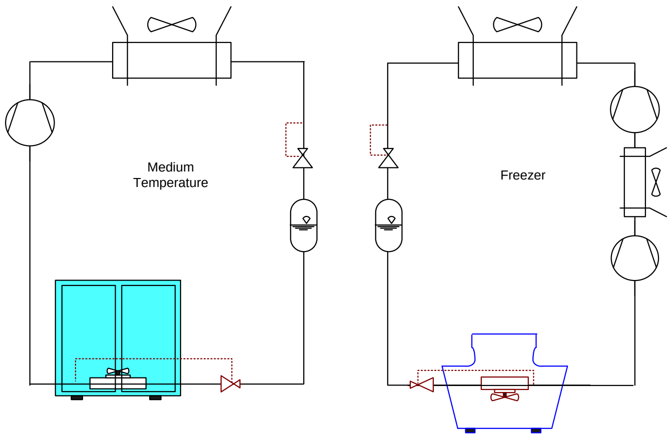


Figure 7: Schematic diagram of CO2 parallel system solution with two-stage compression on the low temperature level.

This is one of the possible solutions using CO2 in supermarket applications. This solution consist of two separate circuits, one of them satisfies the needs of cooling capacity in the medium temperature cabinets and the other covers the needs of the freezers. The system used in the freezer is composed by two-stage compression, it is necessary to avoid the high discharge temperature, while the medium temperature has only a single stage compression process. Both systems use direct expansion (7). The use of CO2 calls for a special design of the various elements in the refrigeration circuit in order to handle the high pressure.

This equation would be used in this system, where the evaporators at different temperatures of phase change are uncoupled. So in order to compare different cooling systems we need to evaluate the energy extracted from the cooling system, Q_0, and the electric energy consumption in the process.

$$\dot{Q}_{o\_freezer} = \dot{m}_{freezer} \cdot \Delta h_{freezer} = \dot{m}_{freezer} \cdot (h_{out\_cab\_freezer} - h_{in\_cab\_freezer}) \qquad \textbf{Equation 18}$$

$$\dot{Q}_{o\_chiller} = \dot{m}_{chiller} \cdot \Delta h_{chiller} = \dot{m}_{chiller} \cdot (h_{out\_cab\_chiller} - h_{in\_cab\_chiller}) \qquad \textbf{Equation 19}$$

To evaluate the equation we need the inlet and outlet enthalpies of the evaporator. The instrumentation to evaluate the terms of the above equations is the same indicate in the Table 1 for low temperature unit and Table 3 for medium temperature unit.

4.3.1.COP medium and low temperature

The electric power is measured and the cooling capacity is calculated. The ratio consists of the definition of the COP.

$$COP_{freezer} = \frac{\dot{Q}_{o\_freezer}}{\dot{E}_{comp\_freezer}}$$ **Equation 20**

$$COP_{chiller} = \frac{\dot{Q}_{o\_chiller}}{\dot{E}_{comp\_chiller}}$$ **Equation 21**

4.3.2.Total COP

The total COP is calculated with the following equation:

$$COP_{tot} = \frac{\dot{Q}_{o\_freezer} + \dot{Q}_{o\_chiller}}{\dot{E}_{comp\_freezer} + \dot{E}_{comp\_chiller}}$$ **Equation 22**

In the schematic system has been located the above sensors and other that are interesting to characterize the compressors behavior. Two flow meter (coriolis flow meters) have been located, one in each circuit, in the liquid part due to the high density and consequently less size of the sensors.

The temperature sensors have been located in the inlet and outlet of the compressor inside a rectangle. These sensors will be necessary for the mass flow estimations if it is not installed a mass flow meter. In the evaporator's outlet are located temperature sensors to directly measuring the useful superheat in the cabinets. The pressure levels of the system are also measured. The power consumption of the compressors is directly measured by a power meter.

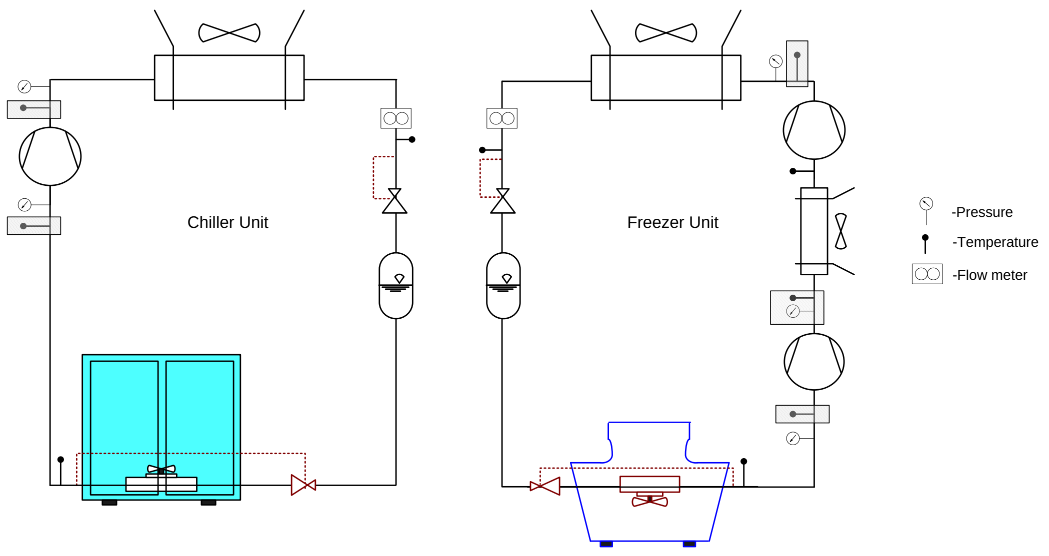


Figure 17: Instrumentation in CO2 Parallel arrangement system

4.4. Cascade systems with CO2

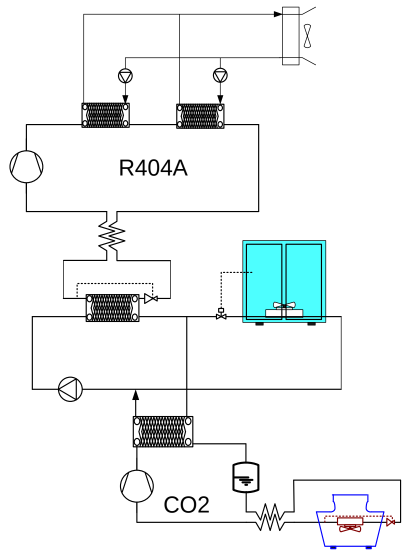


Figure 5: Schematic diagram of R404A-CO2 cascade system with brine at the medium temperature level.

There are industrial applications that require moderately low temperatures. In general, it is not possible to use a single vapor compression cycle for these temperatures because the temperature difference between the condenser and the evaporator is very large in this case. Consequently, the compression ratio is also large so the compressor works with lower efficiency.

To follow using the vapor compression system is used a cascade system. Cascade cycle is a set of simple cycles of vapor compression, where the condenser of the low temperature cycle is connected with the evaporator of the high temperature cycle via a heat exchanger, or heat transfer loop.

Although they can be used two, three or more units in series, for the temperature level used in supermarkets only needs two units. It uses a different refrigerant in each cycle, in order to satisfy the requirements of each interval of temperature and pressure. Being the same or different refrigerant, in general the refrigerant mass flow rates in the two cycles are not the same. The mass flow is determined by the required cooling capacity and the speed of heat transfer between the fluids in the heat exchanger.

An energy balance for the heat exchanger that connect the evaporator and the condenser shows that the mass flow rate in each cycle is determined by changes in enthalpy of each fluid when it gets through the heat exchanger.

In the freezers carbon dioxide is used as a refrigerant while in the high temperature part is used R404A. The main advantage of this system is that conventional components can be used;

design pressure is to normal levels. If system failure occurs then the CO2 is releases to ambient or an auxiliary cooler can be connected to the CO2 units, this auxiliary cooler is rather small in size.

4.4.1.COP medium and low temperature

The cooling capacity in the low temperature unit is calculated using $\dot{Q}_{o\_freezer} = \dot{m}_{freezer} \cdot \Delta h_{freezer} = \dot{m}_{freezer} \cdot (h_{out\_cab\_freezer} - h_{in\_cab\_freezer})$ Equation 4. The energy consumption from the compressors and pumps can be directly measured with a power meter. The variables that need to calculate or measure are:

$$\dot{Q}_{o\_freezer}, \dot{Q}_{o\_chiller}, \dot{Q}_{cond\_freezer}, \dot{E}_{comp\_freezer}, \dot{E}_{comp\_chiller}, \dot{E}_{pump}$$

With the $\dot{Q}_{o\_chiller} = \dot{m}_{chiller} \cdot \Delta h_{chiller} = \dot{m}_{chiller} \cdot (h_{out\_evap\_chiller} - h_{in\_evap\_chiller})$

Equation 7 is calculated the chiller cooling capacity and the heat rejected in the low temperature condenser is calculated with the following equation:

$$\dot{Q}_{cond\_freezer} = \dot{m}_{freezer} \cdot \left(h_{out\_comp\_freezer} - h_{in\_cab\_freezer}\right)$$ **Equation 23**

$$COP_{freezer} = \frac{\dot{Q}_{o\_freezer}}{\dot{E}_{comp\_freezer} + \dfrac{\dot{Q}_{cond\_freezer}}{COP_{chiller}}}$$ **Equation 24**

$$COP_{chiller} = \frac{\dot{Q}_{o\_chiller}}{\dot{E}_{comp\_chiller} + \dot{E}_{pumps}}$$ **Equation 25**

Assuming isenthalpic valve, the enthalpy at the evaporator inlet is the same as the enthalpy of the line between the expansion valves and the heat exchanger output (IHE). Since the state of refrigerant is sub-cooled liquid, we need the measure of the pressure and temperature to determine its enthalpy. In the outlet of the evaporator the refrigerant is superheated vapor, and for this reason using temperature and pressure transducer we can determinate the enthalpy of this point.

4.4.2.Total COP

The definition of the global COP is shown in the next equation.

$$COP_{tot} = \frac{\dot{Q}_{o\_freezer} + \dot{Q}_{o\_cab\_chiller}}{\dot{E}_{comp\_freezer} + \dot{E}_{comp\_chiller} + \dot{E}_{pump}}$$ **Equation 26**

Depending on the type of the compressor the losses are absorbed by the refrigerant, the environment or by the oil cooling system. It is important when we propose the balance equation in the low temperature loop. In the

Equation 23 we don’t use the power consumption by the compressor directly because not all this power goes to the refrigerant.

The balance in the brine loop is:

$$\dot{Q}_{o\_chiller} = \dot{Q}_{o\_cab\_chiller} + \dot{Q}_{cond\_freezer} + \dot{E}_{pump\_brine}$$

Equation 27

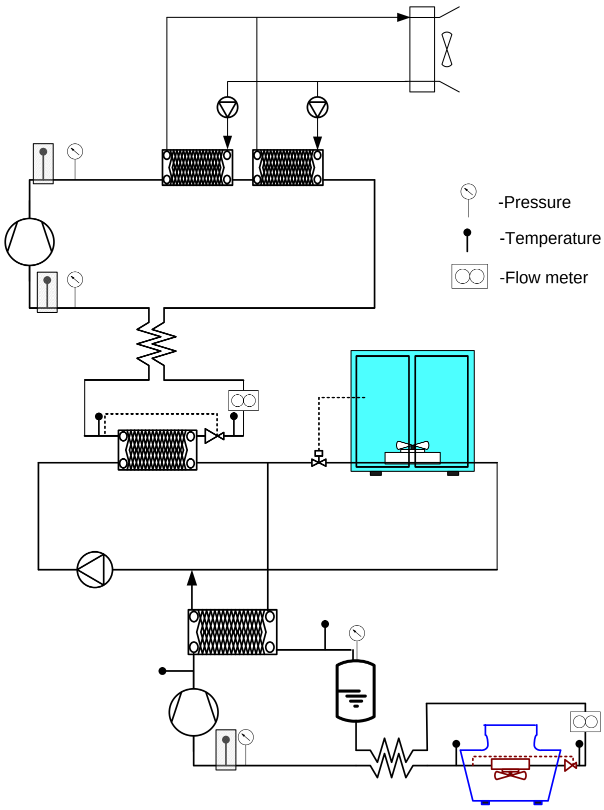


Figure 18: Instrumentation in Cascade system

5. Instrumentation needed

The first important parameters for the supermarket analysis are:

5.1. Temperatures

5.2. Pressures

5.3. Electric power consumption

The location of these sensors in the refrigeration system solution part has been indicated and how to assess the COP with the electric power measurement, by a power meter. Other important part is the data logging and the data processing.

5.4. Data acquisition software

There are 4 different systems used in data collection in the different field measurement installations. Each method has its own advantages and problems:

- IWMAC was used for systems TR1-3, CC2, CC3 and PC1. An interval of 5 minutes for TR1 and TR2 supermarket. (8)
- RDM - with an interval of 15 minutes for CC1 supermarket
- LDS (Long distance service)- for TR4 and TR5
- ViSi+ for RS1-3

During this work only IWMAC was used.

IWMAC: this system presents the following characteristics:

- Data collection via internet.
- Easy display of parameters. The system doesn't only record the information. It is possible to display the information using directly IWMAC.
- Clear plots or graphical display of parameters.
- Possibility to group parameters to download.

The image in Figure 3 is an example of the display window of the website. On the right the window for add new variables for download and on the left the window to see the evolution of the variables.

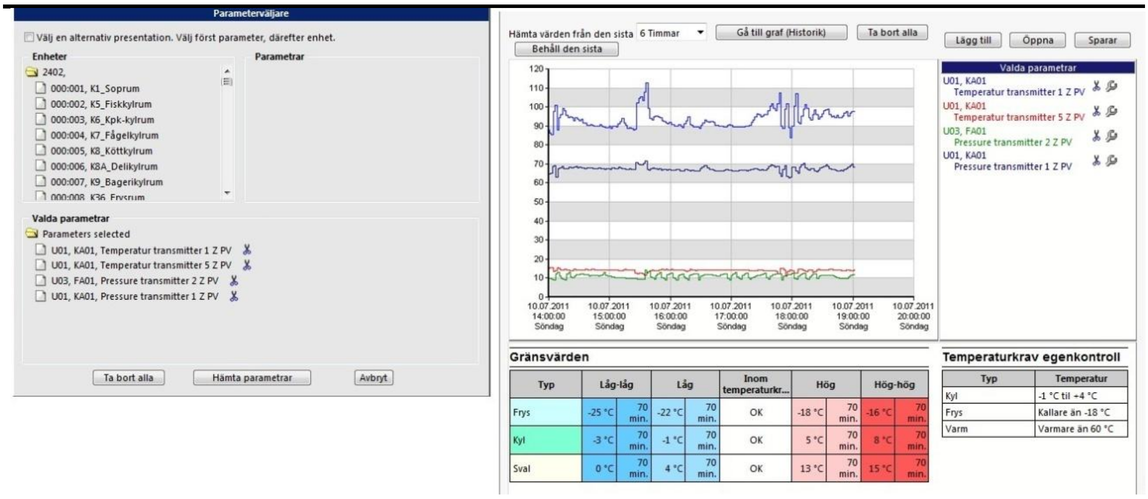


Figure 19: Display window, IWMAC website

Problems:
A major drawback of this system is that it records data as soon as it changes: that results in all the data becoming not normalized as each value recorded in unpredictable time, thus needed to be normalized. It will be explained in details in the normalization section

Another problem is when some measurements are done and there is accidental failure in the system. It is difficult to detect if there is a lack of data unless see on the plot in IWMAC (straight lines instead of fluctuating values). After normalizing the data it is difficult to detect such problem. For this reason it is important to run the filtering.

Long Distance Service (LDS): this program is used for the data acquisition for TR4 and TR5 and presents the next characteristics:

- It has not internet connection and use direct modem connection to download the data.
- The parameters are measured and the data is saved on an on-site computer.
- Data can then be accesses from any computer by connecting to the logging computers via modem, specifying which parameters to collect.
- It is possible to write scripts, determining which data that should be collected, for which period of time and when the collection should take place.
- Store all the values at the same time.

Problems:
"One problem with this method of data collection is that the on-site logging computers have a limited capacity to store data. If the data is downloaded to another computer via a modem once every 24 hours, the data is available with a minimum time interval of two minutes. If the download is made less frequent, for example once a week, the logging computers will have

started to delete data in order to save space on the hard drives. The time interval of the data will not change but there will be gaps of several hours in the data series. This has resulted in major problems with the data collection for the study." (5)

If the connection problems were not detected in time there was a data loss due to limited amount of storage capacity on server side.

Although the measurements are taken frequently and the amount of data is quickly increased, taking into account the new size technology of data storage and the reduced price should not be a problem to take all the measurements each time period.

5.4.1.Sum up

Seeing the different advantages and problems in each acquisition system is recommended for a new acquisition system:

- Data collection via internet. It is an important aspect that allows working with the supermarket without the need to travel. It is interesting to note that in previous projects systems with distances of over 500km apart have been compared. All this work was done while located in Stockholm. Without this possibility for the data collection the analysis would be only possible by people close to the supermarket.

- Easy display of parameters. Possibility of display the information with clear plots or graphical display of parameters.

- Possibility to group parameters to download. It is important to download all the parameters needed at the same time for the studied period of time. If it is not downloading all the parameters at the same time, is more likely any mistake in the analysis.

- Store all the values at the same time. This is really important because it avoids the normalization process.

5.5. Normalization the data:

The data logged by IWMAC system used as an input to the software. The normalization is done by averaging of the values within "Time delta of average", output data time interval, time starting from the first data entry in the input file till the user specified "Date end" time point. Processed in this way 5 minutes average data has been used for further numerical analysis in all the supermarket systems logged with IWMAC.

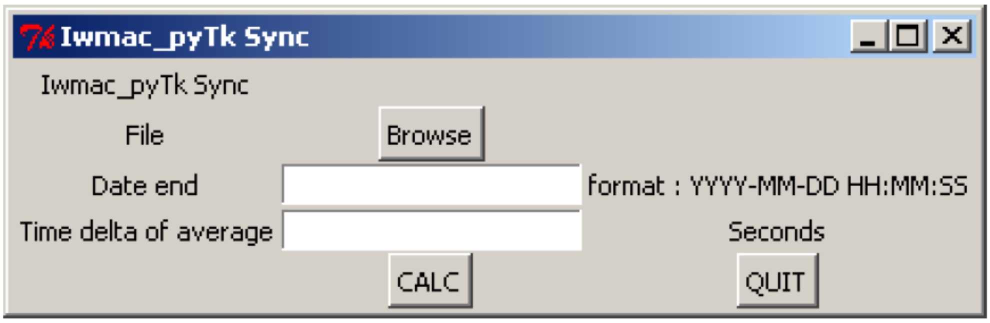


Figure 20: Normalization software

This is the program's interface to normalize the data extract from IWMAC webpage. The program give as a result a new file, with the extension ".OK" that is possible to open with Excel. The first question is if it's necessary filtering the data.

IWMAC used to register the data every 5 minutes, but if the data is the same as the last register, it doesn't register a new value. Currently, it records data when a change in its value occurs, but not more frequently than 2 minutes. This produces different number of measurement for each variable.

In order to know an average value in a week or a month it's necessary the normalization process. It is possible to find the next situation:

Time	Electric Power [kW]
0:00:00	10
0:50:00	20
1:00:00	10

Table 4: One hour electric power measured from iwmac

The compressor unit is consuming 10kW for 50 minutes and 20kW for 10 minutes. For this hour the correct average value is 11.66kW, due to it must be weighed the value over time. The direct average for this hour is 15kW.

The other problem commented is the different number of registers in each variable. The following Figure shows the data extract form IWMAC, for a day (24/6/2009), before filtering. Three parameters are extracted: Outdoor temperature, discharge pressure KA1 (medium temperature unit) and compressor unit active Power.

```
                       Temperatur                                   Pressure                    P: total active power
2009-06-24 00:00:00    8,9     2009-06-24 00:00:00    54,4    2009-06-24 00:00:00    23,6
2009-06-24 00:05:00    8,6     2009-06-24 00:05:00    52,8    2009-06-24 00:05:00    11,6

2009-06-24 23:45:00    9,6     2009-06-24 20:15:00    67,4    2009-06-24 22:55:00    12,4
2009-06-24 23:50:00    9,1     2009-06-24 20:20:00    69,2    2009-06-24 23:00:00    24,9
2009-06-24 23:55:00    9,8     2009-06-24 20:25:00    67,1    2009-06-24 23:05:00    12,3
2009-06-25 00:00:00    9,7     2009-06-24 20:30:00    63,6    2009-06-24 23:10:00    12,1
                               2009-06-24 20:35:00    65,1    2009-06-24 23:15:00    12,2
                               2009-06-24 20:40:00    69,9    2009-06-24 23:20:00    12,1
                               2009-06-24 20:45:00    63,7    2009-06-24 23:25:00    12,0
                               2009-06-24 20:50:00    67,1    2009-06-24 23:30:00    11,9
                               2009-06-24 20:55:00    68,4    2009-06-24 23:35:00    11,8
                               2009-06-24 21:00:00    68,8    2009-06-24 23:45:00    11,9
                               2009-06-24 21:05:00    67,5    2009-06-24 23:50:00    12,0
                               2009-06-24 21:10:00    68,2    2009-06-24 23:55:00    11,6
                               2009-06-24 21:15:00    67,3    2009-06-25 00:00:00    24,2
                               2009-06-24 21:20:00    67,2
                               2009-06-24 21:25:00    66,5
                               2009-06-24 21:30:00    67,3
                               2009-06-24 21:35:00    61,6
                               2009-06-24 21:40:00    62,5
                               2009-06-24 21:45:00    65,6
                               2009-06-24 21:50:00    61,6
                               2009-06-24 21:55:00    60,5
                               2009-06-24 22:45:00    58,3
                               2009-06-24 22:50:00    61,1
                               2009-06-24 23:25:00    57,1
                               2009-06-25 00:00:00    56,3
```

Figure 21: Text file, data from iwmac

	Total (for a day)	Outdoor temperature	Discharge pressure	Electric Power
Number of measures	288	241	282	250

Table 5: Number of measures

The result is only for a day and it's possible to see the differences. These differences on the number of registers increase if we extract stable variables like the compressors running indicator (binary variable) and of course if data is collected for longer periods of time. The more stable a variable is, the fewer registers are made and more memory space is available, probably this is the reason for this data collection strategy. Saving memory space results on extra work when analysis of the system is needed.

The normalization of the data is to match all the different parameters in the same period of time. In the following Figure two parameters are shown, the electric power and the compressor_1 running or not. There are high differences between the frequencies of the registers. Once the data is synchronized then it is possible to start the calculation of different parameters.

The following Figure shows schematically the normalization process. Using all the registered values between the time intervals that we decided, the software averages all the values and returns a single record for this delta time, assigning it to the time step at the end of the period.

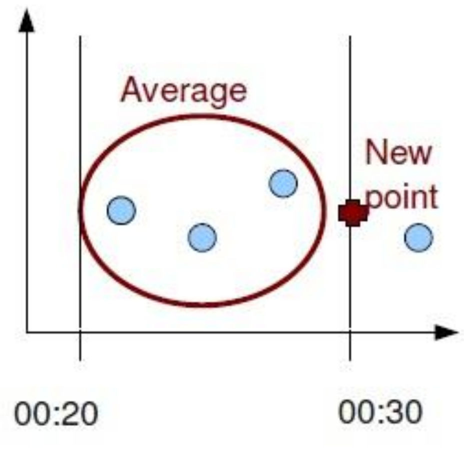
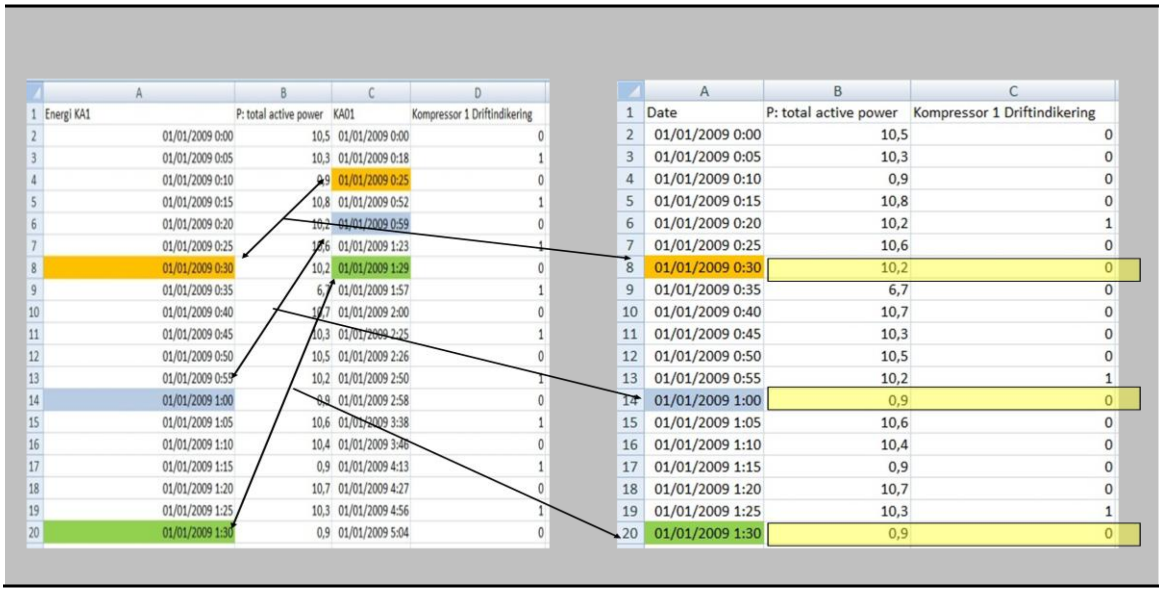


Figure 22: Function normalization software



Before normalization

	A	B	C	D
1	Energi KA1	P: total active power	KA01	Kompressor 1 Driftindikering
2	01/01/2009 0:00	10,5	01/01/2009 0:00	0
3	01/01/2009 0:05	10,3	01/01/2009 0:18	1
4	01/01/2009 0:10	0,9	01/01/2009 0:25	0
5	01/01/2009 0:15	10,8	01/01/2009 0:52	1
6	01/01/2009 0:20	10,2	01/01/2009 0:59	0
7	01/01/2009 0:25	10,6	01/01/2009 1:23	1
8	01/01/2009 0:30	10,2	01/01/2009 1:29	0
9	01/01/2009 0:35	6,7	01/01/2009 1:57	1
10	01/01/2009 0:40	10,7	01/01/2009 2:00	0
11	01/01/2009 0:45	10,3	01/01/2009 2:25	1
12	01/01/2009 0:50	10,5	01/01/2009 2:26	0
13	01/01/2009 0:55	10,2	01/01/2009 2:50	1
14	01/01/2009 1:00	0,9	01/01/2009 2:58	0
15	01/01/2009 1:05	10,6	01/01/2009 3:38	1
16	01/01/2009 1:10	10,4	01/01/2009 3:46	0
17	01/01/2009 1:15	0,9	01/01/2009 4:13	1
18	01/01/2009 1:20	10,7	01/01/2009 4:27	0
19	01/01/2009 1:25	10,3	01/01/2009 4:56	1
20	01/01/2009 1:30	0,9	01/01/2009 5:04	0

After normalization

	A	B	C
1	Date	P: total active power	Kompressor 1 Driftindikering
2	01/01/2009 0:00	10,5	0
3	01/01/2009 0:05	10,3	0
4	01/01/2009 0:10	0,9	0
5	01/01/2009 0:15	10,8	0
6	01/01/2009 0:20	10,2	1
7	01/01/2009 0:25	10,6	0
8	01/01/2009 0:30	10,2	0
9	01/01/2009 0:35	6,7	0
10	01/01/2009 0:40	10,7	0
11	01/01/2009 0:45	10,3	0
12	01/01/2009 0:50	10,5	0
13	01/01/2009 0:55	10,2	1
14	01/01/2009 1:00	0,9	0
15	01/01/2009 1:05	10,6	0
16	01/01/2009 1:10	10,4	0
17	01/01/2009 1:15	0,9	0
18	01/01/2009 1:20	10,7	0
19	01/01/2009 1:25	10,3	1
20	01/01/2009 1:30	0,9	0

Figure 23: Before and after the normalization

Besides the software provided by IWMAC for the data synchronization, it is possible to use a Macro in Excel, created by Vincent Cottineau that make the same work as the software. The differences are:

- We don't need to work with different files and different programs.
- It avoids the process to export the data form text to Excel two times.
- It doesn't average between the points in a period. It takes the last measure and it is maintained until a new register.
- There are fewer problems with the synchronization, in case of the number of compressor running due to the few registrations.

When this macro takes the values it is lost information because it doesn't average in each period all the points. It takes the last value registered in each period. If the delta time is 2 hour there would be more differences, but choosing a small delta time, 2 or 5 minutes, the final result is the same.

5.6. Filtering guidelines

Although the supermarket has a good instrumentation, there are always different problems that have as a consequence wrong register values.

The origins of this could be:

- The way how the values are registered, for some measurement points the value is recorded at the time of the measurement, such as the temperature and pressure sensors. In case of electric power consumption the averages are taken over a period of time.
- In the calculation template some assumptions had to be made, for example the internal superheat, it should not be a problem using this assumption in general especially when dealing with averages but may not fit well with every single point at each time step.
- For some points in the system differences in measurement accuracy may result in unrealistic values of thermo physical properties, example is the sub-cooling closer to the saturation line.
- The system is dynamic and the response time of the different parameters varies. For example, the pressure transducer will have quicker response than a temperature sensor attached to a pipes surface.

Thus the filtering is needed to improve the quality of analysis. It was done by using Excel filtering by deciding a range in which the value is expected and eliminating all the data out of this range. The same procedure has been routinely done for all the data collected.

This project studies the filtering process, the method followed in other reports and we will study a new method.

The following Figure shows the process that was followed in previous reports to evaluate the performance of supermarkets. After the normalization, explained in the earlier section, the parameters are introduced in the calculation template created for the system.

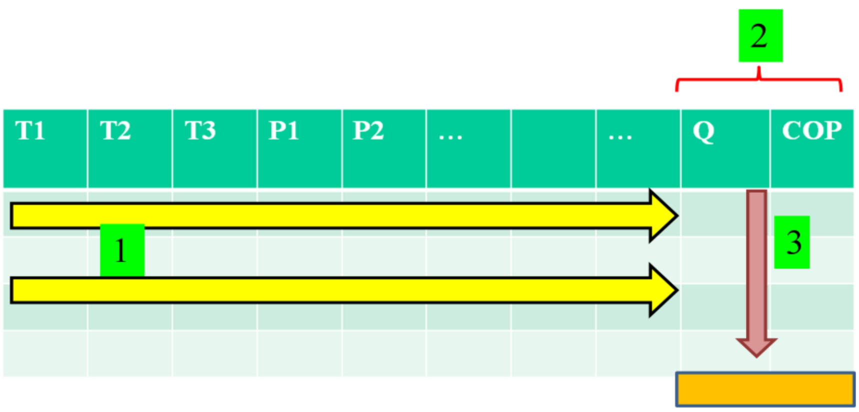


Figure 24: Calculation, filter and average

For each line of the template, it is said, for each time period used in the normalization, it is calculated all the needed parameters in order to calculate the COP and the cooling capacity.

For a week, with 5 minutes as average period in the normalization, each new intermediate variable necessary for the calculations adds 2.016 new values to the template. If in each template there are around 50 columns, for each month there are about 400.000 cells/ month.

After the calculation of the important parameters, filters are introduced in the columns of the Q (cooling capacity) and COP column. These filters delete rows with unexpected values that provide negative cooling capacities or COP higher than 7 or lower than 1. It is necessary to eliminate this values due to it can distort the analysis with unrealistic extreme values.

Finally it is obtained the important parameters averaging for a month. This lasts values are used to compare between systems, and they are the indicators used to analyze a complete year.

Advantages of the method followed in other reports, first method:

- It's possible to see the evolution of the COP and cooling capacity for a month, for a week, or for a day.

- The filtering process is more effective. It is difficult that a wrong combination of pressure and temperatures (some sensor's problem) give us a coherent COP.

- When the excel template being created, it's the same work, to calculate one line, than calculate 1.000 thousand line, we only need to drag the cells.

- On the other hand filtering the pressure and temperature, with their expected values or range, it is possible to obtain a COP out of range.

Example: CO2 cycle without IHE, condensing temperature 22°C, evaporation temperature -10°C, with 1K of sub cooling.

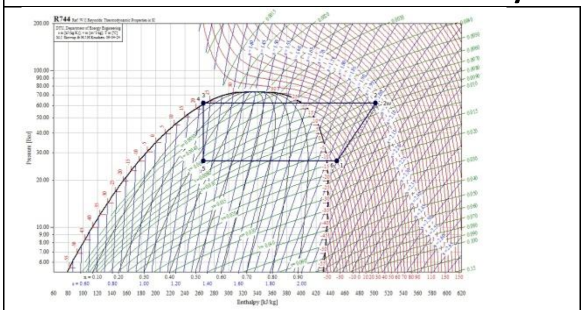
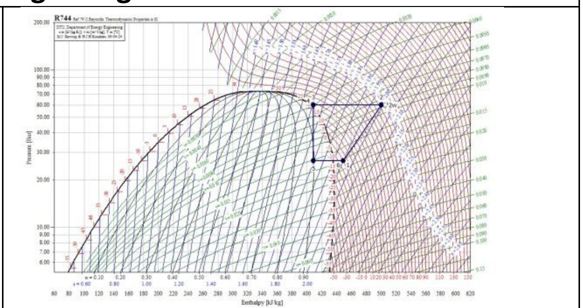
CO2 cycle single stage	
	
COP ≈3	COP≈0.7

Figure 25: COP difference due to 1K of sub cooling

If there is any problem with the temperature sensor measurement, because it is usually on the surface of the pipe, the pressure will be in the correct range, but the COP calculated is out the range. Filtering the COP this mistake would have been detected. It would also be detected with a column of sub-cooling value (exit temperature of the condenser minus saturated temperature of the condenser), whenever negative, but it implies a new filter.

Disadvantages of the method followed in previous reports in this project, first method:

- We work with large documents, with many values. It makes the computer slow.
- The document to plot the evolution in a year is difficult to work with.
- When a new filter is add to the file it needs time to calculate and the same when it is open a new file.

Other question is when there is a modification in the systems and we want to analyze this effect. In this case it is necessary the calculations more in detail, each 5 minutes, to see the new behavior and compare with the old.

In the reports has been used this way to analyze each systems. But we can use the same method changing the order of the steps: filtering first, average, and calculate after the COP. This method will be analyzed and the results will be compared.

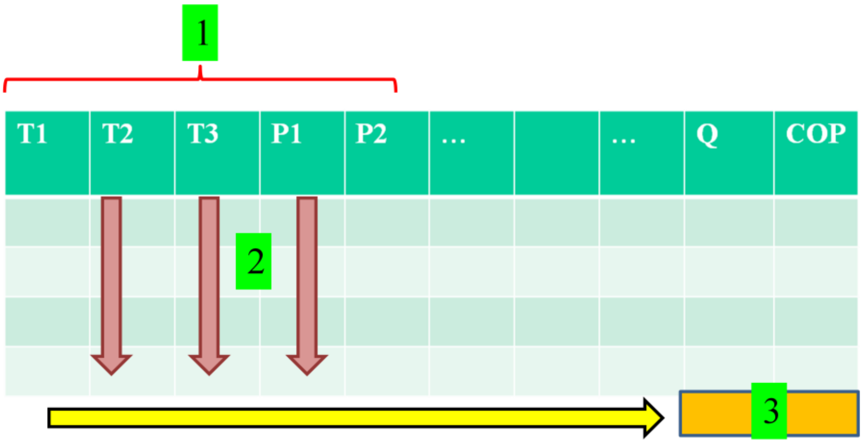


Figure 26: Average and calculate

5.6.1.Selection of the study period

For this analysis there are different possibilities:

1	Period length	**1 day**	1 week	1 month
2	Season	Winter	**Summer**	
3	Ambient temperature variation	+/-3	+/-5	**+/-7**
4	System	**TR**	RS	CC
5	Unit	Freezer	**Chiller**	

Table 6: Study day

The objective is to see the difference between the two methods for the day with higher fluctuations in temperature and cooling load. If the two methods give similar result for the day with greater fluctuations in temperature, for other day the two methods will be valid. The selection has been done in the table above with yellow color. The reasons are as follow:

1. For a day the fluctuation between the day and night period are represented. There are more deviations about the average value. If we increase the studied period, the results are covered by the average values, and the differences between the two methods will be lower.

2. In the summer period there are more fluctuations in the cooling capacity, and in this period it is higher as well.

3. The higher temperature differences during the day, the higher expected deviations between the two methods.

4. The systems TR, concretely TR1 use floating condenser. With high ambient temperature the pressure is elevate, crossing the trans-critical point. In this zone the properties of the refrigerant change a lot and the differences between the two methods will be prominent.

5. The freezer unit in the supermarkets doesn't present high fluctuations in cooling capacity, and electric power consumption. The differences are higher in the medium temperature cabinets, and for this reason if there are differences between the two methods, it will be higher in this case.

One of the available period studied is June 2009, in D. Frelechox Master Thesis (9). It has been search a day, in this month, with a great variation in temperature between day and night. The day chosen has been 24 of June. In the following Figure we can see the variation of ambient temperature:

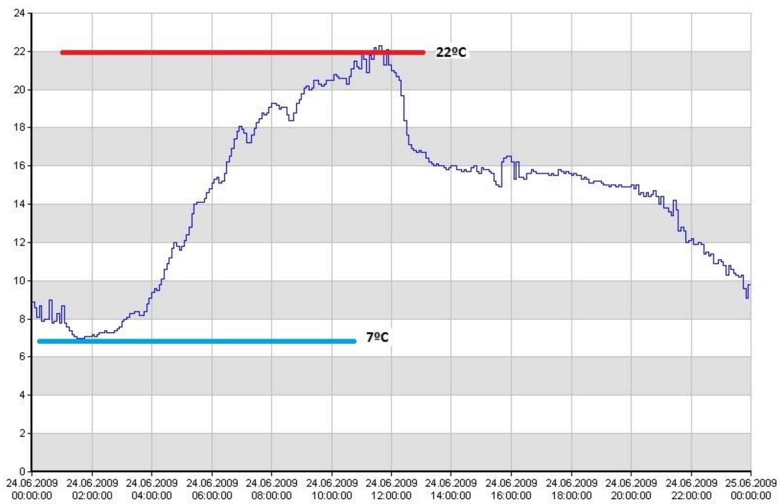


Figure 27: Ambient temperature, study day

This large variation has as a consequence variations in the power consumption (green), and in the discharge pressure (red), following the ambient temperature (blue). It is possible to see this behavior in the following Figure:

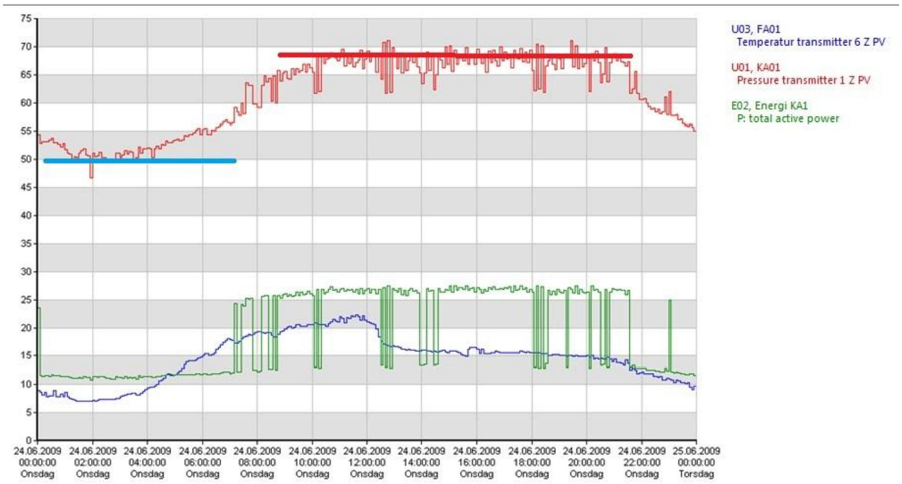


Figure 28: Compressor discharge temperature (red), electric power (green), ambient temperature (blue)

5.6.2.Results

The differences between the two methods are shown in the following plot. The relative differences during the study period between the two methods are:

In cooling capacity and COP: +/-3%
In enthalpies, pressures and temperatures: +/-1%

Although the results showed good correlation between the two methods, it is important to note that in the date analyzed there weren't too many inconsistent lines, necessary to remove. If there are lines with COP values for example 1000 or -600 and it is not removed before average, the results for this month won't be correct. The filtering part is really important.

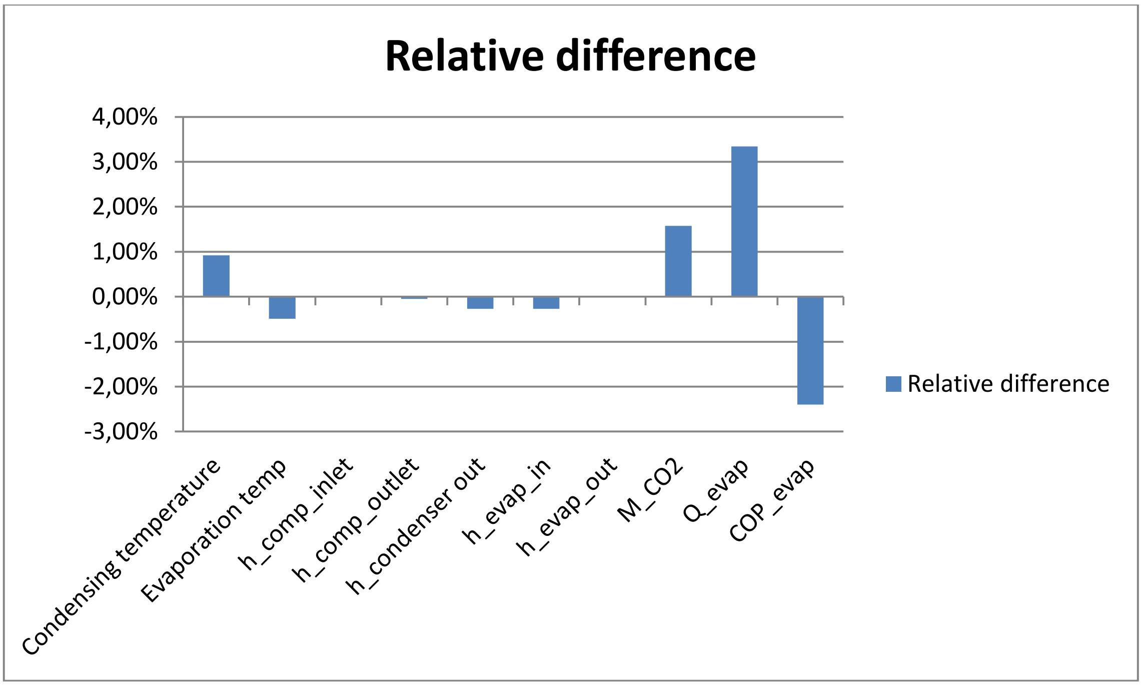


Figure 29: Differences 2 methods (relative)

5.6.3.Conclusions

The best procedure is to calculate for each line of data and then filter for the cooling capacity and COP and then average. In the second method the values that produce inconsistent COP or cooling capacity are not removed, and it will affect to the average result.

The arguments for not using the new method are:

- The accuracy of the method is directly related with the goodness of the data logger.
- The benefit is only computational.

The potential of the filtering process and the possibility to see the evolution during a day or week implies to follow with the same method.

6. Evaluation of compressor performance

6.1. Function and description:

The compressor has two basic functions in the refrigeration system. It has to maintain the pressure in the evaporator so that the liquid refrigerant can evaporate at the required temperature. On the other hand it compresses the refrigerant so that it can be reject the heat to the ambient.

The basic function of compressor control, therefore, is to adjust the capacity of the compressor to the actual demand of the refrigeration system so that the required products temperature can be maintained.

If the compressor capacity is bigger than the demand, the evaporating pressure and temperature will be lower than that required, and vice versa. Additionally, the compressor should not be allowed to operate outside of the acceptable temperature and pressure range, in order to optimize its running conditions. (10)

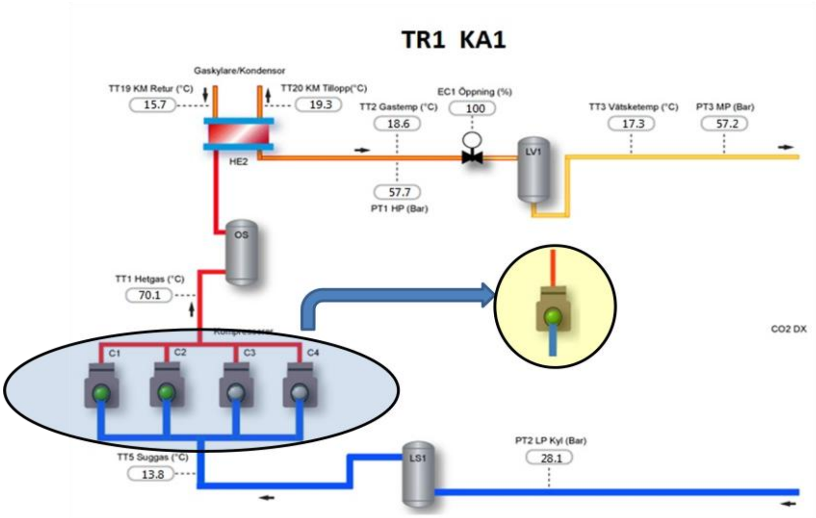


Figure 30: CO2 trans-critical system named TR1, iwmac software scheme

The different supermarkets analyzed have different number of compressors in parallel per refrigeration unit (medium and low temperature unit). These compressors suck the refrigerant from a common suction line, compress it and discharge into the condenser. This suction line contains the mixture of the outlets of each cabinet.

The use of a number of small compressors instead of a single large compressor means a greater difficulty in the control process, but is justified by the following reasons:

1. The most important is it reduces operating cost through greater control of capacity and power consumption. This is achieved by compressor switch-on sequences that allow the parallel system to match its power with the capacity needed.

2. Several compressors in parallel ensure safety in case of failure of any compressor. In case of failure or shutdown requirements for maintenance, by performing a bypass between compressors can keep the plant in continuous operation.

6.2. Compressor manufacturer information

The characterization of the compressor's performance is made by the compressor manufacturer in different normalized tests. The information extracted in this test is used for the approval of the compressor.

It has been analyzed the current instrumentation of each system and one of the restrictions is that in real refrigeration systems there aren't flow meters. To solve this problem it is possible to use the operation points provided by the compressor manufacturer. The mass flow can be indirectly calculated. The compressor's manufacturer provides the information in a table with the follow information:

Table from compressor manufacturer	
$\left\{\dot{Q}_0, \dot{P}_e\right\} = f\left(T_{evap}, T_{cond}, SC, SH\right)$	Q cooling capacity P electric power SC sub cooling (0K) SH superheat (internal) (10K)

Table 7: Information from compressor manufacturer

The information provided by the compressor manufacturer cannot be used directly but needs to be worked before.

It has been chosen a Dorin CO2 compressor model TCS373-D, and Bock CO2 compressor model HGX34/150-4 CO2 T. The Dorin compressor TCS373-D has been chosen because is used in the CO2 trans-critical supermarkets named TR1, TR2 and TR3 in other reports. The Bock compressor HGX34-150-4-CO2T has been chosen because it has similar characteristics, electric power and size. Then we can redo all the analysis made in other reports, such as (10), (6), (4) with other compressor; show how to work with the information provided by the compressor manufacturer.

The information provided is the following:

- Dorin CO2 (11)

transcritical single stage						
model	rpm	swept volume [m^3/h]	suction NPT	discharge NPT	weight [kg]	oil charge [kg]
TCS340/4-D	1450	3.5	1/2	1/2	141	1.8
TCS350/4-D	1450	4.3	3/4	1/2	144	1.8
TCS362/4-D	1450	5.4	3/4	1/2	147	1.8
TCS340-D	2900	7.0	3/4	1/2	144	1.8
TCS351-D	2900	8.8	3/4	1/2	148	1.8
TCS362-D	2900	10.7	3/4	1/2	152	1.8
TCS373-D	2900	12.6	3/4	1/2	152	1.8

TRANSCRITICO MONOSTADIO / TRANSCRITICAL SINGLE STAGE							
modello / model type	t_ev	p_suc	tgc_out	p_dis	beta	Q	P
TCS373-D	-20	19,72	15	75	3,803	26,0	11,8
			25	75	3,803	22,3	11,8
			35	90	4,564	16,3	12,2
			40	110	5,578	13,1	13,5
	-15	22,93	15	75	3,271	32,0	12,4
			25	75	3,271	27,4	12,4
			35	90	3,925	20,9	13,6
			40	110	4,797	16,8	14,9
	-10	26,50	15	75	2,830	39,5	12,5
			25	75	2,830	33,8	12,5
			35	90	3,396	25,9	13,9
			40	110	4,151	23,8	16,5
	0	34,86	15	75	2,151	56,4	14,5
			25	75	2,151	48,5	14,5
			35	90	2,582	37,1	16,8
			15	100	2,869	52,7	17,9
			25	100	2,869	46,1	17,9
			40	110	3,155	34,2	19,5
	5	39,69	15	75	1,890	66,6	14,7
			25	75	1,890	57,1	14,7
			35	90	2,268	44,5	17,7

Figure 31: Dorin compressor TCS373-D operation points

- **Bock CO2** (12)

CO_2 Type	Number of cylinders	Displacement 50 / 60 Hz	Electrical data ③ Voltage ①	Max. working current ②	Max. power consumption ②	Starting current (rotor locked)	Weight	Connections Discharge line DV ⑤	Suction line SV ⑥	Oil-charge
		m³/h		A	kW	A	kg	mm	mm I inch	Ltr.
				PW*1+2		PW*1/PW*1+2				
HGX34/110-4 CO_2 T	4	9,90 / 11,80	④	24,9	14,7	85/110	194	22	28 / 1 1/8	2,5
HGX34/110-4 S CO_2 T	4	9,90 / 11,80	④	31,0	17,7	110/141	197	22	28 / 1 1/8	2,5
HGX34/130-4 CO_2 T	4	11,30 / 13,60	④	28,5	16,9	85/110	194	22	28 / 1 1/8	2,5
HGX34/130-4 S CO_2 T	4	11,30 / 13,60	④	35,3	20,3	110/141	197	22	28 / 1 1/8	2,5
HGX34/150-4 CO_2 T	4	12,90 / 15,40	④	33,5	19,2	110 / 141	197	22	28 / 1 1/8	2,5
HGX34/150-4 S CO_2 T	4	12,90 / 15,40	④	39,3	23,1	127 / 158	200	22	28 / 1 1/8	2,5
HGX34/170-4 CO_2 T	4	14,50 / 17,40	④	37,6	21,7	110 / 141	196	22	28 / 1 1/8	2,5

CO_2 Performance data 50 Hz

Type			Cooling capacity Q_0 [W] / Power consumption P_e [kW] — Evaporating temperature °C: 15	10	5	0	-5	-10	-15	-20	-25	-30	-35	-40
HGX34/150-4 CO_2 T	t_c °C	SUBCRITICAL												
	10	Q							39950	33650	27850	22550	17850	13700
		P							8,35	8,50	8,55	8,50	8,25	7,85
	15	Q					51950	44700	37900	31600	25800	20550	15900	11850
		P					8,85	9,15	9,35	9,45	9,40	9,20	8,85	8,25
	20	Q				56200	48700	41600	35000	28900	23350	18400	14100	10450
		P				9,65	10,05	10,30	10,45	10,40	10,25	9,90	9,35	8,60
	25	Q				50350	43250	36650	30550	25000	20050	15750	12100	
		P				11,00	11,30	11,50	11,55	11,45	11,10	10,60	9,85	
	30	Q				40750	34500	28750	23600	19050	15150	11950		
		P				12,50	12,75	12,80	12,70	12,45	11,95	11,20		
HGX34/150-4 CO_2 T	t_{gas} °C	TRANSCRITICAL												
	30	P_{V2}				75	75	75	75	75	75	75		
		Q				41300	35050	29400	24300	19800	15800	12300		
		P				13,05	13,30	13,35	13,20	12,85	12,25	11,40		
	35	P_{V2}				90	90	90	90	90	85			
		Q				40700	34400	28750	23650	19050	14250			
		P				15,85	15,80	15,55	15,10	14,35	13,00			
	40	P_{V2}				100	105	105	105	100	85			
		Q				36100	31000	25500	20550	16150	7150			
		P				17,55	18,05	17,45	16,55	15,05	13,00			
	45	P_{V2}				110	110	110	110	110				
		Q				31350	26200	21500	17150	12000				
		P				19,20	18,75	18,00	16,90	15,05				
	50	P_{V2}				110	110	110	110	100				
		Q				23950	20050	16450	13150	7750				
		P				19,20	18,75	18,00	16,90	15,05				

Figure 32: Bock compressor HGX34/150-4 CO2 T operation points

The manufacturer tested the compressor in a simple cycle changing the evaporator and condenser pressure, and presents the results in a cycle with 10K as superheat, and without sub cooling. For each evaporator temperature is modified the condenser pressure.

With this information it is possible to know we can know the cycle followed by the refrigerant and calculate the mass flow in each operation point:

$$\dot{Q} = \dot{m} \cdot \Delta h \rightarrow \dot{m} = \frac{\dot{Q}}{h_{evap\_out} - h_{evap\_in}} \rightarrow \left\{ \begin{array}{c} \dot{Q} \\ h_{evap\_out} = f(SH, T_{evap}) \\ h_{evap\_in} = f(SC, T_{cond}) \end{array} \right\}$$

Q cooling capacity
SC sub cooling (0K)
SH superheat (internal) (10K)

With the real mass flow, measured by the compressor manufacturer in the compressor's test, the density and the swept volume flow rate $\dot{V}_c$ it can be calculated the volumetric efficiency and the total efficiency, using their definitions.

$$\eta_v = \frac{\dot{m}}{\dot{V}_c \cdot \rho_{in\_comp}} \rightarrow \left\{ \begin{array}{c} \dot{m} \\ \dot{V}_c \\ \rho_{in\_comp} = f(SH, T_{evap}) \end{array} \right\}$$

Mass flow
Swept volume flow rate
SH superheat (internal) (10K)

$$\eta_v = f(PR)$$

$$\eta_{tot} = \frac{\dot{m} \cdot \Delta h_{is}}{\dot{E}_{electric\_power}} \rightarrow \left\{ \begin{array}{c} \dot{m} \\ \Delta h_{is} = f(SH, T_{evap}) \\ \dot{E}_{electric\_power} \end{array} \right\}$$

Mass flow
Enthalpy differences between the inlet and isentropic outlet
SH superheat (internal) (10K)

$$\eta_{tot} = f(PR)$$

or

$$\eta_{tot} = f(PR, T_{evap})$$

The following four plots show the results of the volumetric efficiency correlation for three CO2 and one R404A compressors:

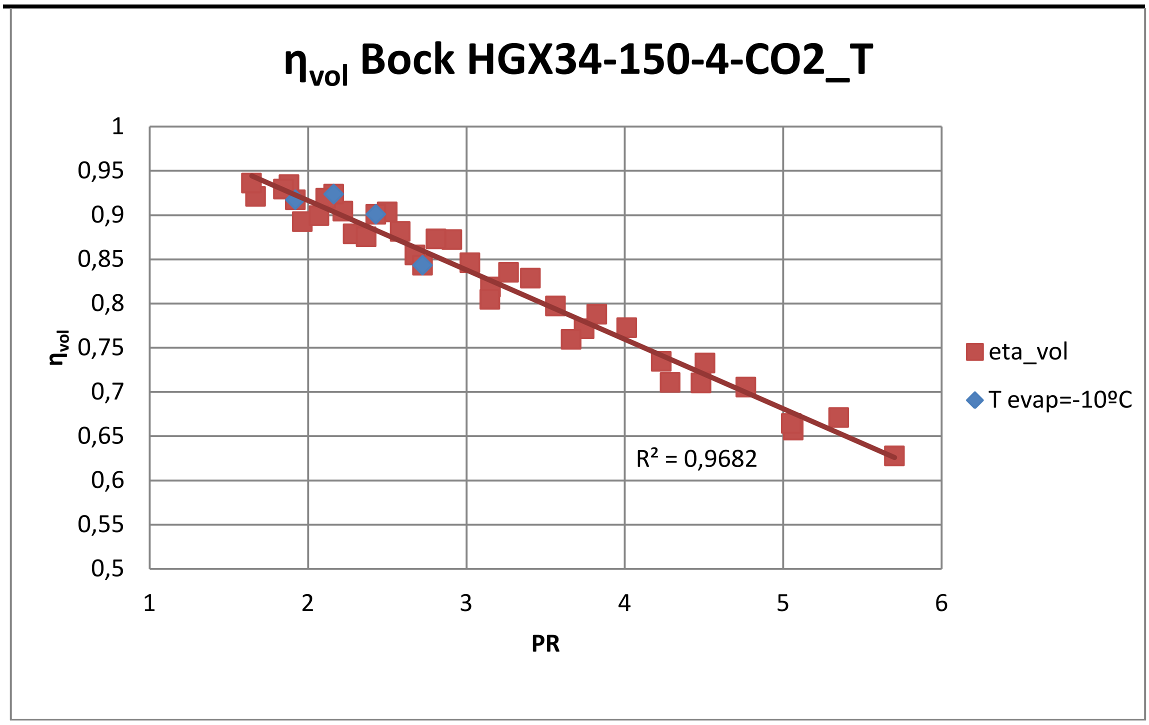


Figure 33: Volumetric efficiency curve Bock compressor HGX34-150-4-CO2

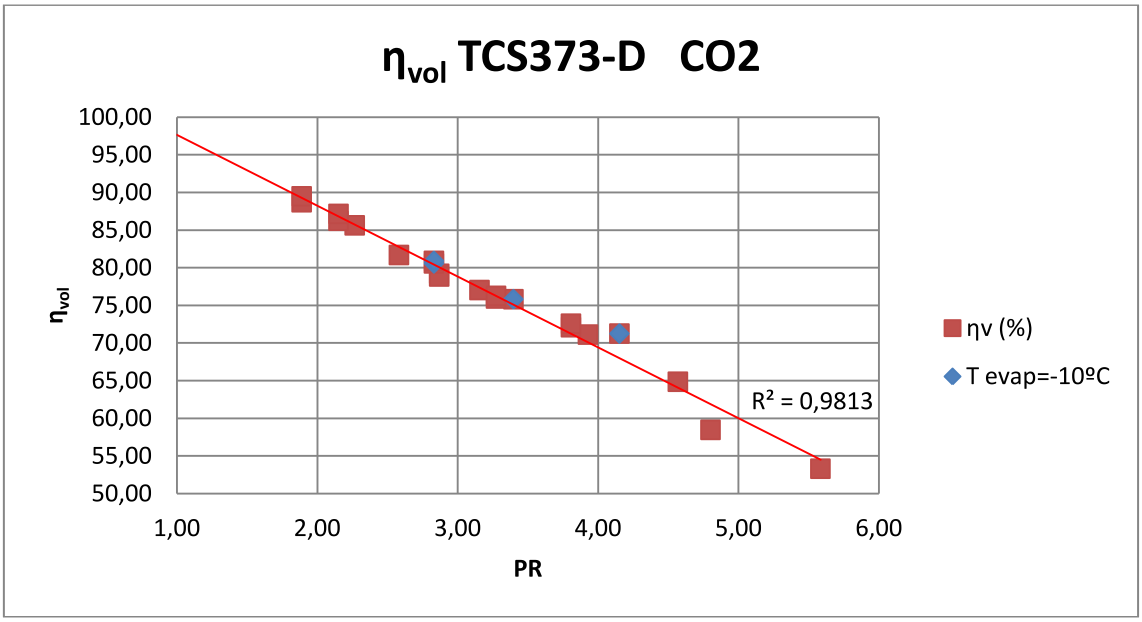


Figure 34: Volumetric efficiency curve Dorin compressor TCS373-D CO2

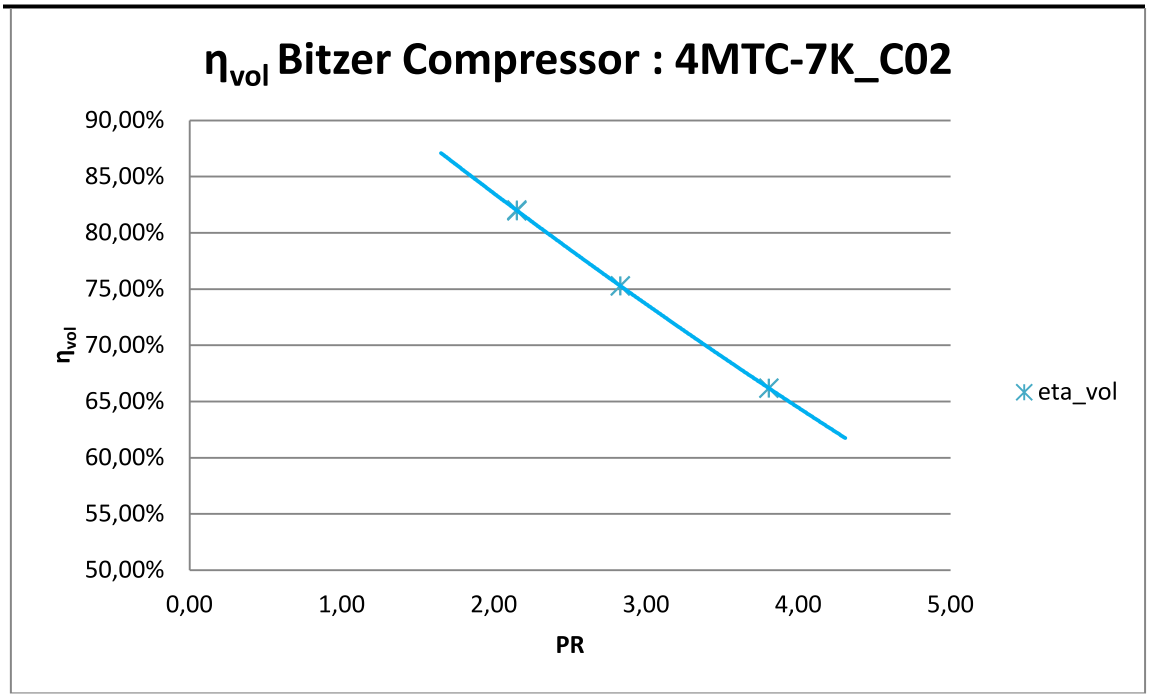


Figure 35: Volumetric efficiency curve Bitzer compressor 4MTC-7K CO2

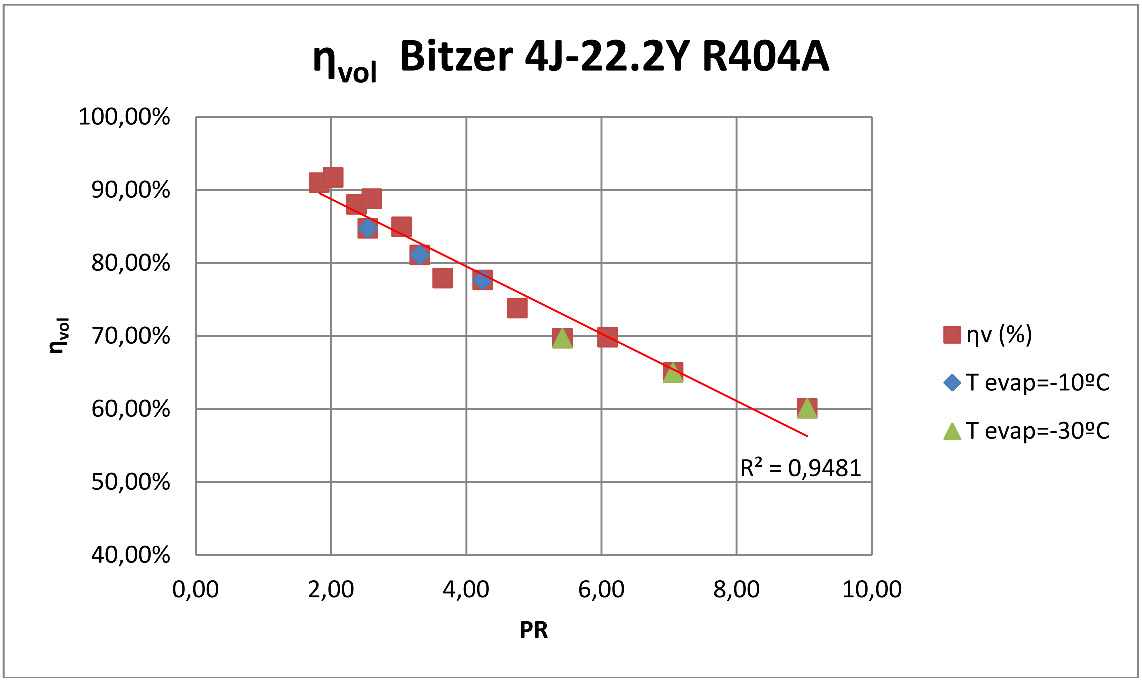


Figure 36: Volumetric efficiency curve Bitzer compressor 4J-22.2Y R404A

Although it is a high linear correlation, it is possible to find deviations around +/-2.5% for the R404A Bitzer compressor and around 2% for CO2 compressors that introduce an uncertainty factor in the estimation of the mass flow. The R^2 correlation coefficient is showed in the next table.

Bock HGX34-150-4-CO2_T	Dorin TCS373-D CO2	Bitzer 4MTC-7K_C02	Bitzer 4J-22.2Y R404A
R^2=0.9682	**R^2=0.9815**	**R^2=1 Only 3 operation points**	**R^2=0.9481 linear**

Table 8: R^2 coefficient for the volumetric efficiency curve

The information source is from the compressor manufacturer, and the tighter adjusted points the better correlation estimates.

For the total efficiency is possible to represent two curves: one curve fitting all the operation points from the compressor manufacturer, or a second curve correlating the points for each evaporator temperature. The result is the next:

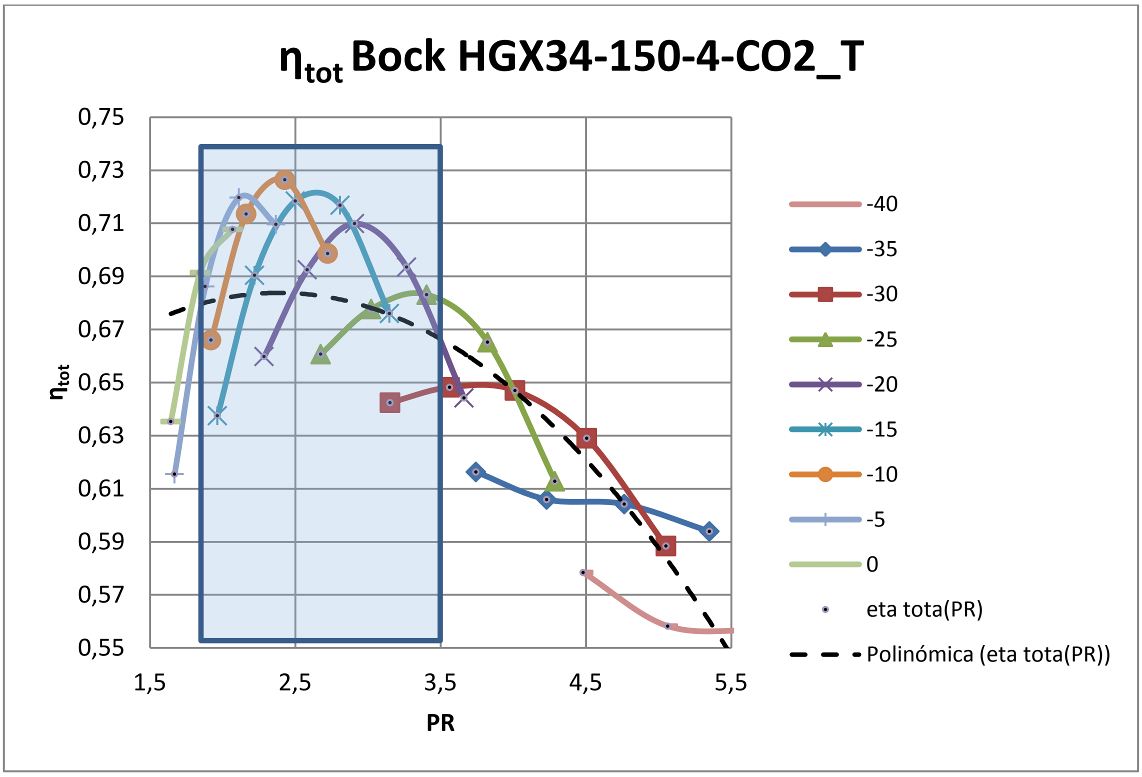


Figure 37: Total efficiency curves Bock compressor HGX34-150-4-CO2

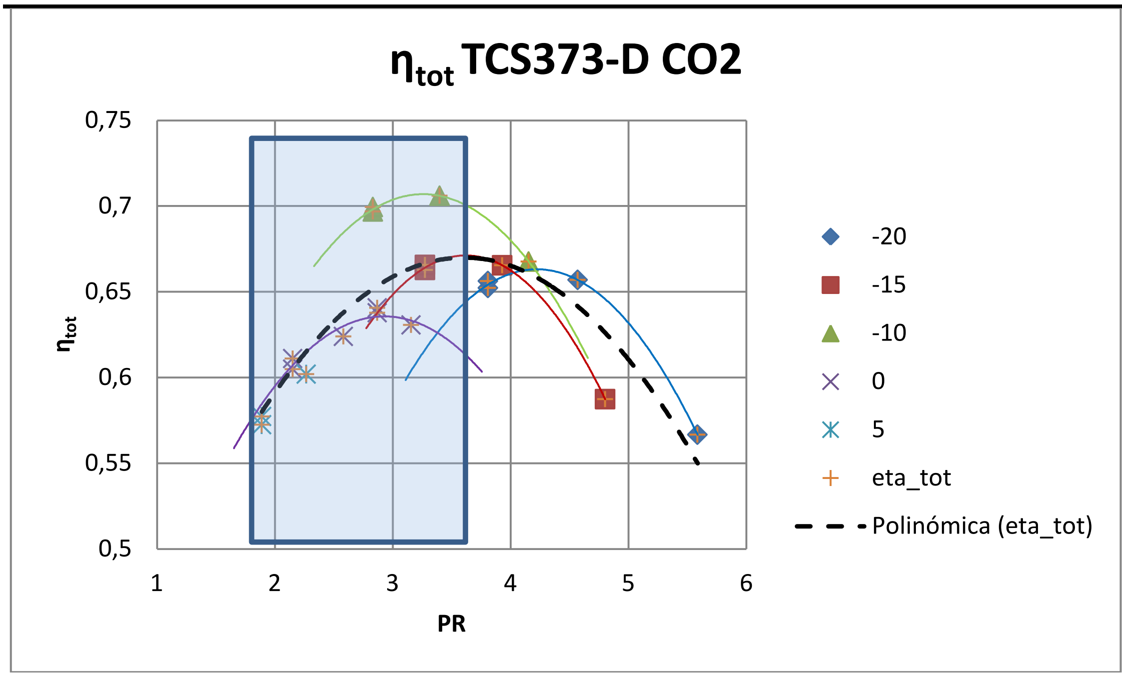


Figure 38: Total efficiency curves Dorin compressor TCS373-D CO2

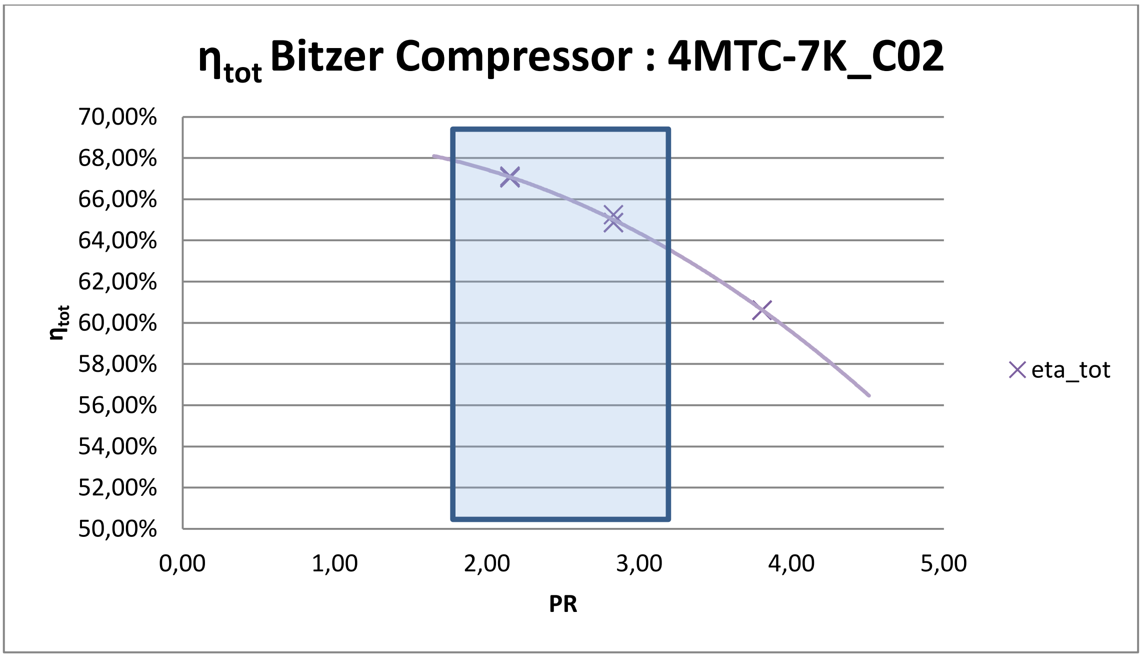


Figure 39: Total efficiency curve Bitzer compressor 4MTC-7K CO2

In the before compressor for each evaporation temperature there are only a pressure ratio. For this reason there are only one curve that correlate the fourth evaporation points.

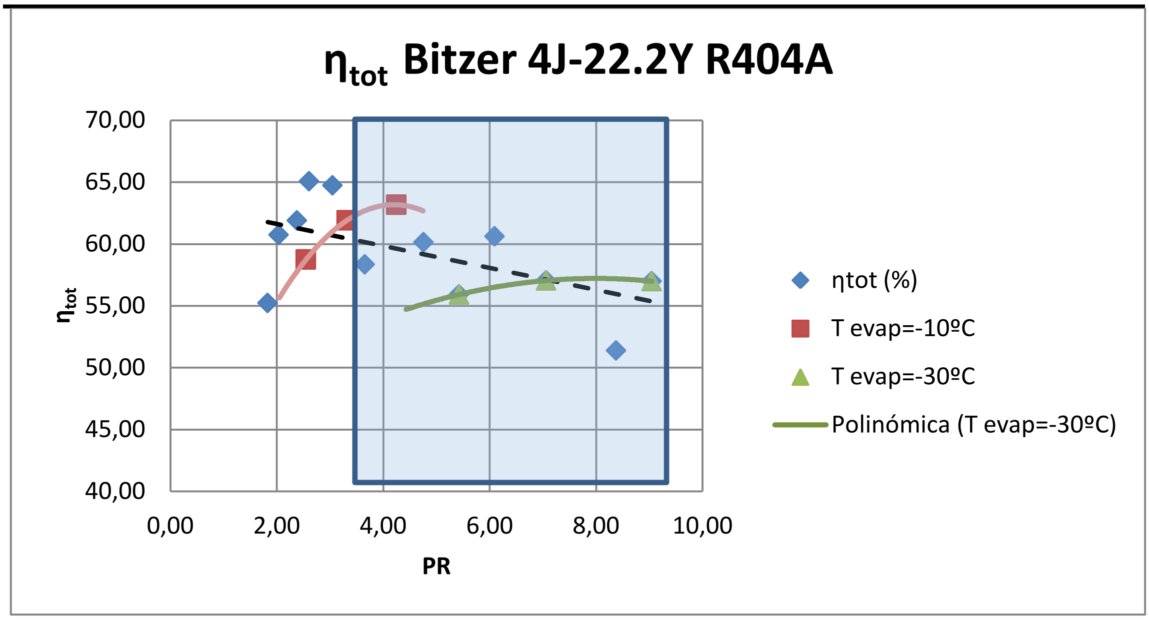


Figure 40: Total efficiency curves Bitzer compressor 4J-22.2Y R404A

The dark rectangle indicates the operation area for these compressors under supermarket conditions. Although it's possible to correlate the total efficiency with the pressure ratio, in this case the fluctuations are higher and the uncertainty introduced with each calculation with this total efficiency should be taken into account. It is possible to find points in +/-5% around the calculated correlation there is a difference in accuracy between curve fitting for all the PR points and curve fitting for the pressure ratios at a fixed evaporation temperature..

HGX34-150-4-CO2_T	TCS373-D CO2	Bitzer 4MTC-7K_C02	Bitzer 4J-22.2Y R404A
R^2=0.6692	R^2=0.7068	R^2=0.9984	R^2=0.3745

Table 9: R^2 coefficient for the total efficiency curve, fitting all the operation points

The total efficiency and the volumetric efficiency will be the parameters needed for the mass flow estimations and for this reason has been introduced. It will be developed in the chapter about the indirect mass flow measurements. In the volumetric efficiency method is necessary to know the nominal electric power per compressor for the supermarket operation conditions.

In the next plots are showed the electric power consumption for each compressor analyzed. It has been correlated for each evaporation temperature one curve as a function of the pressure ratio. The electric power estimated is useful to determine the number of compressors running in systems when it's measured only the total electric power per cooling unit and one of the compressors has frequency control. This number will be important in the volumetric efficiency method for estimate the mass flow.

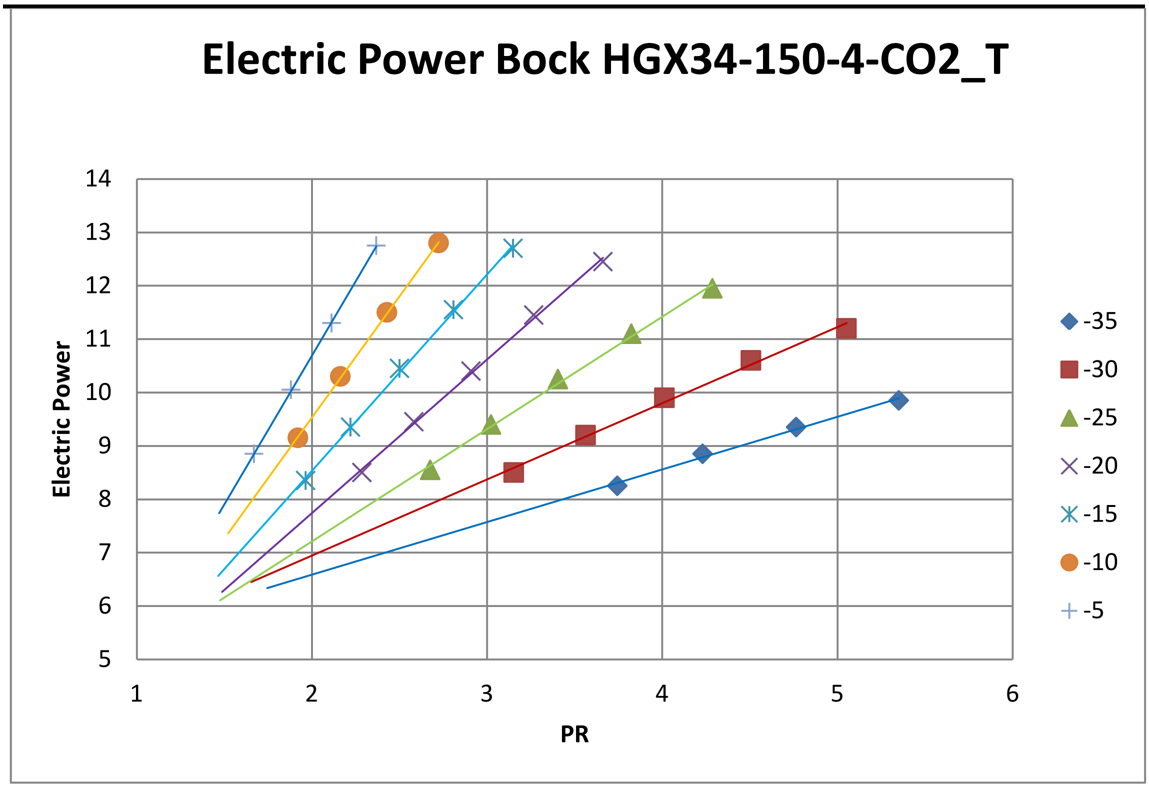


Figure 41: Electric power Bock compressor HGX34-150-4-CO2

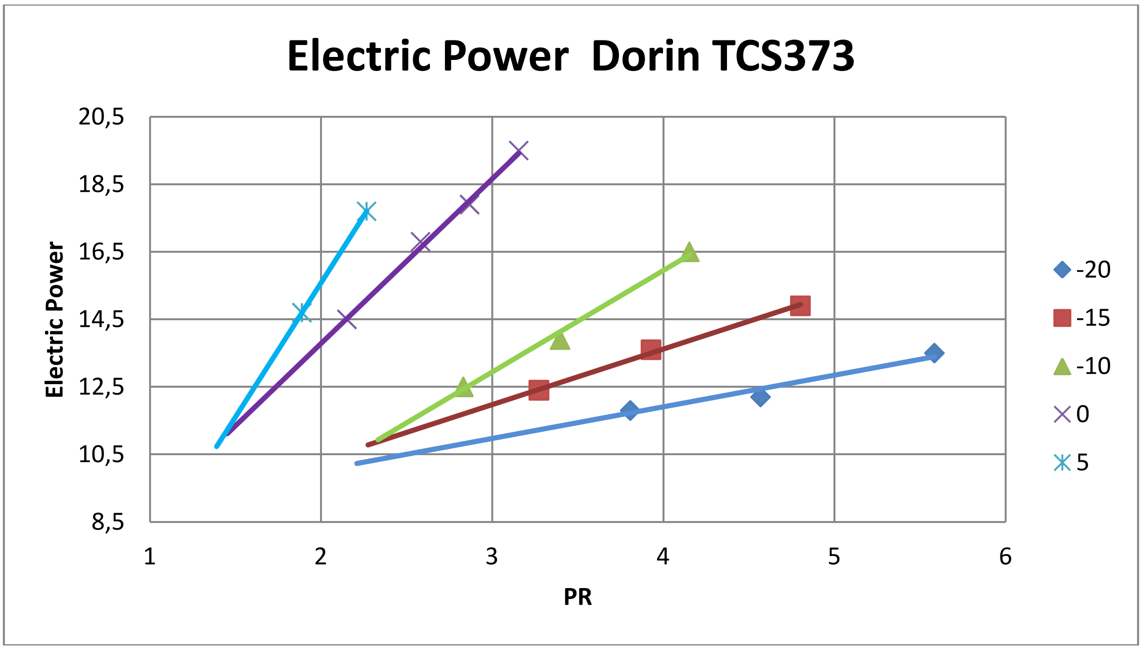


Figure 42: Electric power Dorin compressor TCS373-D CO2

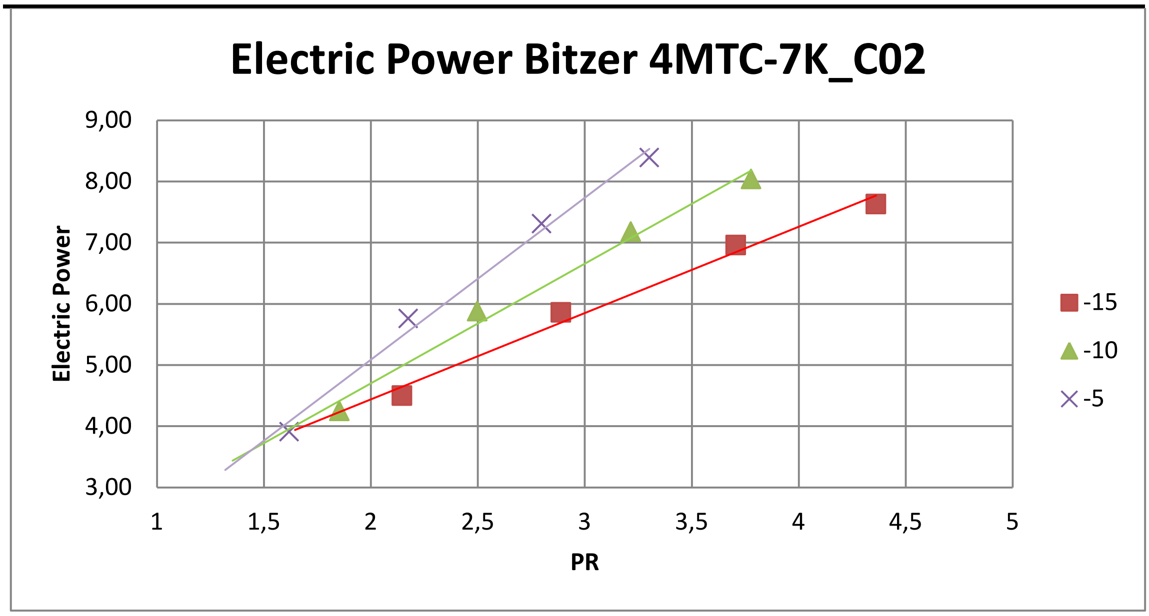


Figure 43: Electric power Bitzer compressor 4MTC-7K CO2

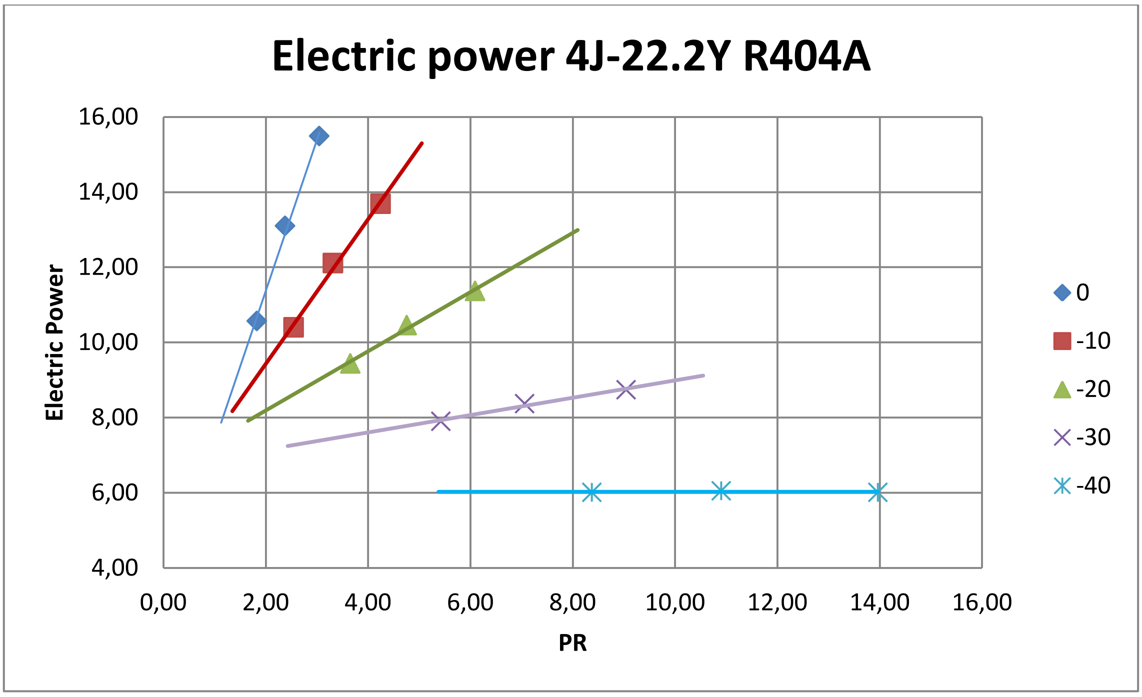


Figure 44: Electric power Bitzer compressor 4J-22.2Y R404A

It is possible to see the linear correlation and the change in the slope when the evaporation temperature is modified. For low evaporation temperatures, the high pressure ratio drops the volumetric efficiency and consequently the mass flow. The electric power remains constant when the pressure ratio is increased due to the reduction of mass flow.

Using this information it's possible to estimate the electric power consumption per compressor running in parallel measuring the inlet and outlet pressure in the compressor.

With the measurement of the high and low pressure around the compressor it is calculated the PR and the evaporation temperature. For example, if the evaporation temperature in the system is -8°C it is possible to interpolate between the lines 0°C and -10°C.

$$\frac{\dot{E}_{0^\circ C}-\dot{E}_{-10^\circ C}}{0-(-10)}=\frac{\dot{E}_{T\_evap}-\dot{E}_{-10^\circ C}}{T_{evap}-(-10)}$$

Equation 28

$$\dot{E}_{T\_evap}=0.1\cdot\left(\dot{E}_{0^\circ C}-\dot{E}_{-10^\circ C}\right)\cdot\left(T_{evap}+10\right)+\dot{E}_{-10^\circ C}$$

Owing to the correlations are linear, it is easy to parameterize the different operation points of the compressor. Knowing the pressure before and after the compressor, it is possible to estimate the electric power. It can show anomalous deviations of the theoretical electric consumption, and this parameter will be important when there are compressors with frequency control.

Other interesting plot with manufacturer information is the electric power consumption divided by the inlet density. This type of curve represents all the operation points provided by the compressor manufacturer with high correlation. Combining this curve with the density curve as a function of the evaporation temperature for 10K as superheat it can be obtained directly the electric power consumption by the compressor manufacturer with the evaporation temperature and the pressure ratio, without the necessity to interpolate.

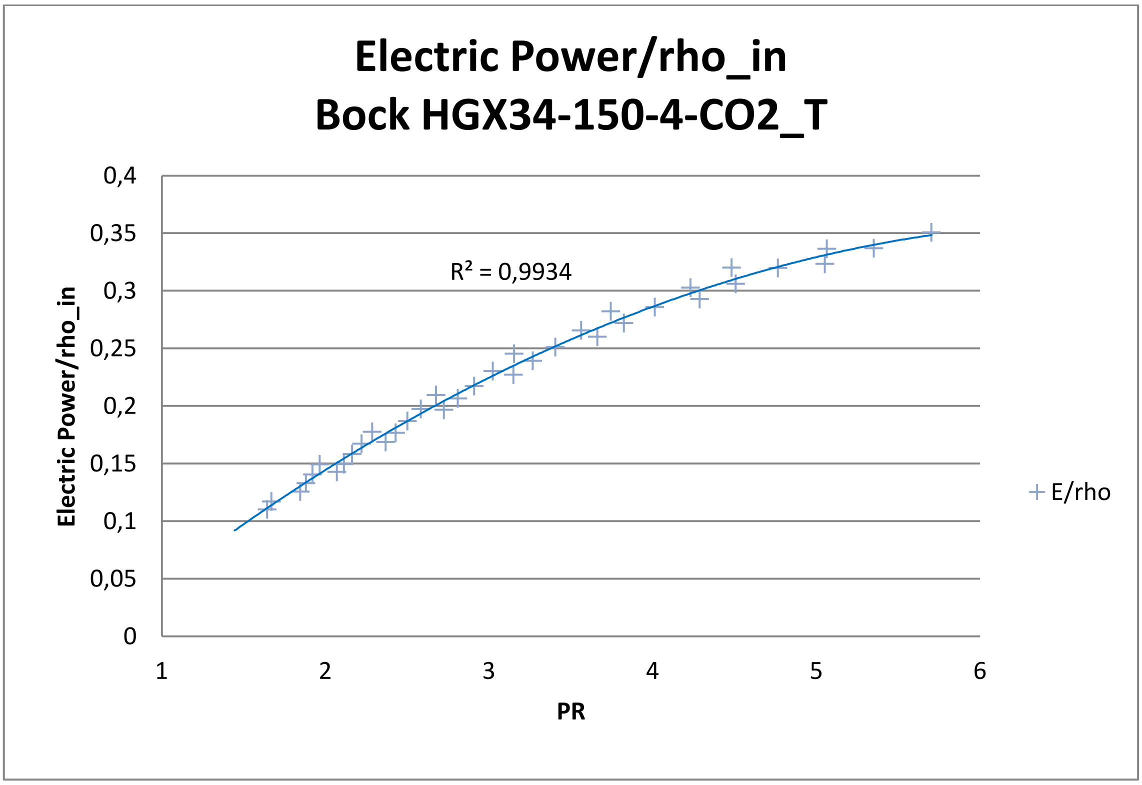


Figure 45: Electric power divided by inlet density Bock compressor HGX34-150-4-CO2

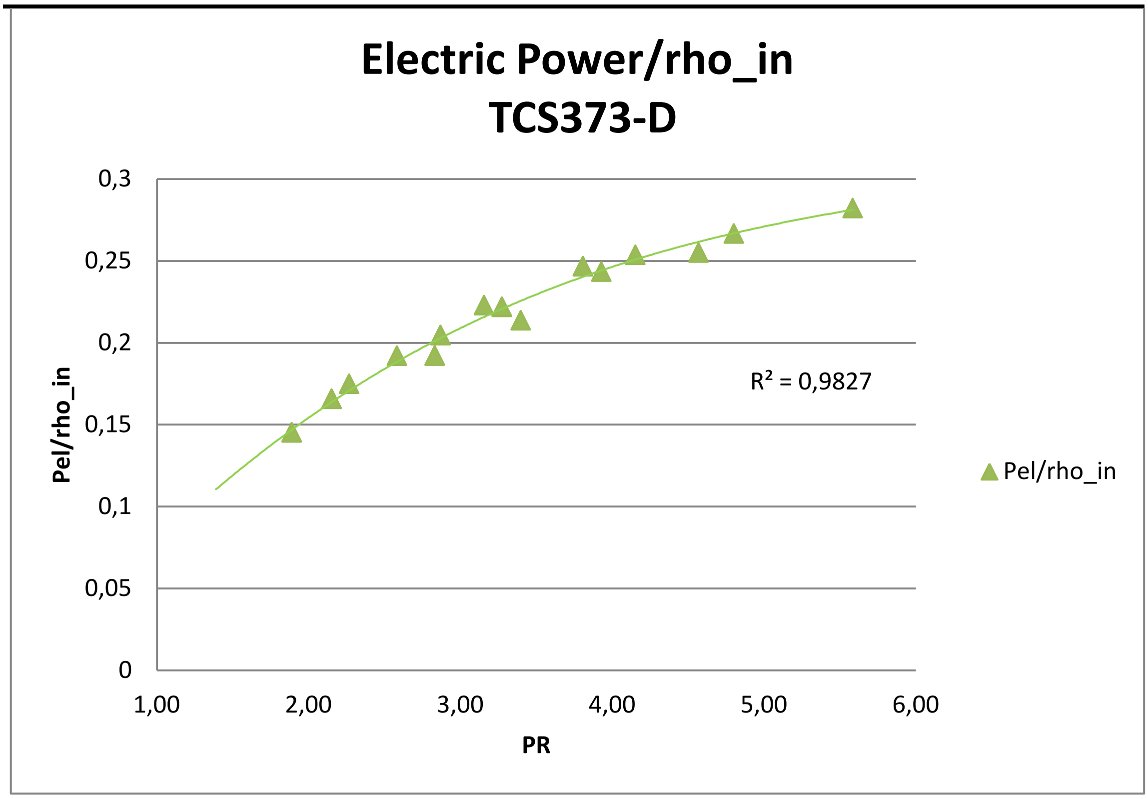


Figure 46: Electric power divided by inlet density Dorin compressor TCS373-D CO2

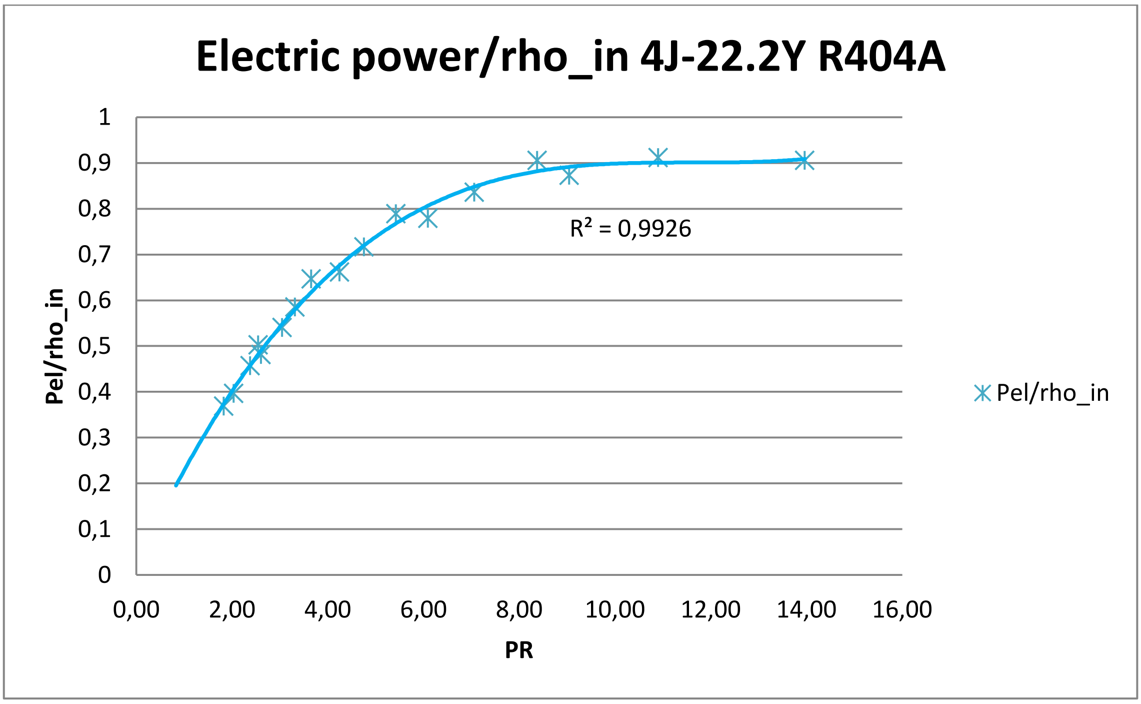


Figure 47: Electric power divided by inlet density Bitzer compressor 4J-22.2Y R404A

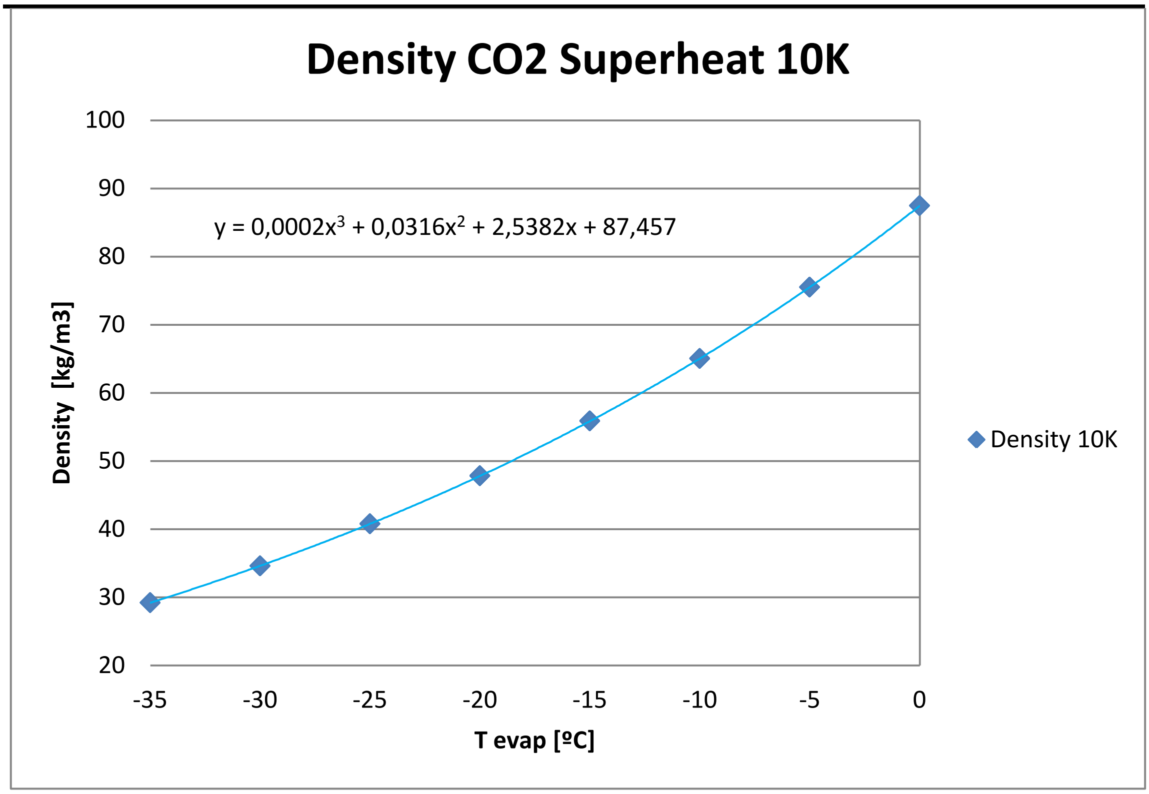


Figure 48: CO2 density for each evaporation temperature with 10K as superheat

After work with all the information about compressor it will be used the correlations in order to calculate the important parameters for the supermarket analysis.

7. Mass flow: direct measurement

One of the needed parameter to evaluate the performance of the system is the mass flow. This parameter's measurement is always a difficult process but is a key factor to obtain good results.

$$\dot{Q} = \dot{m} \cdot \Delta h$$

The different types of flow meter are named with its guidance price:

- Gas mass flow: The TGM flow meter has a price from 2,300$ (13)
- Liquid Flow Meter: Range around [500$ and 2,000$] and the Flow Monitor 500$ (14)
- Magnetic flow meter: the price for magnetic flow meter starts in 1,375$ (15)
- Ultrasonic Flow Meter FM500: Transit time/Doppler flow meter with datalogger &10 FT cable. $3,120. (16)
- Coriolis mass flow: In case of CO2, the fluid passing the mass flow meter will be liquid when the operation is sub-critical and super-critical fluid in the trans-critical operation. The flow meter will also have to operate at high pressure levels. It has been consulted two companies who provide this mass flow meters:

  "The price for a mass flow meter based on coriolis starts from around 5.200$". (17)

  "Mass flow coriolis: KCM 1500.CF-HD-F.DN25.PN4 0, 4,600$".(18)

  "Coriolis Mass flow meter C-Flow KCM 1500 Compact-Design and C-Flow Transmitter Pris 5.700$".(13)

The information was requested from different companies and they selected for the application the same model from the same mass flow manufacturer.

The range of price is a good reason why in supermarkets it isn't install mass flow meters to know the cooling capacity. We have to consider we will need a mass flow meter per refrigeration unit. In each supermarket there are around 4 units (depends on the supermarket, 2 medium temperature units and 2 low temperature units). It's preferred other methods to estimate the mass flow if we compare the cost with the price of the compressor 4J-22.2Y (6,500$), one of the compressors that has been be analyzed in the compressor manufacturer part.

8. Mass flow: indirect measurement

There are 3 methods for estimate the mass flow. Two of this methods use parameters extracted from the compressor manufacturer information, the third method is based in different studies. In the three methods it is necessary to measure the inlet and outlet pressure, and the temperature inlet of the compressor. Depends on the method the inlet variables will be used to calculate the inlet density, or the inlet enthalpy and other sensors or estimations will be needed. Each possible solution has advantages and disadvantages.

8.1. Volumetric efficiency method:

This method uses the definition of the volumetric efficiency for estimate the mass flow. The volumetric efficiency must be calculated before.

$$\dot{m} = \eta_v \cdot \dot{V}_c \cdot \rho_{in\_comp} \rightarrow \left\{ \begin{array}{c} \eta_v = f(PR) \\ \dot{V}_c \\ \rho_{in\_comp} = f(SH, T_{evap}) \end{array} \right\}$$

Volumetric efficiency
Swept volume flow rate
Inlet compressor density

Advantages:

- This method is based on information extracted directly from the operations points of the compressor, real tests.

- The volumetric efficiency presents great correlation with the PR, and it gives us higher precision.

- It doesn't depend on the heat losses by the compressor body, only depends on the inlet conditions. This is an important aspect. The ambient temperature of the machine room in supermarkets isn't always the indoor temperature. Due to the noise level or the lack of space, machine room can be added next to the supermarket. This space is not well insulated and in cold climates like Sweden the temperature variations may be important. This temperature will have influence the heat losses, the lower temperature, higher heat losses.

  In some of the supermarkets analyzed is measured the machine room temperature. The average machine room temperature for a week in February is 16°C and for a week in July is 30°C. The compressor heat losses will be different in each period.

- The density is calculated with the inlet compressor conditions.

The following Figure represents the inlet and outlet pressure and temperature for the compression unit in the medium temperature cooling circuit KA1 in the supermarket TR1. The outlet conditions present high fluctuations while the inlet conditions are more constant. The inlet and outlet pressure and temperature are the parameters needed for the mass flow indirect measurements. An estimation considering the compressor inlet conditions will be more accurate due to the greater stability of these variables in supermarkets.

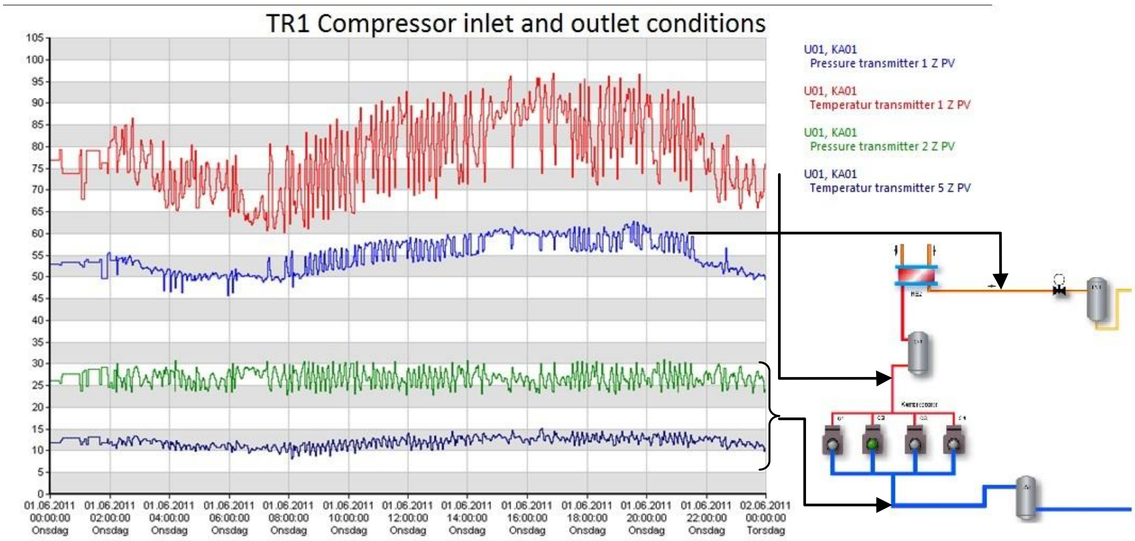


Figure 49: Inlet and outlet conditions for the compressor unit in CO2 system TR1

Disadvantages:

- For the swept volume flow rate it's necessary to know the number of compressors running.

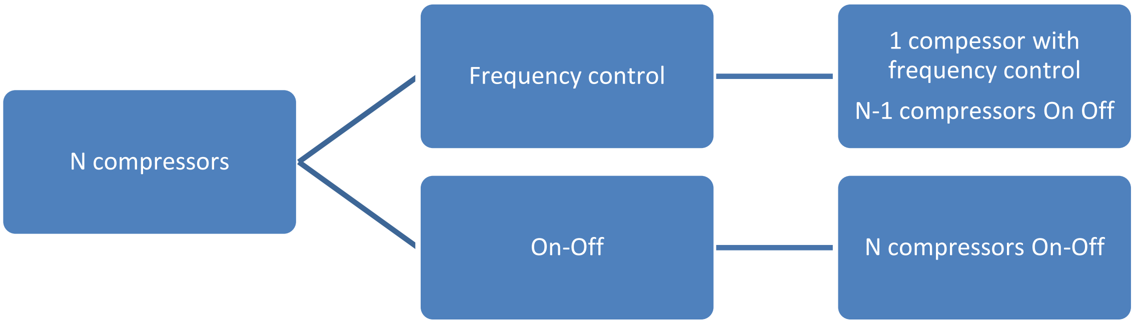
This number will be an integer in systems without frequency control, and a decimal number in systems with one of the compressors controlled with frequency control. The next equation shows how to calculate the number of compressor running if there are direct measurements of the feeding frequency and how many compressors are running.

$$\dot{V}_{s\_total} = \dot{V}_s \cdot \left(N_{compressors\_on-off\_running} + \frac{f}{50} \right) \qquad \textbf{Equation 29}$$

It is not a common procedure to measure the number of compressors running. Instead the electric power consumption of all the compressors in a certain unit is measured.

8.1.1.Number of compressors running

One of the important parameter in the volumetric efficiency method is the number of compressor running in parallel at the same time. There are two possibilities to study depending on the control of these compressors.



In the above scheme is represented the different possibilities about the control of the compressors in refrigeration systems in supermarkets that has as a consequence different methods to estimate the number of compressors running.

8.1.2. On-Off compressors

When the refrigeration unit has on-off compressors it is possible to estimate the number of compressors running through different variables registered in each refrigeration systems. In the following images are showed some examples.

The first image comes from cascade systems. In this system the number of compressors running was estimate with the percentage of open of the expansion valve. In the figure is represented the opening of the expansion valve in three cooling units for a week. The opening rage for one compressor or two depends on the expansion valve. We need to plot first this variable in order to know which range is for one compressor and which for two compressors.

The curious thing about this system is it records the opening of the valve and not important parameters such as pressures or power consumption.

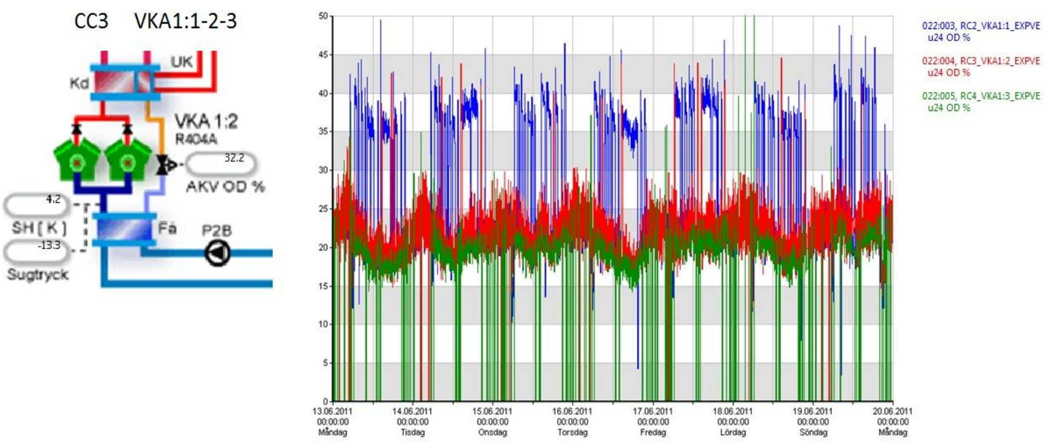


Figure 50: Cascade system CC3

In the following figure is represented the compressor capacity [%] and the electric power consumption [kW] in a cascade system for 2 hours (29/06/2011). In this system is possible to download a variable that indicate the number of compressors running. There are 5 compressors in parallel, and the variable adds 20% to the value when a compressor starts. In this case is possible to obtain the same information from the electric power consumption measurement.

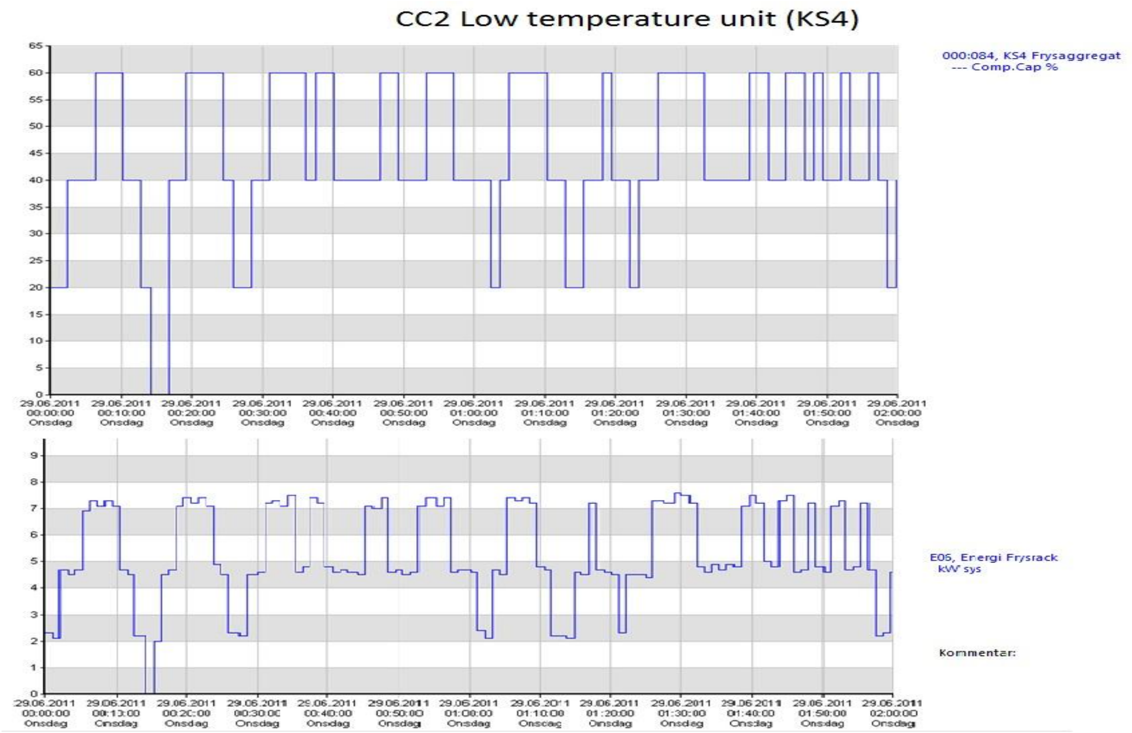


Figure 51: Cascade system CC2, electric power and number of compressors running

In the above image is plotted the electric power consumption in TR1 in different units, KA1 and KA2. KA1 has on-off compressors, and KA2 has 1 compressor with Frequency control showing different electric curves. While in KA1 the power change in steps, in KA2 the electric power can change gradually.

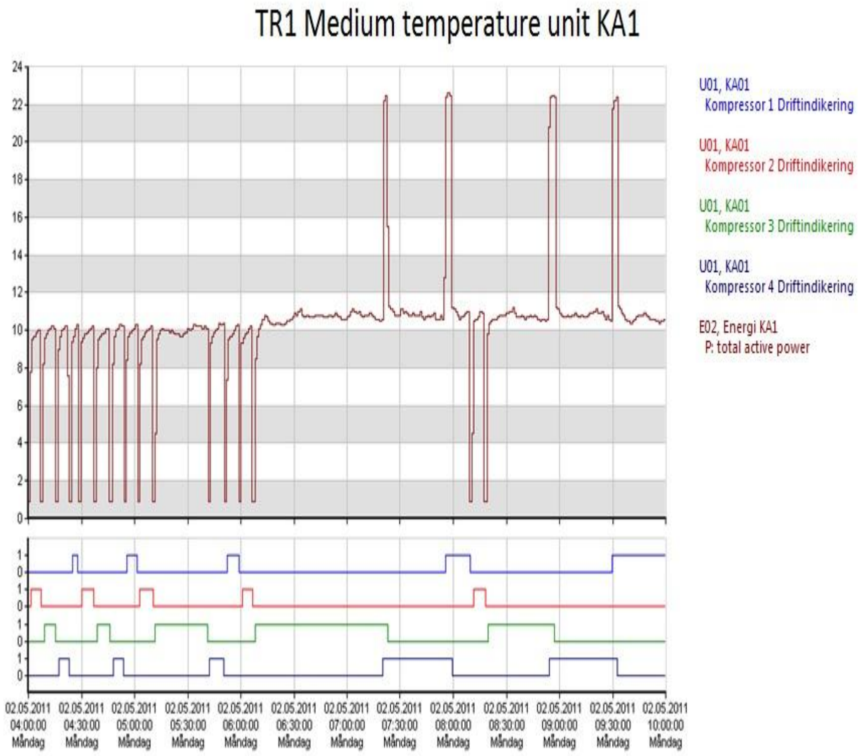


Figure 52: Parallel system TR1, KA1, electric power and number of compressors running

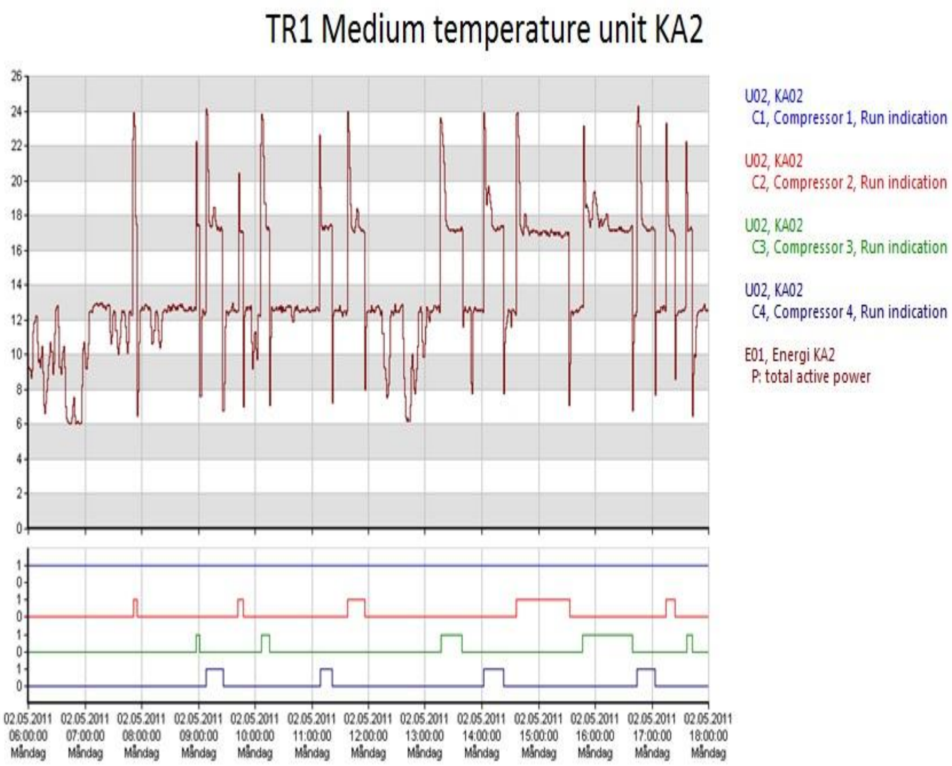


Figure 53: Parallel system TR1, KA2, electric power and number of compressors running

Although for each cooling unit there is a variable in each compressor that indicates when the compressor is running or not, in previous projects never uses this variable. The number of variables to download is too big, and is preferred estimate this parameter from the electric power consumption instead of download 1 variable more per compressor, in total around 12 new variables. If one of our variables measured is the power consumption by the compressors, the result is showed in the next plot:

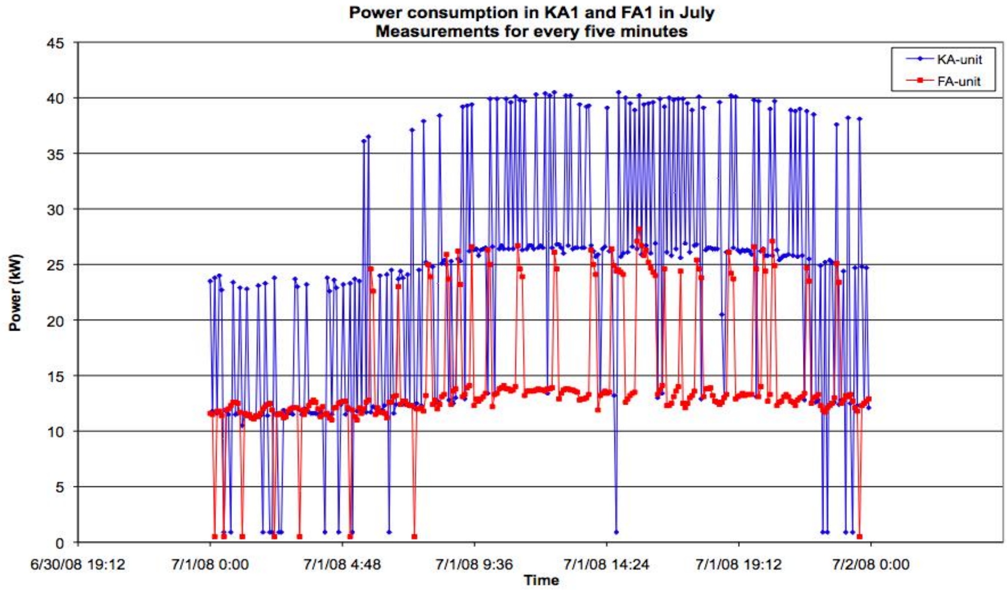


Figure 54: Compressor electrical power measured for one day in July 2008 (01.07.08) in TR1 Supermarket. (6)

In the above plot can be determined the number of compressors running at the same time with adequate accuracy.

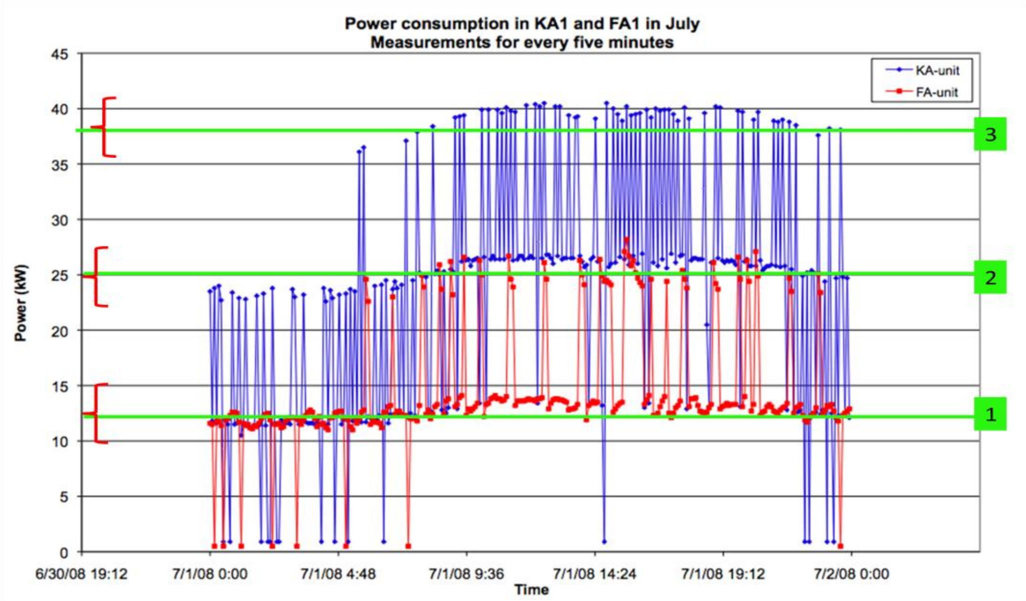


Figure 55: Definition range for number of compressor running

The solution was to define correct intervals, in this case can be [2-15-30] kW. This was the method used and was introduce in the templates for estimate the number of compressors running.

8.1.3. Frequency control compressors

If in the system there is a compressor with frequency control it is necessary the nominal power per compressor for estimate the number of compressor running. If the compressor is fed with 40Hz, the electric power consumption will be 40/50*$P_{nominal}$ and the number of compressor running will be 40/50=0.8. The electric power consumption divided by the nominal power gives the same result.

8.1.4. Conclusion

The conclusion of this part is the need to know the nominal power per compressor for the operating conditions in the systems in order to know the number of compressor running and estimate the mass flow if the volumetric efficiency method is used. But the nominal power deepens on the evaporator and condenser pressure as it has been showed in the compressor manufacturer part. It is necessary the electric power consumption curves.

From the analysis, it is suggested the use of the total power consumption of the compressor unit. Each group of compressors operating in parallel in every unit must be measured separately.

The next two methods have the same advantage. They obtain directly the mass flow, but they need the electric power consumption of the compressors unit. This measurement is not a problem because it will be needed for the calculation of the COP.

8.2. Total efficiency method:

This solution proposes to use the definition of the total efficiency for estimate the mass flow:

$$\dot{m} = \frac{\eta_{tot} \cdot \dot{E}_{electric\_power}}{\Delta h_{is}} \rightarrow \left\{ \begin{array}{c} \eta_{tot} = f(PR)\_or\_f(PR, T_{evap}) \\ \dot{E}_{electric\_power} \\ \Delta h_{is} = f(SH, T_{evap}, T_{cond}) \end{array} \right\}$$

Total efficiency
Electric power
Isentropic enthalpy differences

Advantages:

- This method use inlet conditions of the compressor.
- It doesn't need the number of compressors running since it is implicit in the power consumption
- No matter how the compressors are controlled, on-off or frequency controlled because this control will produce modifications in the electric power consumption and these modifications are contemplated in the method.

Disadvantage:

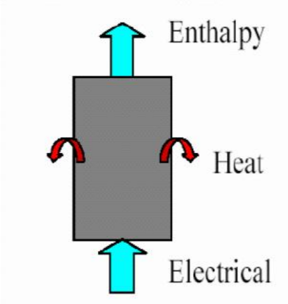
- Previously it has been showed the precision of the total efficiency correlated with all the points. The correlation between the total efficiency with the pressure ratio has more scatter. The possible errors in the estimation using the total efficiency correlated for all the points is around +/-5% respect the calculated efficiency, and the relative error it's about (+/-5%)/63%=+/-7.9%. The average total efficiency is 63%.

The solution to increase the precision of this estimation is to correlate all the points for each evaporation temperature and proceed in the same way as the electric power consumption. Knowing the evaporation pressure (or temperature) and it's fluctuation, and using the two polynomial curve for the evaporation temperature that contain the mean evaporation temperature, it is possible to interpolate calculating the total efficiency for this point.

As we have seen the most precise method is using the correlation for the volumetric efficiency but to complete the estimation of the mass flow is necessary to know the number of compressors running. If we want good results it is the other important parameter. The objective is not to lose with the number of compressors running the accuracy gained with the volumetric efficiency.

8.3. ClimaCheck method:

Other solution is to use the method from ClimaCheck. This method consists of consider the compressor like a black box:



"The necessary parameters can be calculated by the following equation:

$$\dot{m} = \frac{\dot{E}_{electric\_power} - \dot{Q}_{losses}}{\Delta h_{compressor}} \rightarrow \left\{ \begin{array}{c} \dot{E}_{electric\_power} = Electric\_Power\_measured \\ \dot{Q}_{losses} = Heat\_losses\_body \\ \Delta h_{compressor} = h_{out\_comp} - h_{in\_comp} \end{array} \right\}$$

The heat losses from the compressor in above equation are assumed to be around 7% of the electrical input, as previous studies have established that the heat losses for hermetic and semi hermetic compressors lie in the range of 3 to 10% (19). This assumption is mostly precise since 50% increase or decrease in this value would result only in 3 to 4% error in the measurements according to the heat rejection rates established by Asercom (20). The conclusion from this is that the compressor can be used as a for field condition accurate "flow meter" for a refrigerant circuit." (21)

Advantages:

- It is an easy method to apply.
- It does not need a specific compressor study.

Disadvantages:

- It's an assumption and the real losses fluctuate. It is necessary to check this method. The differences are the refrigerant used. In case of CO2 the discharge temperature is higher than R404A. If there are more losses to the ambient, the outlet enthalpy of the compressor will be lower and the opposite. For these reason considers the same body heat loss for all the compressors doesn't seem reasonable.

- It is good for sight evaluation but not for detailed system analysis

The most common method used in this project is the volumetric efficiency.

8.4. Comparison between the methods: experimental measurements of CO2 heat pump test rig

The three methods was used to estimate the mass flow and compare with the real mass flow measured using the information provide for Zahid Anwar and Yang Chen about "Experimental measurements of CO_2 heat pump test rig". (22)

In this experiment was used a semi-hermetic compressor from Dorin TCS362-4D with 5.4 m^3/hr swept volume at 1450 rpm. The characteristic curves are showed in the next two plots:

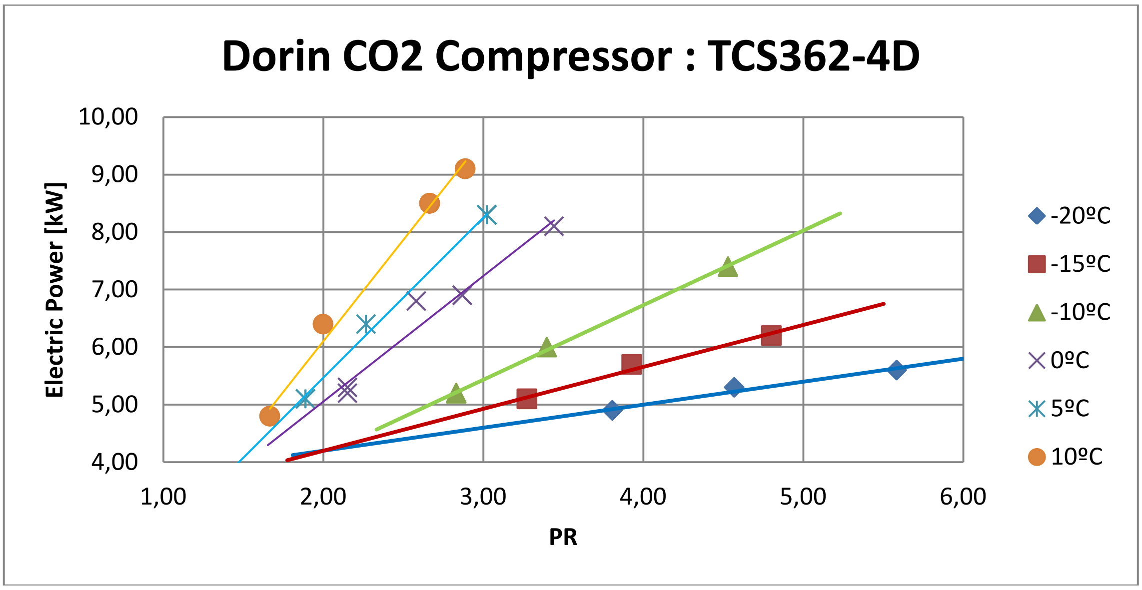


Figure 56: Electric power Dorin compressor TCS362-4D

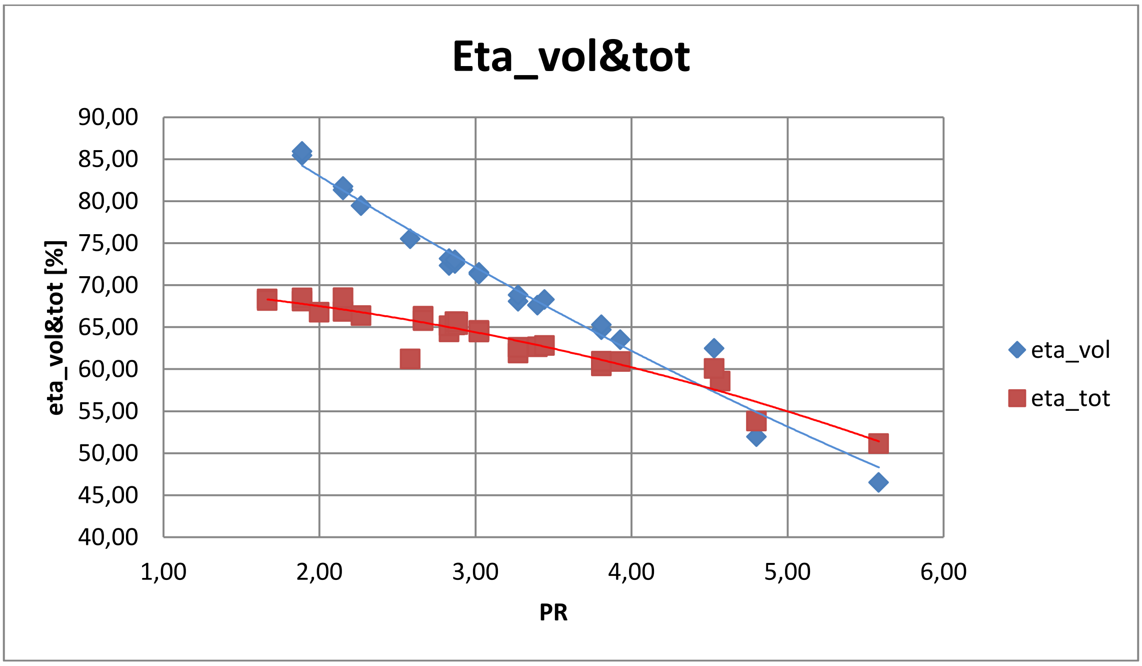


Figure 57: Volumetric and total efficiency curves Dorin compressor TCS362-4D

It was used a C-Flow coriolis mass flow meter (KCM 600) for CO_2 mass flow measurement. Temperature measurement was done with Pt-1000 element from Danfoss.

8.4.1. System Layout

Division of applied thermodynamics at Energy department KTH has a prototype of CO_2 heat pump, this is vapor compression cycle based water to water heat pump with nominal heating capacity of 30kW. The schematic diagram of the system is shown in the following plot:

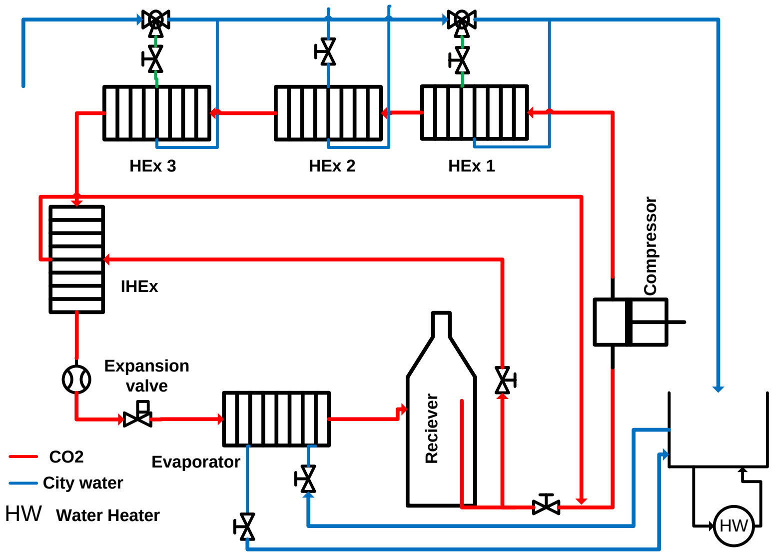


Figure 58: Schematics of the heat pump at KTH (23)

It is composed by a compressor, gas cooler, expansion device and evaporator. Plate heat exchangers are used.

8.4.2. Methodology

System was operated at fixed evaporation temperature and gas cooler side pressure was varied by changing the compressor speed (from 1050-1800 RPM) and adjustment of expansion valve. Reading for temperatures, flow rate, pressures were taken when the system get stable.

It was run four tests with the evaporation temperature: 0°C, 5°C, 0°C, -5°C. There were all the needed parameters for make the comparison between the three methods. It was measured the inlet and outlet temperature and pressure, and was measured the mass flow with a flow meter. Due to the accuracy limitation of the data acquisition equipment it was read the value manually and also calculated the mass flow based on energy balance and compressor data.

8.4.3. Testing results

In the following figure is represented the mass flow direct measured and the mass flow calculated using the three methods: Climacheck, total efficiency and volumetric efficiency. It is indicated the 4 test with different evaporation temperature, A=0°C,B= 5°C, C=0°C,D= -5°C. The test A aims at analyzing the heat recover in the gas cooler varying the water set point in the gas cooler but it has not influence in the mass flow analysis.

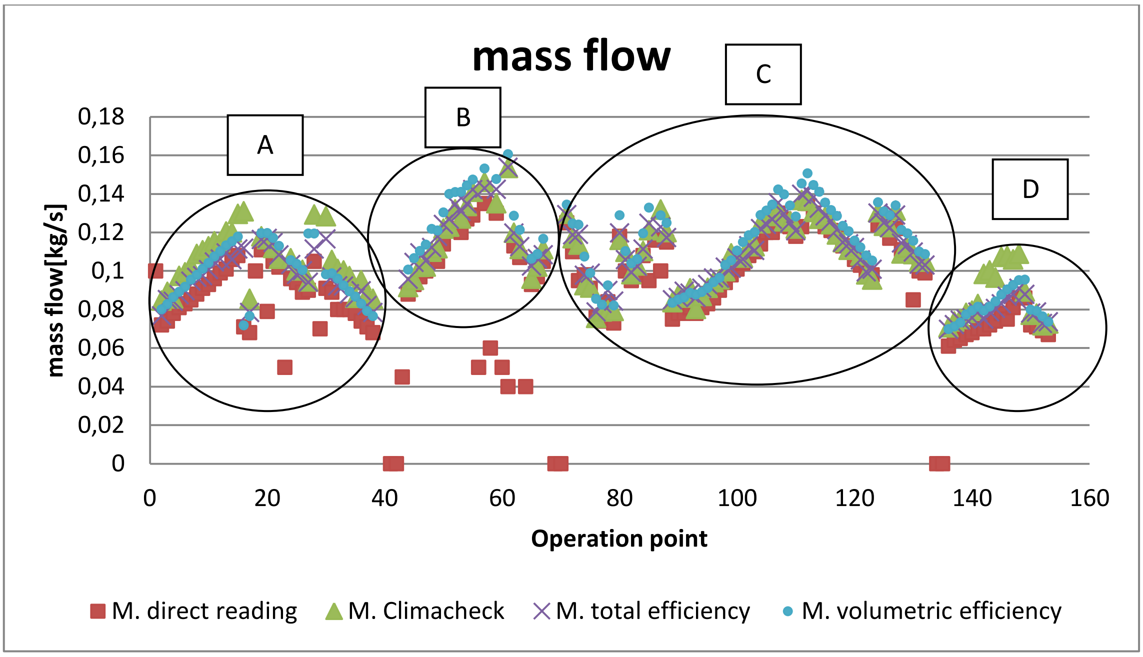


Figure 59: Mass flow estimation and measurement

The correlation of all the methods is a priori good although there are some differences. From the test information is possible to obtain the total efficiency, the volumetric efficiency and the heat losses since it is measured the mass flow, the electric power, rpm, the inlet and outlet pressure and the inlet and outlet temperature from the compressor.

When we start to analyze a new supermarket the only information that would be available is the compressor manufacturer information. We would obtain the total and volumetric efficiency curves and we would estimate the mass flow. But in this test the mass flow is directly measured hence we can compare the real measurement, with the estimations.

The differences between the three methods come from the differences between the parameter that names the method: total efficiency (calculated and from manufacturer), the volumetric efficiency (calculated and from manufacturer) and the heat losses (calculated against 7%).

If the total or volumetric efficiency calculated from the test information correspond with the parameters calculated from manufacturer information, the mass flow estimation will correspond with the measured mass flow. If the heat losses calculated in the test are 7%, the mass flow estimated using the Climacheck method will correspond with the measured mass flow.

The following plots show the differences between the total and volumetric efficiency, and these parameters from compressor manufacturer, calculated from the curves and the pressure ration for each operation point.

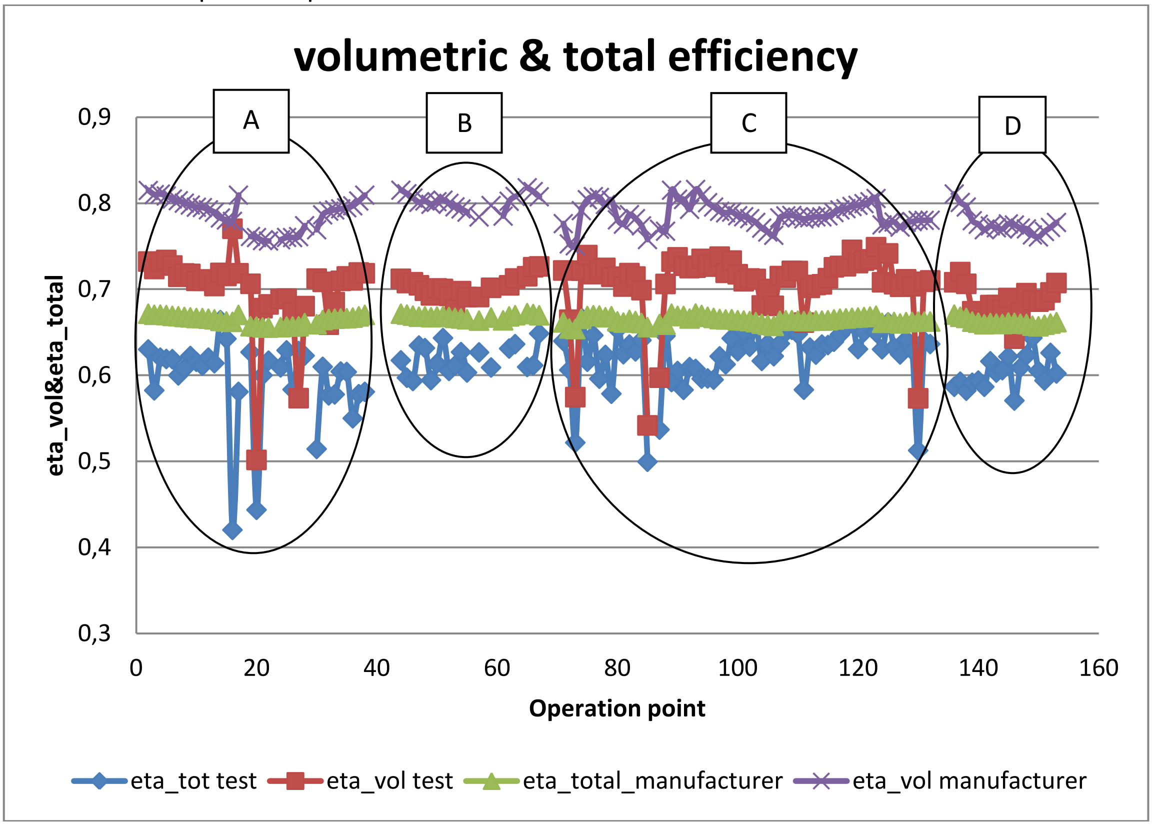


Figure 60: Total and volumetric efficiency, form manufacturer and from test measures

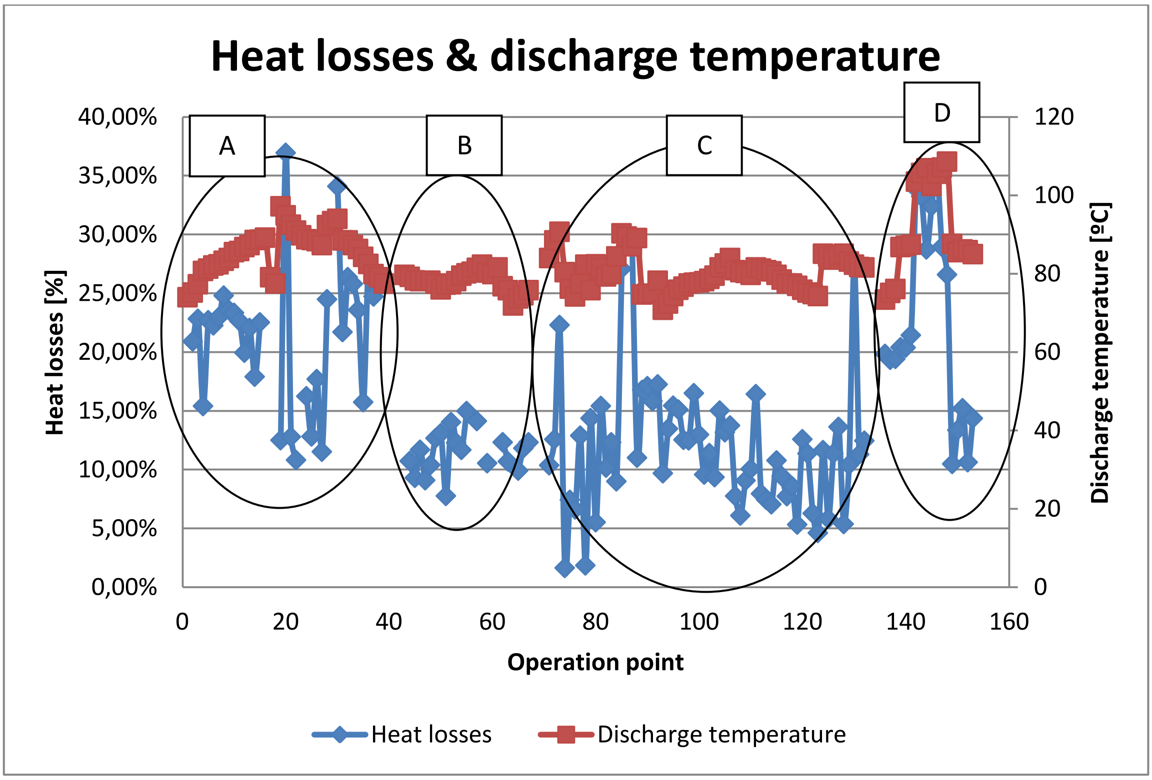


Figure 61: Heat losses calculated, compressor discharge temperature

There are 4 different tests, changing the evaporation temperature, and the result has been grouped for each test. For each operation point has been calculated the total efficiency, the volumetric efficiency and the heat losses. And using the pressure ratio has been calculated the volumetric efficiency and the total efficiency from compressor manufacturer information. It has been average all the operations point for each test.

If it is calculated the different mass flows and compared with the measured mass flow, the differences comes from the differences in the total and volumetric efficiency (compressor manufacturer information with respect to the calculated from the operation points in the test).

The following figure shows for each test the differences in the mass flow with respect to the measured mass flow, using the total, the volumetric or the Climacheck method for the mass flow estimations. For example, in the test A, the mass flow calculated using the volumetric efficiency overestimates the mass flow 12.56%; the mass flow calculated with the total efficiency method overestimated the mass flow 11.6%; the mass flow calculated using the Climacheck method overestimates the mass flow in 18.5%.

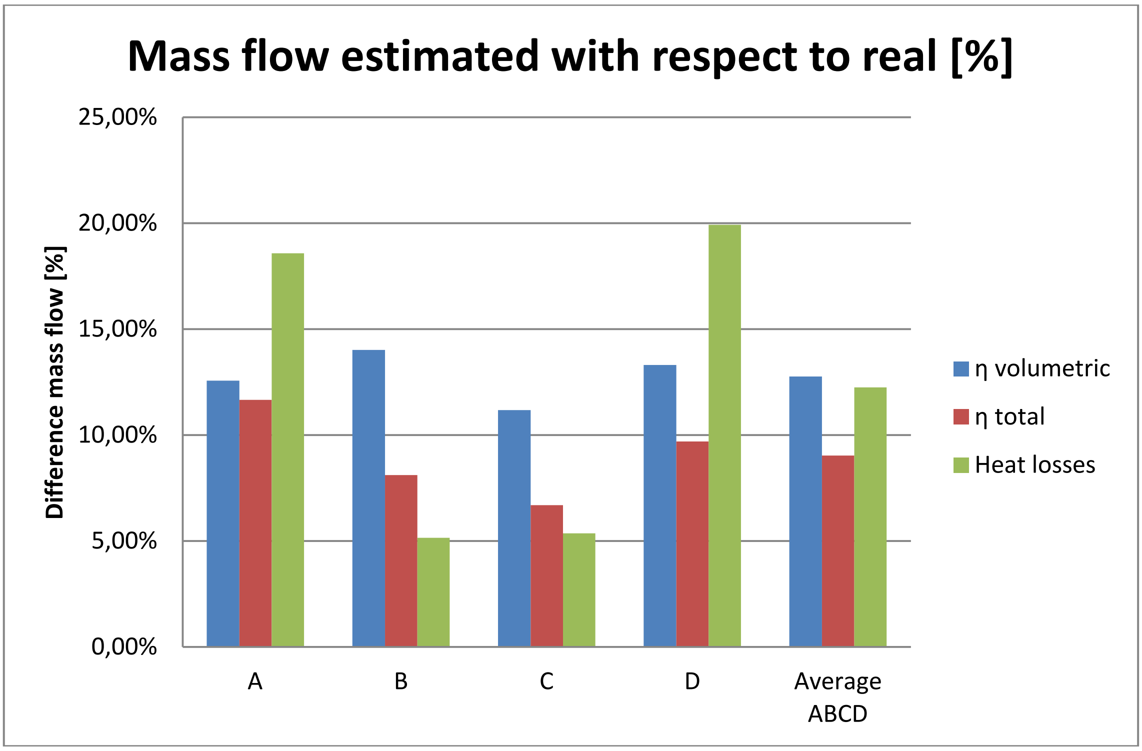


Figure 62: mass flow estimated with respect to real mass flow for each mass flow estimation method

The first conclusion is that the 3 methods overestimate the mass flow. The information from manufacturer always is the best results, with the compressor in the optimum conditions.

The real results of the cooling capacity using the volumetric efficiency and the total efficiency may have an uncertainty of 10%, and the same for the COP since it is directly proportional. The conclusions between mass flow and total efficiency are that can give as similar result, with similar accuracy although in this test the total efficiency method is 3% closer than the measured mass flow.

On the other hand for the ClimaCheck method the variations of the heat losses measured have high correlation with the Exit temperature of the compressor.

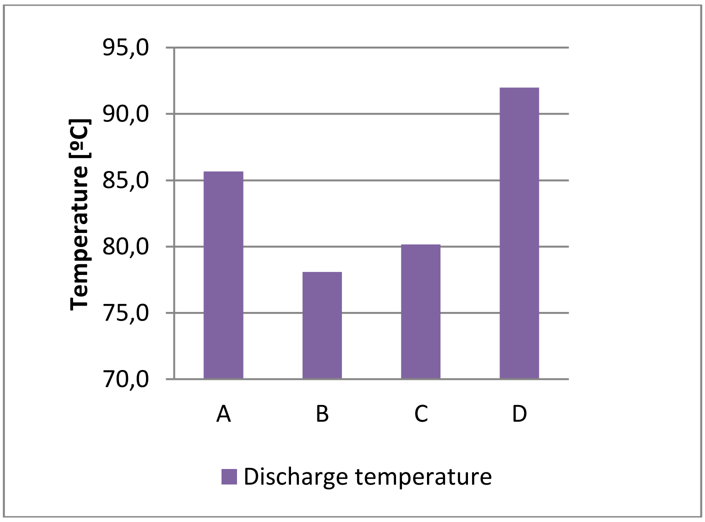


Figure 63: Compressor discharge temperature, average for each test

For the last test, D, there are some points that present higher superheat than others in the compressor inlet. This superheat results in an increase in discharge temperature and the result is higher compressor heat losses. The operation points 7 to 14 present around 15K of superheat higher than other points. Consequently the discharge temperature is around 20K higher. It produces higher heat losses seeing the figure.

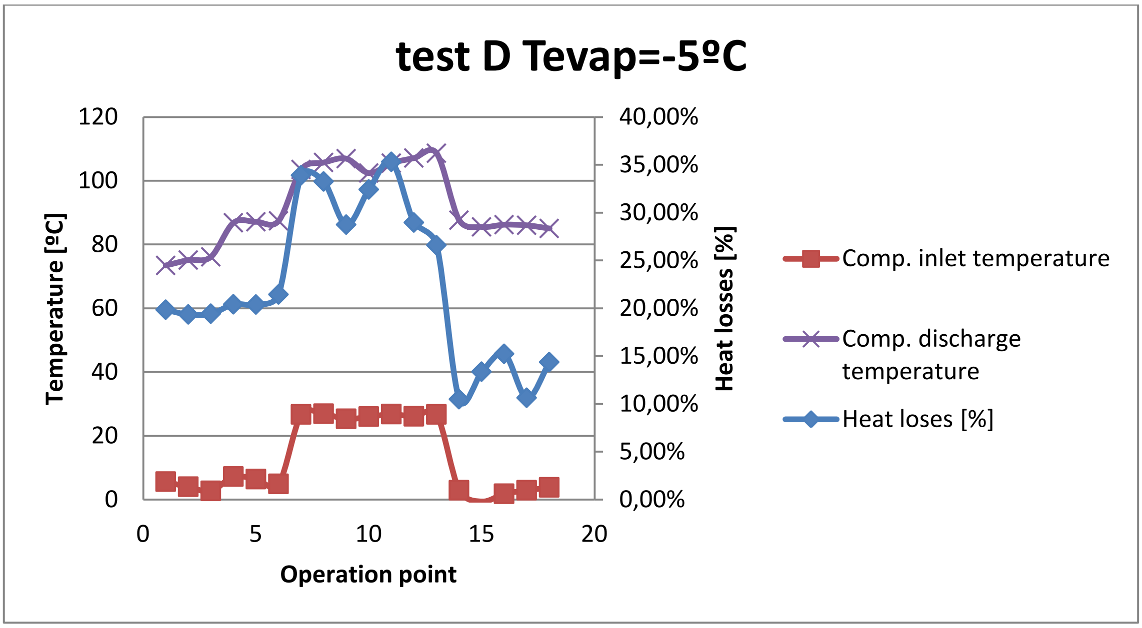


Figure 64: Test D inlet and discharge temperature and heat losses

It is possible to conclude the ClimaCheck method probably is good for conventional refrigerants, but the high discharge temperature in CO2 advice to review the assumption of 7% as heat losses. In this test the heat losses fluctuate between 10% and 20%. The heat losses fluctuates a lot and this is another reason to question the method

9. Internal Superheat

One of the parameters needed to determine the cooling capacity extracted from the cabinets in the supermarket is the outlet temperature in each cabinet. In the refrigeration systems there are a useful superheat (internal superheat), produced inside the cabinet. This superheat is taken into account as a cooling capacity. Between the exit of the cabinet and the inlet of the compressor is produced an external superheat, due to the temperature difference with the ambient. This effect is more evident in direct expansion refrigeration circuits with long pipes.

The result of this effect is installation of a temperature sensor in the compressor inlet, and one temperature sensor in the exit of each cabinet in order to know the enthalpy difference through it. The most accurate method to estimate the cooling capacity in each cabinet would consist of know the mass flow and enthalpy differences and with the mass flow calculate the cooling capacity delivered to each cabinet. But if it is estimated the mass flow in each cooling circuit due to the absence of sensors, it is not consistent to try to estimate the mass flow in each cabinet, from a previous estimation.

9.1. Proposal in previous supermarket's analysis:

The method used to estimate the exit temperature in each cabinet is the direct measurement. In the different systems are installed a temperature sensors in each cabinet and it is registered when a change in its value is produced. All the cabinets are treated as a single, in the same way that all the compressors in parallel are analyzed like a compression unit. The solution taken to know the enthalpy difference through the cabinets is to average the temperature outlet in each cabinet for all the cabinets supplied with the same compressor unit.

For each month and each cooling unit is download the exit temperature in each cabinet. This file is normalized in time, and is average, each 5 min, in order to obtain an estimate as close and accurate as possible to estimate the average enthalpy difference in the cabinets, and with the mass flow estimated with other methods, calculate the cooling capacity.

This method implies:

1. Temperature sensor placement in the refrigerant outlet in each cabinet.
2. High number of variables to store.
3. Download the data.
4. Normalization of the data.
5. Average for all the cabinets in each unit.

9.2. Applied in previous supermarket's analysis:

Due to the IWMAC's supermarkets have available all the outlet temperature in each cabinet, the above method could be applied. In systems where each cabinet is easily linked with a cooling unit, the process was as described above. But in other works, the study has been done selecting 5 low temperature and 5 medium temperature cabinets, taking their average outlet temperature in order to estimate the cooling capacity in all the cabinets in this cooling unit. It is assumed that the behavior of the other cabinets will be similar to these 10 cabinets, and the results can be extrapolated.

9.3. New assumption:

Each cabinet has its own regulation, and the superheat is controlled by the expansion valve. The thermostatic expansion valves are essentially reducing valves between the high-pressure side and the low-pressure side of the system. These valves, which are the most used devices, automatically control the liquid-refrigerant flow to the evaporator at a rate that matches the system capacity to the actual load. Once a valve is properly adjusted, further adjustment should not be necessary.

Owing to the expansion valve regulates around the set point superheat, it can be expected an oscillatory behavior around the desired superheat. If this behavior corresponds to a real behavior, a possible assumption could be to take the reference point as outlet temperature in the cabinet. That's why it has been checked the validity of this hypothesis.

The cabinets, supermarkets and study periods chosen for the analysis are in the next table:

SYSTEM	UNITS	CABINETS	STUDY PERIODS
TR1	KA1	MT_Cab_19.3	1-Mar-2011 to 12-Jun 2011
		MT_Cab_25.1	May-2010
	KA2	MT_Cab_30.5	1-Jun-2010 to 31-Aug-2010
		MT_Cab_32.2	
		LT_Cab_44.1	
	FA1	LT_Cab_45.2	
		LT_Cab_46.1	
TR2	KA3	MT_Cab_DK20.1	7-feb-2011
	KA3	MT_Cab_DK21.1	6-Jun-2011
	FA1	LT_Cab_DF34.1	7to13Feb-2011
	KA1	MT_Cab_RK3.1	6to12Jun-2011
TR3	KA2	MT_Cab_K1	7-feb-2011
	FA2	LT_Cab_F69	6-jun-2011
	FA1	LT_Cab_F59	7to13-Feb-2011
	KA1	MT_Cab_K2	6to12-Jun-2011
CC3		LT_Cab_K31.A	7to13-Feb-2011 and 7-Feb-2011
		LT_Cab_K32.A	6to12-Jun-2011 and 6-Jun-2011

Table 10: Systems and cabinets analyzed

The choice was made with the main objective to obtain representative results. In the different supermarkets were selected low and medium temperature cabinets, and in systems with two medium and two low temperature units, one cabinet from each unit.

The different study periods in the first supermarket were 3 months in summer, 4 month before summer and 1 month from the last year. In this study was considered that average value for 3 months would be too long period. As the graphs and comparisons to see the evolution of a system takes monthly values, for the next for the following studies it was decided to reduce the period of time. The following periods were one day in February and one in June, and a whole week during these months.

9.4. Results of the comparison:

In the following graphs are showed the superheat set point and the superheat measured in each cabinet for the systems TR1 and TR2. It is represented

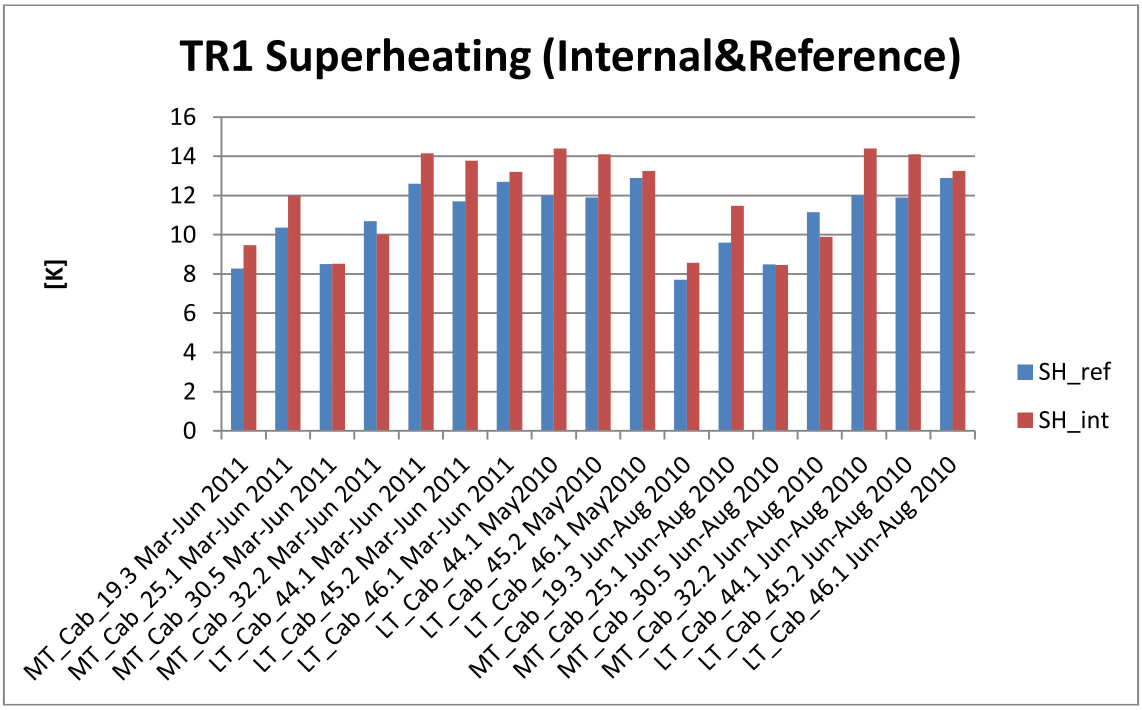


Figure 65: TR1 internal and reference superheat

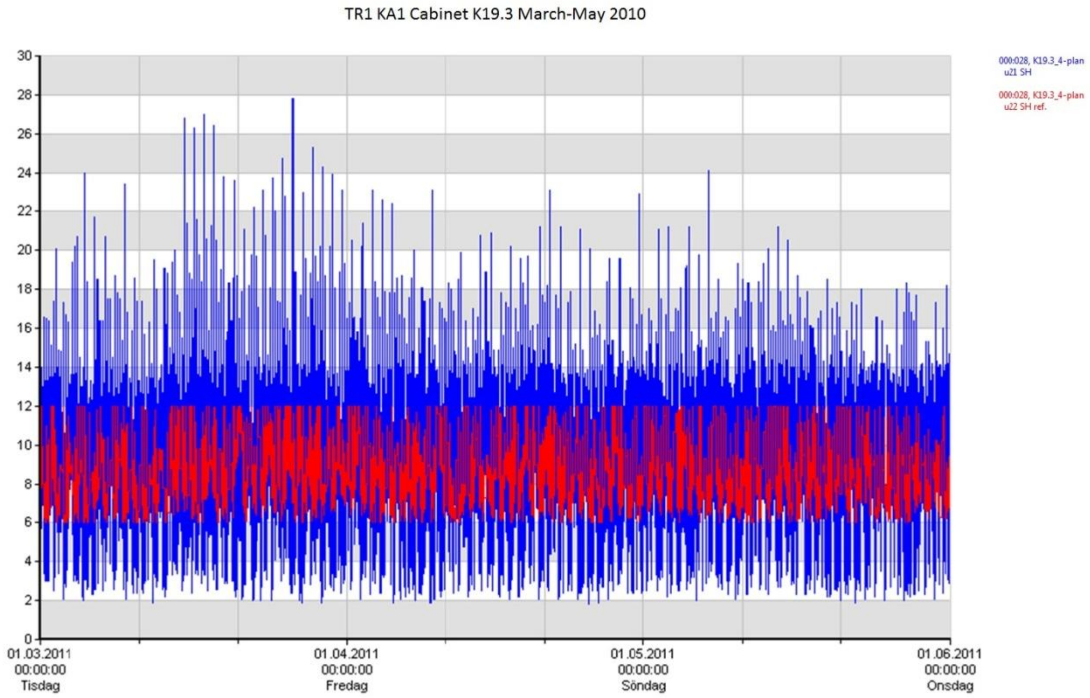


Figure 66: TR1, KA1 Cabinet K19.3,internal(blue) and reference superheat(red). 3 month March to May 2011

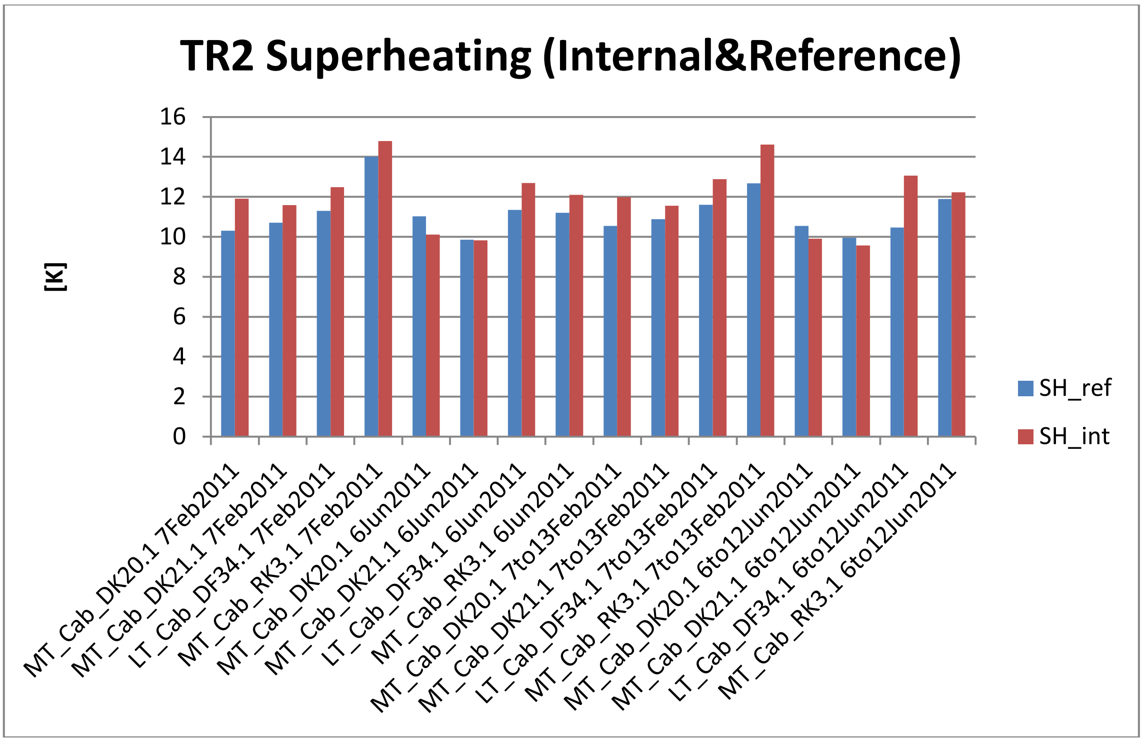


Figure 67: TR2 internal and reference superheat

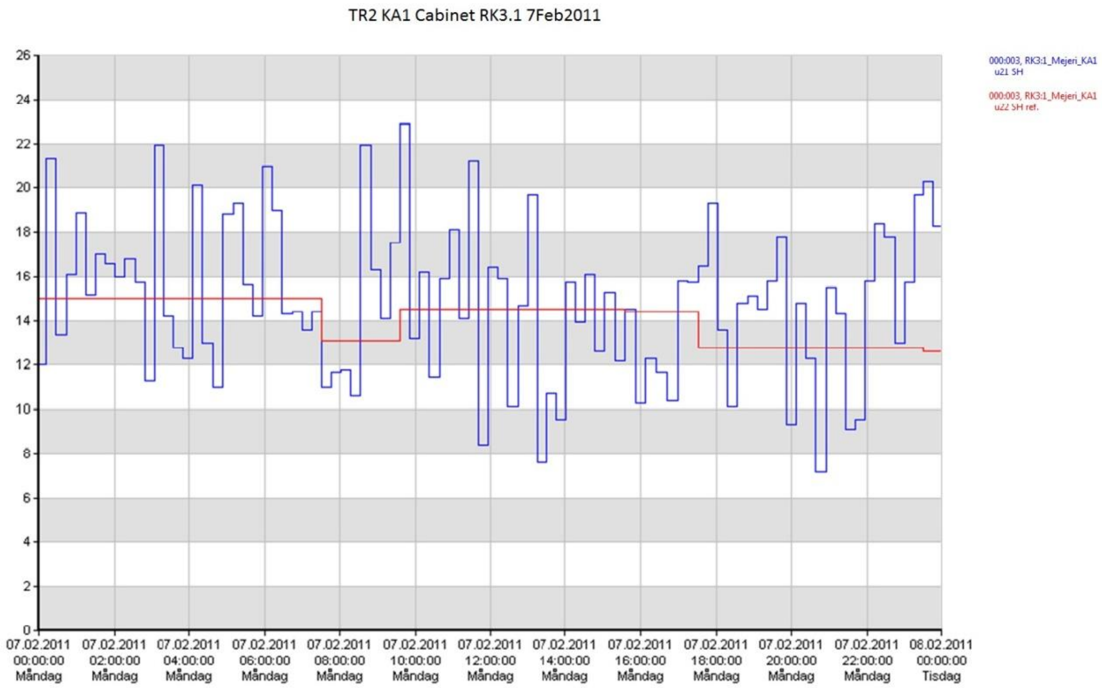


Figure 68: TR2, KA1 cabinet RK3.1 internal (blue) and reference superheat (red) 1day 07/02/2011

The internal superheat presents and oscillatory behavior around the superheat set point.

The average superheat set point and superheat measured for each systems and each study period is showed in the following bar graph:

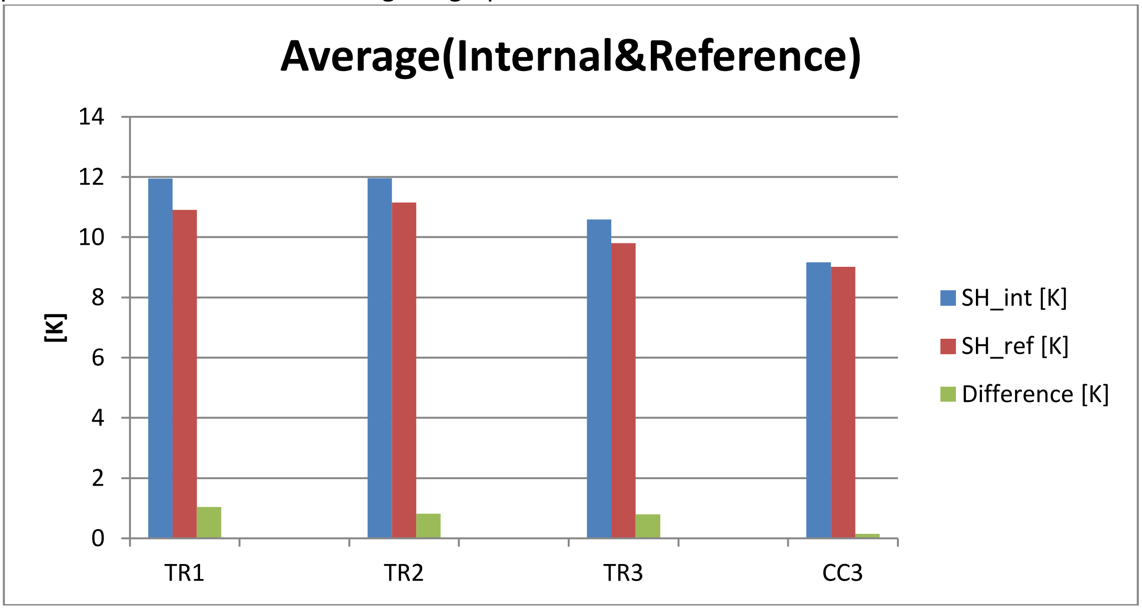


Figure 69: Average internal and reference superheat.

9.5. Conclusions:

One possible conclusion of this study is that taking the reference superheat in order to estimate the enthalpy differences in each cooling unit produce only slight variations in the results. For one stage CO_2 cycle with evaporation temperature -10°C, condensing temperature 20°C and sub-cool 5K, the difference in the cooling capacity taking the reference 10K as superheat, or 12K the real measured superheat is 1.2%. In the comparison between systems, the average difference is less than 2°C for the systems and cabinets. With this information, depending on the accuracy of the analysis that it is required for a supermarket this assumption can be used or not.

For a detailed analysis, with measurements in all the cabinets and easy access to this information, the method should be the first detailed. But there are systems in which for the analysis is taken a limited number of low temperature cabinets and medium temperature cabinets, 5 of 40, averaging the outlet temperature and taking this value for the analysis. In this case taking a sample is introduced uncertainty which could be similar to taking the reference superheat.

For a first general analysis of a system, or in systems where there aren't these temperature sensors in the cabinets, can be used for estimate the cooling capacity.

10. The influences of external superheat on the mass flow estimation methods

In previous reports different supermarkets have been analyzed. Due to in these installation the refrigerant mass flow isn't measured, always comes from manufacturers' information. The parameter used to estimate the mass flow in previous works has been the volumetric efficiency although it has been proposed two different methods.

The study made by Vincent Cottineau "Optimization and calculation of supermarket refrigeration systems (23) shows the superheat in different CO2 supermarkets for the medium and low temperature unit.

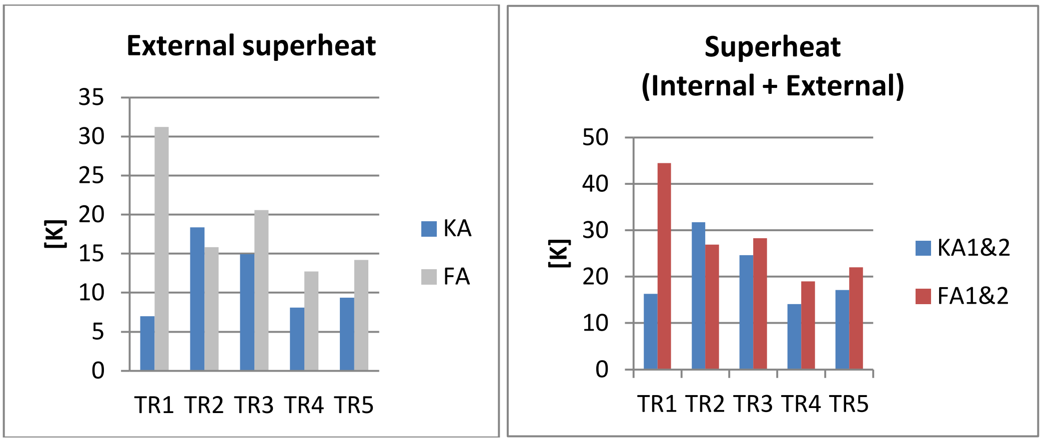


Figure 70: External and total superheat in CO2 systems (24)

There are refrigeration systems with units that have 20K, 30K or 40K as total superheat. In the information provided by the manufacturers the test conditions is 10K superheated gas in the inlet of the compressor. For these reason, we need to evaluate this effect about the compressor electric power.

10.1. References about the superheat effect on the mass flow

From compressors manufacturer:

Bitzer software: has been analyzed the software provided by Bitzer for the compressors selection. One of the input parameters in this software is the internal and external superheat and it is allowed to modify. The assumptions made by the software are:

1. The electric power consumption remains constant when the superheat is modified. It is easily observed in the software.

2. The volumetric efficiency of the compressor remains constant when the superheat is modified. This can be deducted from the results presented by the software. When the superheat is increased the mass flow is reduced according to the following formula:

$$\frac{\dot{m}_{SH=30K}}{\dot{m}_{SH=10K}} = \frac{\rho_{SH=30K}}{\rho_{SH=10K}} \leftarrow \frac{\eta_v \cdot \dot{V}_c \cdot \rho_{SH=30K}}{\eta_v \cdot \dot{V}_c \cdot \rho_{SH=10K}}$$ **Equation 30**

They assume constant volumetric efficiency.

From EN 13771-1 and EN12900:

On the other, for the detailed analysis the European Standard has been studied. You must know all the information related to the manufacturer's test made in their compressors.

"The correction factors applicable to the performance data relating to superheat shall comprise:

a) Change in refrigerating capacity (or mass flow) as a function of the superheat;
b) Change in power absorbed as a function of the superheat.

Correction factors for different values of superheat shall be based on experimental data" (24)

"Power consumption: (…) It is assumed the power consumption remains constant within a range of superheat +/-5K." (25)

"In the limits apply in this standard is assume constant the volumetric efficiency" (25)

Article: "A compressor simulation model with corrections for the level of suction gas superheat"

In the article "A compressor simulation model with corrections for the level of suction gas superheat" by Dabiri and Rice (26) is proposed the Dabiri's method. The ratio between design (map) conditions and actual (new) conditions:

$$\frac{\dot{m}_{new}}{\dot{m}_{map}} = 1 + F \cdot \left(\frac{\rho_{new}}{\rho_{map}} - 1 \right)$$ **Equation 31**

Where F is a chosen percentage of the theoretical mass flow rate increase and where the densities are evaluated based on suction port conditions. F = 0.75 is usually used. This method is difficult to apply because of the proposed correction factor is the result of experience with R22, the experience is from 1982, it was obtained for heat pumps.

The result showed by the above formula is that the mass flow reduction is lower than the density reduction. The conclusion is a slight increase in the volumetric efficiency.

Article: “Influence of the superheat associated to a semi hermetic compressor of a trans-critical CO2 refrigeration plant” (27)

In the article “Influence of the superheat associated to a semi hermetic compressor of a trans-critical CO2 refrigeration plant. D. Sánchez , E. Torrella , R. Cabello, R. Llopis” (27)

The idea is to implement the equations proposed in this article in order to generate all the information in this test, after analyze parametric the influence in the volumetric efficiency without leaving the limits.

In the following Figure is showed the process followed by the refrigerant in hermetic and semi-hermetic compressors:

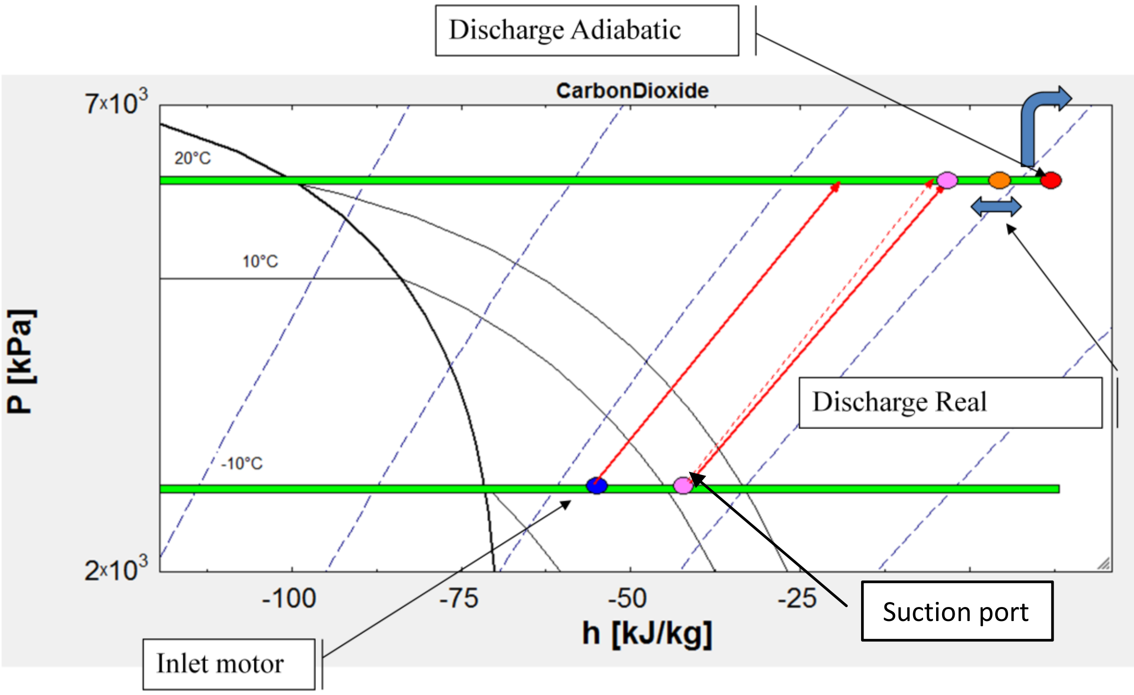


Figure 71: Compression process in a semi-hermetic compressor, p-h diagram

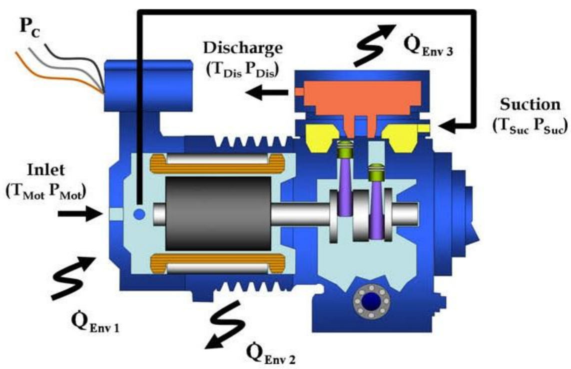


Figure 72: Compression process in a semi-hermetic compressor

The equations proposed in this article are:

$$\dot{m} = \frac{\eta_v \cdot V_G}{\upsilon_{suc}}$$ **Equation 32**

$$P_C = \frac{\dot{m} \cdot \left(h_{Dis,s} - h_{Suc}\right)}{\eta_C} + \dot{m} \cdot \left(h_{Suc} - h_{Mot}\right)$$ **Equation 33**

$$\eta_V = a_0 + a_1 \cdot P_{Mot} + a_2 \cdot P_{Dis} + a_3 \cdot \upsilon_{Suc} + a_4 \cdot N$$
$$SH_{SC} = a_0 + a_1 \cdot P_{Mot} + a_2 \cdot P_{Mot}^2 + a_3 \cdot P_{Dis} + a_4 \cdot P_{Dis}^2 + a_5 \cdot T_{Mot} + a_6 \cdot N$$
$$\eta_C = a_0 + a_1 \cdot P_{Dis} + a_2 \cdot SH_{SC} + a_3 \cdot T_{Mot} + a_4 \cdot N$$
$$T_{Dis} = a_0 + a_1 \cdot P_{Mot} + a_2 \cdot P_{Dis} + a_3 \cdot SH_{SC} + a_4 \cdot T_{Mot} + a_5 \cdot N$$

Depends on the External Superheat

Figure 73: Expressions (5)-(8) in(27) for the volumetric efficiency, super heat in the motor portion, total efficiency and discharge temperature

Coefficient	η_V	η_C	SH_{SC} (°C)	T_{Dis} (°C)
a_0	1.149768	0.781749	11.559877	68.391494
a_1	0.001028	−0.000956	0.003480	−2.453539
a_2	−0.003592	−0.003812	−0.010213	1.009196
a_3	−13.660815	0.003565	0.183598	1.311928
a_4	0.000059	0.000033	−0.000168	0.646575
a_5	–	–	−0.259202	0.002591
a_6	–	–	−0.004376	–
ε_{MAX}	2.1%	3.5%	±1.7	±2.0

Table 11: (Table 3 in(27)): Coefficients and maximum estimation errors for the expressions of the Figure 73.

Parameter	Validity range		Variable	Validity range	
η_V (−)	0.56	0.83	P_{Mot} (bar)	20.35	34.38
η_C (−)	0.61	0.77	P_{Dis} (bar)	74.15	104.90
SH_{SC} (°C)	4.70	21.67	T_{Mot} (°C)	−7.48	9.02
T_{Dis} (°C)	77.00	149.24	N (rpm)	1148.83	1601.90

Table 12: (Table 4 in (27)): Validity range for expressions of the Figure 73

Before this parametric analysis was checked this equations with the information provide for **Zahid Anwar and Yang Chen** about "Experimental measurements of CO_2 heat pump test rig" (22). This information has been used in the comparison of the mass flow methods..

The features of this test were similar to the article only with different compressor size. The Equation 32 and Equation 33 were checked, using the expressions form the Figure 73, with the real measurements from this test. The result is showed in the next plots:

10.2. Validity of the equations from the Article (27) on the measurements in the "Experimental measurements of CO_2 heat pump test rig" (22)

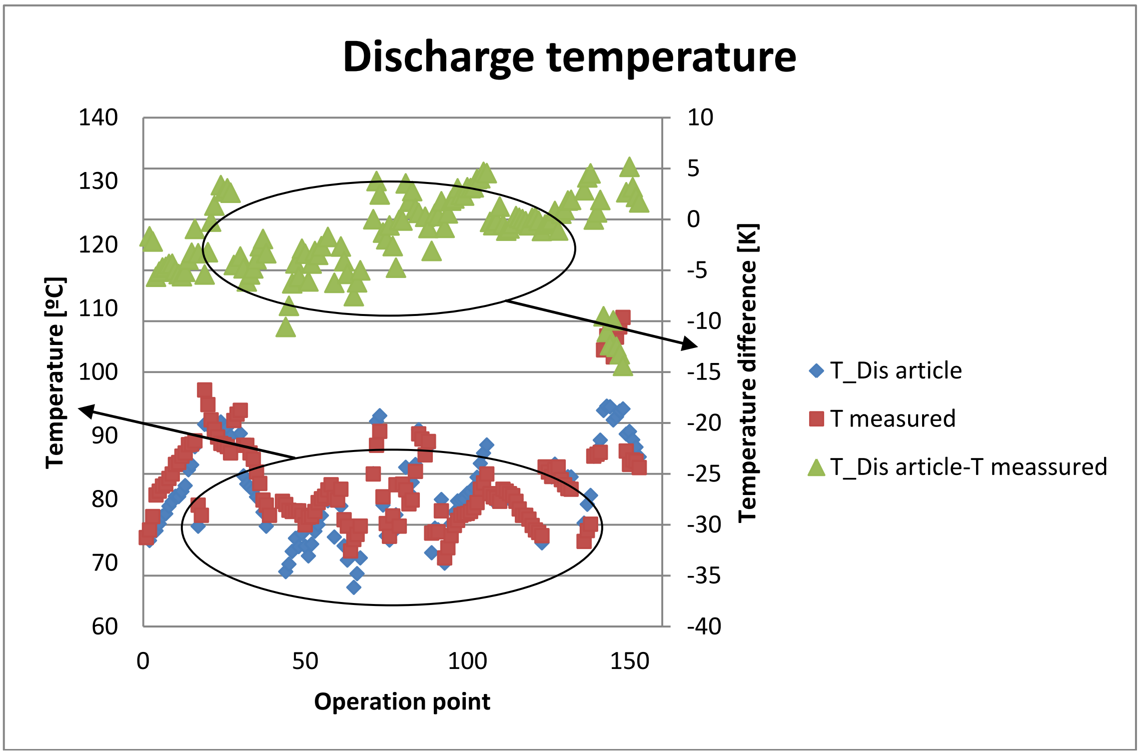


Figure 74: Discharge temperature differences between the measured and calculated

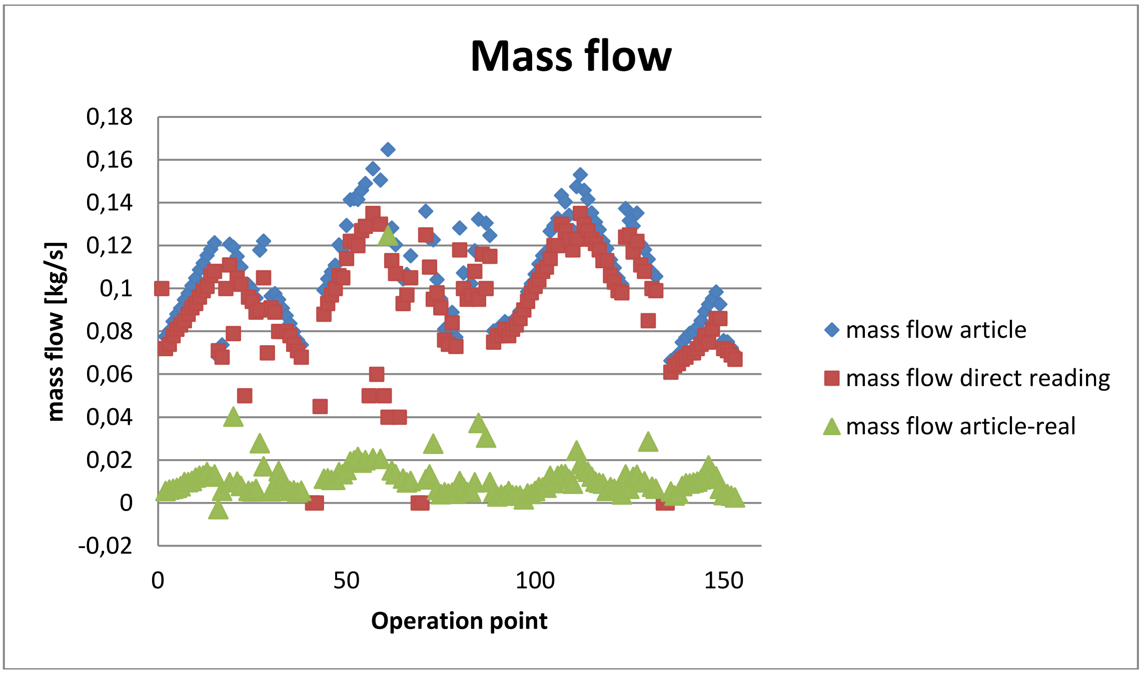


Figure 75: Mass flow differences between the measured and calculated

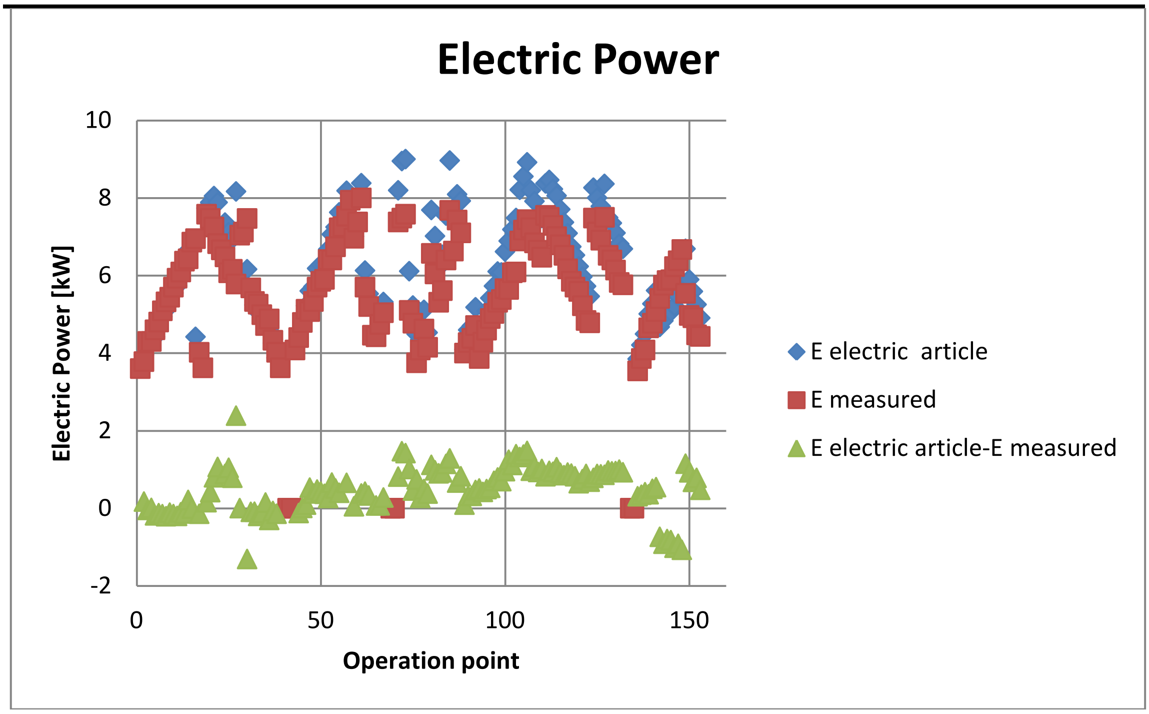


Figure 76: Electric power differences between the measured and calculated

The equation's test fits good results with the experimental data. After checking the validity of the equations for a different test than the article it has been made a parametric analysis about the superheat effect in the volumetric efficiency, without leaving the intervals in which the article indicate that the equations are valid.

10.3. Parametric analysis of the equations

Using the equations from (27) is calculated the mass flow and with this mass flow is possible to define 2 volumetric efficiencies: from the inlet density of the motor *(eta_vol_inlet)* and from suction port density *(eta_vol).*

$$\eta_v = \frac{\dot{m}}{\dot{V}_c \cdot \rho} \begin{cases} \rho_{inlet\_motor} \\ \rho_{suction\_port} \end{cases}$$

Equation 34

The result is showed in the next bar graph:

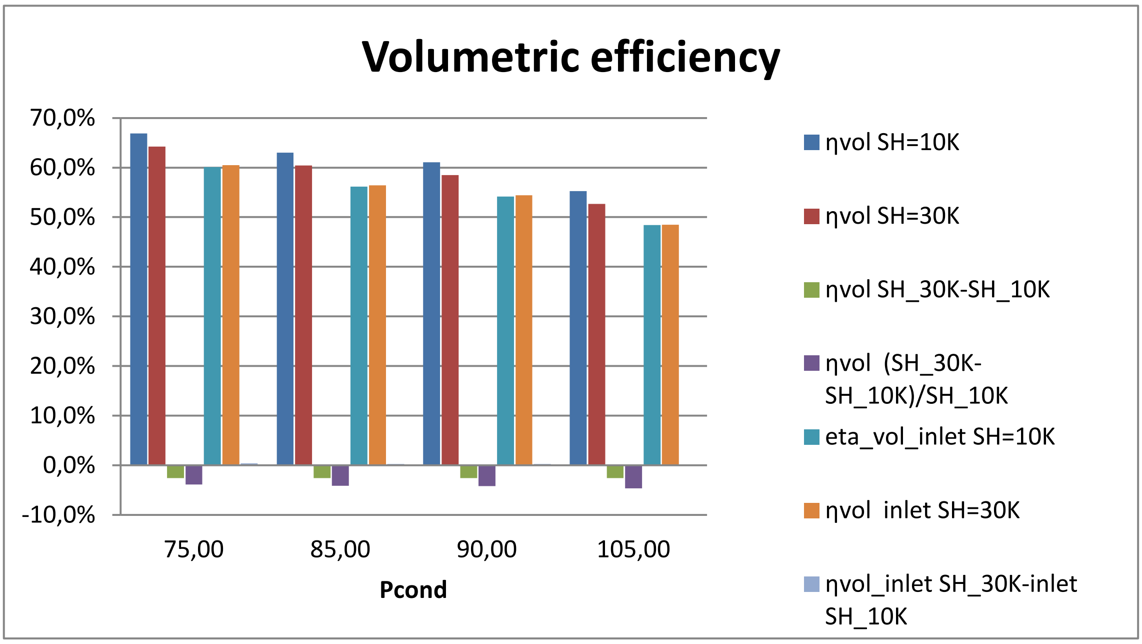


Figure 77: Parametric analysis of the equation in order to know the effect of the superheat on the volumetric efficiency

Although form suction port the relative reduction in the mass flow is about 4%, the volumetric efficiency form inlet conditions, remains constant, and increases slightly.

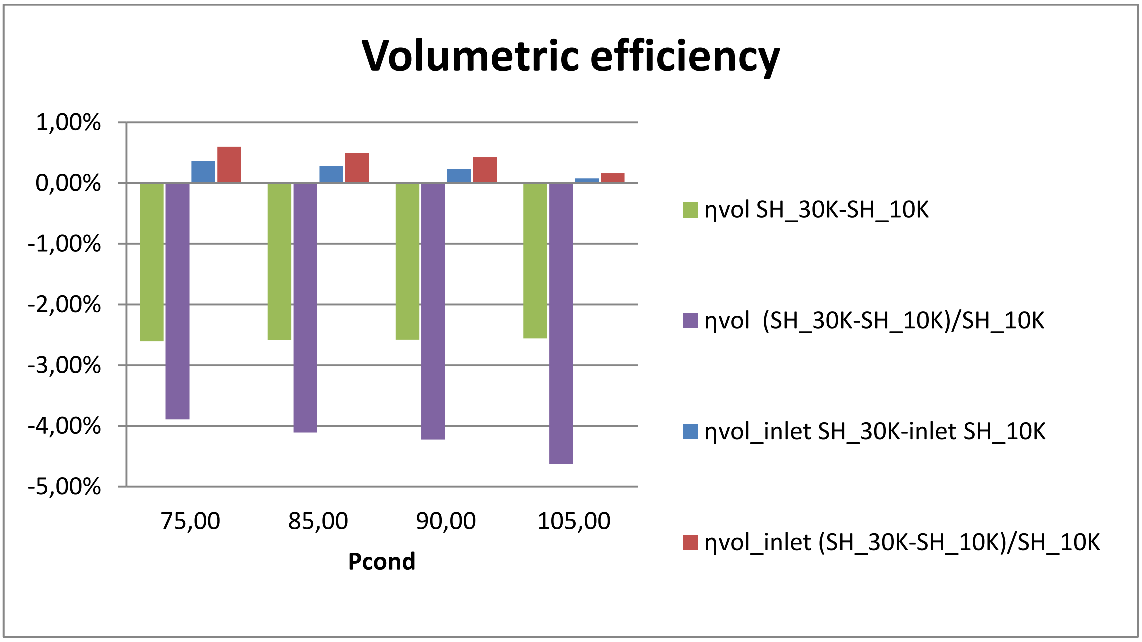


Figure 78: Absolute and relative differences between the volumetric efficiency with 10K and 30K of superheat for different condensing pressure. It is studied the volumetric efficiency from suction port and from inlet conditions.

10.4. Conclusion

The behavior described agree with Dabiri's method using motor inlet conditions. The same parametric analysis was made for the total efficiency calculated with the expression from the Figure 73. The total efficiency presents larger differences than the volumetric efficiency. Therefore the same differences are presented in the electric power consumption by the compressor calculated with the total efficiency due to the

$$P_C = \frac{\dot{m}\cdot\left(h_{Dis,s} - h_{Suc}\right)}{\eta_C} + \dot{m}\cdot\left(h_{Suc} - h_{Mot}\right) \quad \text{Equation 33.}$$

The validity of this analysis is questioned by the variation of the electric power. This variations are high, that not correspond with different articles in which it was studied the influence of IHE in refrigeration systems with CO2. (28). In this article the electric power variation due to the IHE is small. This means that the reduction of refrigerant mass flow rate compensates for the increase of specific compression work.

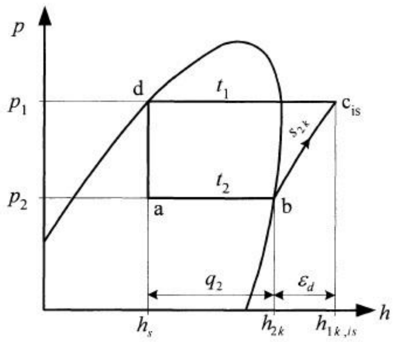
The electric power consumption remains constant in (28). In the Dabiri and Rice`s article (26) the electric power variation is either 0 or increases by 3-5%, lower than the parametric analysis result.

The consideration constant volumetric efficiency with superheat will follow been valid because this analysis does not provided results closer than the real behavior for the electric power consumption.. As a future work, this assumption will be checked in KTH Lab.

11. COP based on coefficients

In this last part, is described a method to estimate the COP. Instead of using the thermodynamic properties are calculated different coefficients. Of course these coefficients are based on the thermodynamic properties, but once are obtained, the correlations allowed to use only pressures and temperatures.

This method is described in the “Refrigerating Engineering”(29) but it has been proposed some modifications. The first correlation compares the basic refrigeration cycle with the coefficient of performance of the Carnot refrigerator.

 $COP_{2d} = \frac{q_2}{\varepsilon_d}$	$COP_{2C} = \frac{T_{evaporator}(K)}{T_{condenser} - T_{evaporator}}$
Basic refrigeration cycle	Coefficient of performance of the Carnot refrigerator

The relation between these two COPs is defined as “Carnot efficiency”.

$$\eta_{Cd} = \frac{COP_{2d}}{COP_{2C}} \qquad \textbf{Equation 35}$$

The first difference introduced is the basic refrigeration cycle used. This basic refrigeration cycle in our case has assumed 10K as internal superheat. Instead of start in saturated conditions, it has been calculated form 10K as superheat for different reasons:

1. It corresponds with the tables provided by compressor manufacturer about the compressor’s operation points.
2. It is more realistic for any cooling system due to the inlet of the compressor is always superheat vapor to avoid liquid in the compressor.
3. The coefficients that evaluate the internal and external superheat the effect in the COP per degree are more constant when we move away from the saturation zone.

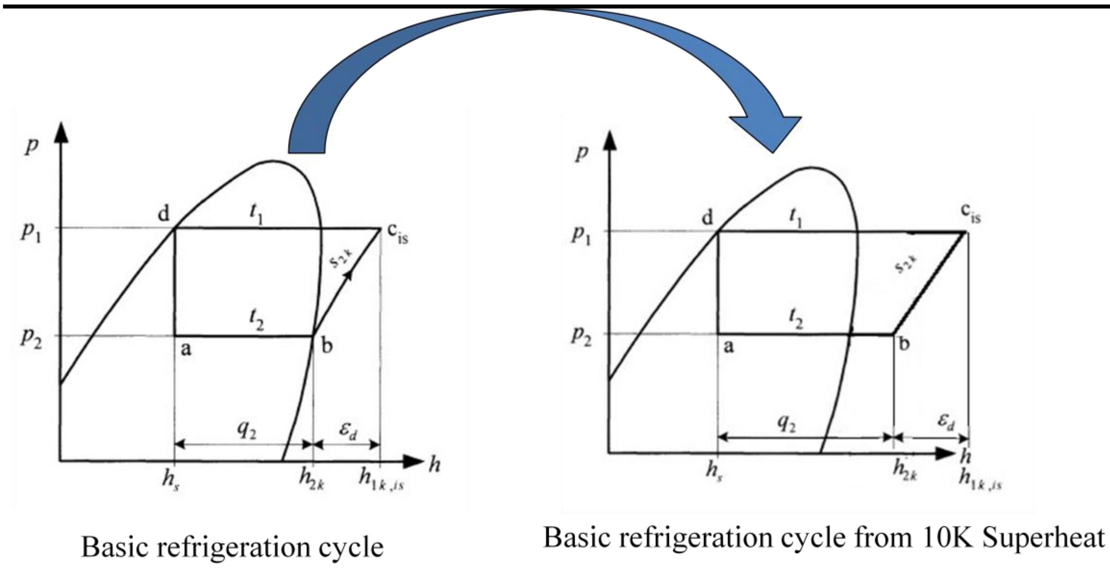


Figure 79: Basic refrigeration cyclte: from 0K and 10K as internal superheat

The Carnot efficiency will be:

$$\eta_{Cd} = \frac{COP_{2d\_SH10K}}{COP_{2C}}$$ **Equation 36**

After the Carnot efficiency, is calculated the coefficient y1 and y4, defined in the same book. These coefficients evaluate the effect in the COP per degree of sub cooling and external superheat. A new coefficient defined y5 is used. It evaluates the effect in the COP per degree when the superheat set point is different than 10K.

These coefficients are calculated from the basic cycle with 10K as superheat. In the "Refrigerating Engineering" (29) is presented this coefficient, but it is not indicate the Delta temperature used for the calculation. The first thing was determinate that the ΔT used was 5K. It was simulated for different superheat in order to see the influence of the ΔT used.

The inputs for the equations are:

- High pressure →T condenser
- Low pressure →T evaporator
- T inlet compressor
- T before expansion valve
- Superheat reference cabinet

For the real COP in our system the next formula is used:

$$COP_{2t} = \eta_{Cd} \cdot COP_{2C} \cdot (1 + y1 \cdot \Delta T_{subcool}) \cdot (1 + y4 \cdot (SH - 10)) \cdot (1 + y5 \cdot (SH_{set\_point} - 10)) \cdot \eta_{tot\_comp}$$

Equation 37

$$\dot{Q} = COP_{2t} \cdot \dot{E}$$ **Equation 38**

11.1. Carnot efficiency ηCd

It has been correlated the Carnot efficiency for each evaporation and condensation temperature for CO2 and R404A, and the Carnot refrigerator COP is calculated using the same temperatures because it is easier than correlate directly the COP_{2d_SH10K}. For the described refrigeration systems it is possible to calculate the COP using the Ideal Carnot Cycle and the different coefficients.

11.1.1. CO2 subcritical regime

It is represented the Carnot efficiency η_{Cd} using CO2 for sub-critical regime. After each figure is presented an equation that correlates all the points showed in the figure.

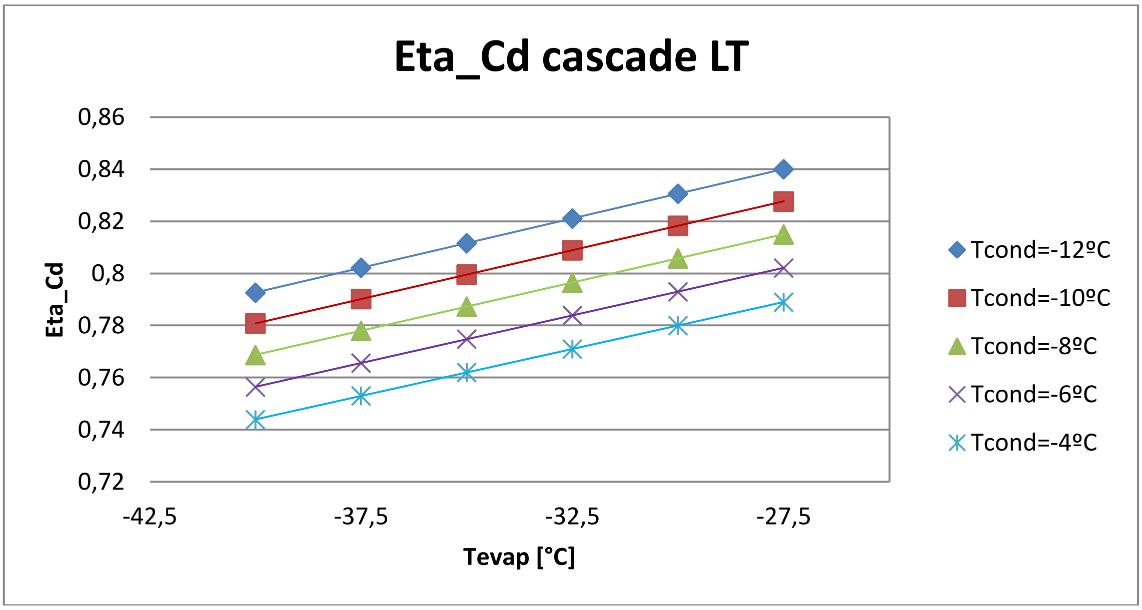


Figure 80: η_{Cd} for the low temperature unit in a cascade system with CO2

Low temperature in cascade system with CO2
$\eta_{Cd} = (0.0037 \cdot T_{evap\_LT} + 0.9169) - 0.00623 \cdot (T_{cond\_LT} - (-8))$

Table 13: Equation that correlates all the points from the Figure 80

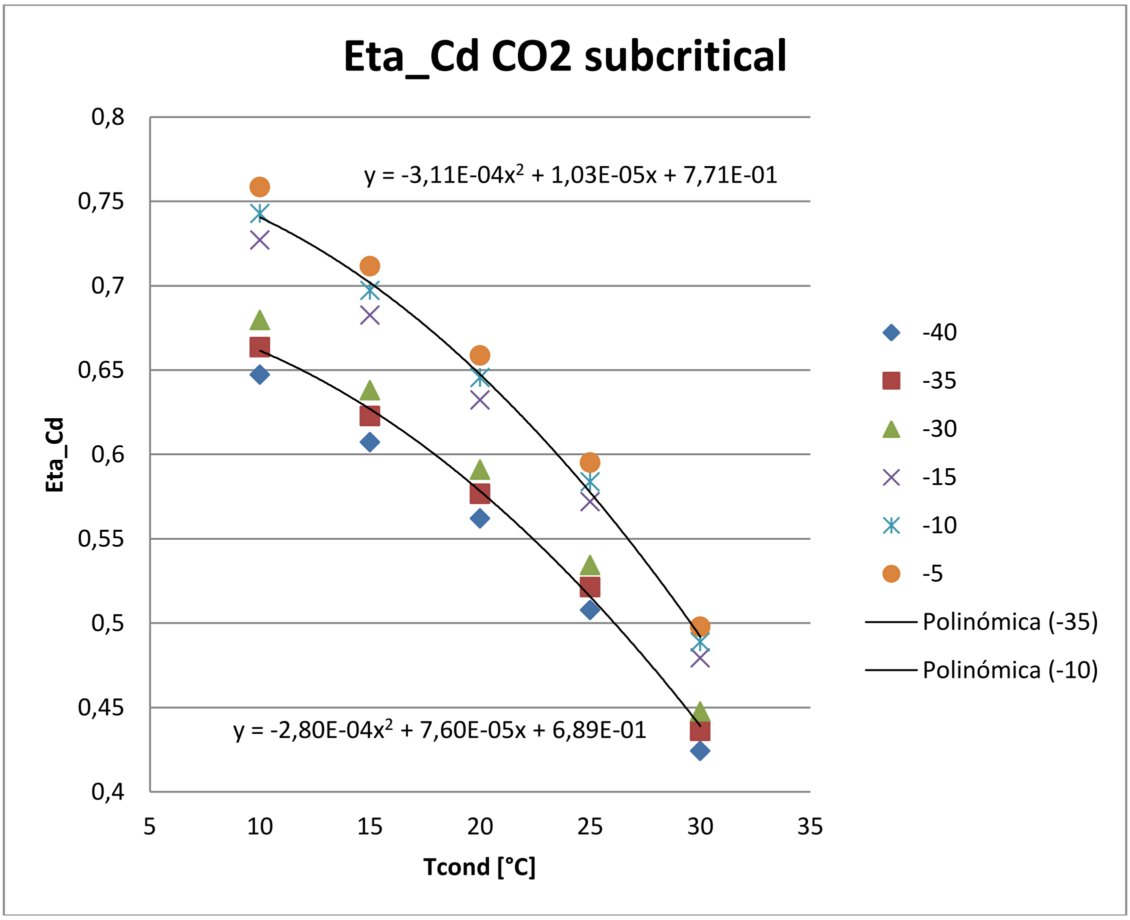


Figure 81: η_{Cd} for the low and medium temperature unit in a parallel arrangement system with CO2, subcritical operation

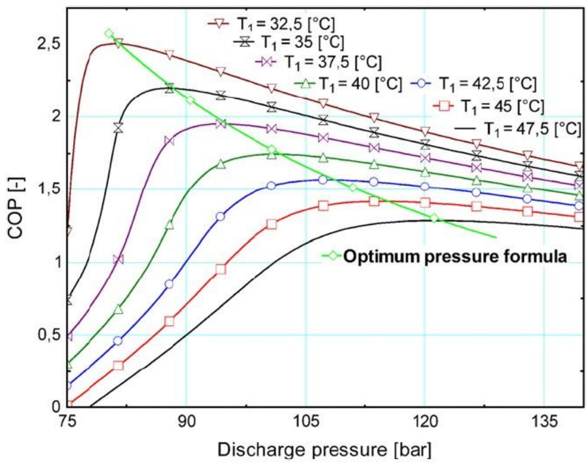
CO2 parallel arrangement system, low and medium temperature, subcritical operation
Low temperature: $\eta_{Cd} = -2.8 \cdot 10^{-4} \cdot T_{cond}{}^2 + 7.6 \cdot 10^{-5} \cdot T_{cond} + 0.689 + 0.00283 \cdot \left(T_{evap\_LT} - (-35)\right)$
Medium temperature: $\eta_{Cd} = -0.00031 \cdot T_{cond}{}^2 + 1 \cdot 10^{-5} \cdot T_{cond} + 0.771 + 0.00257 \cdot \left(T_{evap\_MT} - (-10)\right)$

Table 14: Equations that correlate all the points from the Figure 81

11.1.2. CO2 Trans-critical regime

In “Refrigerating engineering”(29), it is only calculated the coefficients for subcritical regime. To begin the analysis for the trans-critical region must be defined a new condensing temperature. There are two possibilities: to use the optimum pressure or define the virtual saturation temperature.

For each gas cooler outlet temperature has been used the optimal pressure for maximum COP, extracted from “Theoretical evaluation of trans-critical CO2 systems. Samer Sawalha”(7)



$$P_{opt} = 2.7\cdot\left(T_{amb} + T_{gc,app}\right) - 6.1$$
$$T_{amb} + T_{gc,app} = T_{gc\_out}$$

Figure 82: COP of CO2 trans-critical cycle vs. discharge pressure at different gas cooler exit temperatures (denoted T_1) (7)

For each trans-critical pressure is calculated the gas cooler outlet temperature. Using this pressure and temperature is calculated the COP and the Carnot efficiency. The definition of the sub-cooling in trans-critical regimen is the gas cooler optimum minus the real gas cooler outlet temperature.

The other solution consists of using the virtual saturated temperature. It uses the correlation between the saturated pressure and saturated temperature extending the correlation for the trans-critical regime.

The differences between the two curves are showed in the next plot:

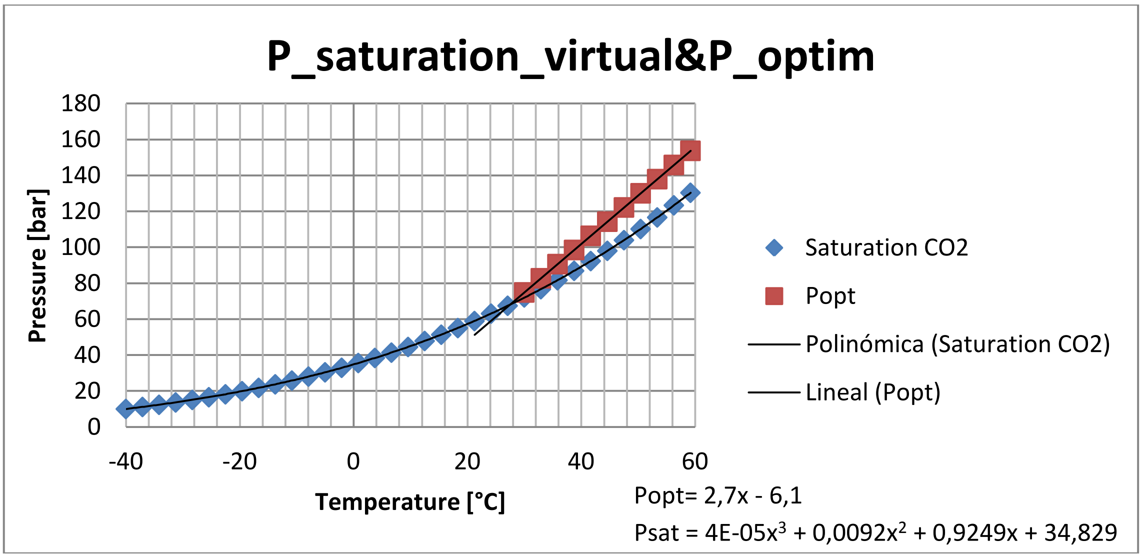


Figure 83: Virtual and optimal saturation pressure vs. Temperature, trans-critical regime

Firstly it was used the virtual saturated curve. In the trans-critical region this curve represents the point where the isotherm lines change the slope, the turning point. But the problem appeared when we started to calculate the coefficient y1 (effect in the COP per K of sub cooling). 1K of sub-cooling from the virtual saturated temperature increase the COP:

T_{gc_out}	35°C	37.5°C
Y1[%/°C]	30%/°C	18%/°C

5K of sub-cooling from the virtual saturated temperature increases the COP:

T_{gc_out}	35°C	37.5°C
Y1[%/°C]	12.5%/°C	11%/°C

The large fluctuation of y1 and the significant influence of the degrees of sub-cooling changed this point of view and it was chosen the optimum pressure for the trans-critical analysis. Using the optimum pressure the y1 coefficient is between 2.4%/°C and 3.8%/°C, it will be showed in the sub-cooling analysis. In our supermarkets we have a condensing pressure, and a gas cooler outlet temperature. It was calculated the gas cooler outlet temperature which causes the actual pressure is the optimal pressure.

It is used the optimum gas cooler exit temperature as condensing temperature for the Carnot efficiency and using this temperature is defined the sub-cooling. The $T_{gc\_out\_opt}$ is calculated from the operating discharge temperature following the

$$P \rightarrow \left\{ \begin{matrix} P = P_{opt} \\ P_{opt} = f\left(T_{gc\_out}\right) = 2.7 \cdot T_{gc\_out\_opt} - 6.1 \end{matrix} \right\} \rightarrow T_{gc\_out\_opt}$$ Equation 39:

$$P \rightarrow \left\{ \begin{matrix} P = P_{opt} \\ P_{opt} = f\left(T_{gc\_out}\right) = 2.7 \cdot T_{gc\_out\_opt} - 6.1 \end{matrix} \right\} \rightarrow T_{gc\_out\_opt}$$ **Equation 39**

The sub-cooling definition is as follow:

$$SC_{subcooling} = T_{gc\_out\_opt} - T_{gc\_out} \qquad \textbf{Equation 40}$$

In the following table is showed the comparison between the two possibilities (optimum pressure and virtual condensation pressure) for the trans-critical regime:

	Optimum pressure	Virtual condensation pressure
Pressure [Bar]	80	80
$T_{gc\_out\_opt}$ or $T_{cond\_virtual}$	$T_{gc\_out\_opt}$ =31.9°C	$T_{cond\_virtual}$ =35°C
COP	2.97	1.95
COP_{1K} $1K_{sub\text{-}cooling}$	3.12	2.45
Increase [%/K] (COP_{1K})	5%/K	25.6%/K
COP_{5K} $5K_{sub\text{-}cooling}$	3.51	3.23
Increase [%/K] (COP_{5K})	3.6 %/K	13.13 %/K

Table 15: Differences between the two possibilities for the trans-critical regime, Optimum and virtual pressure.

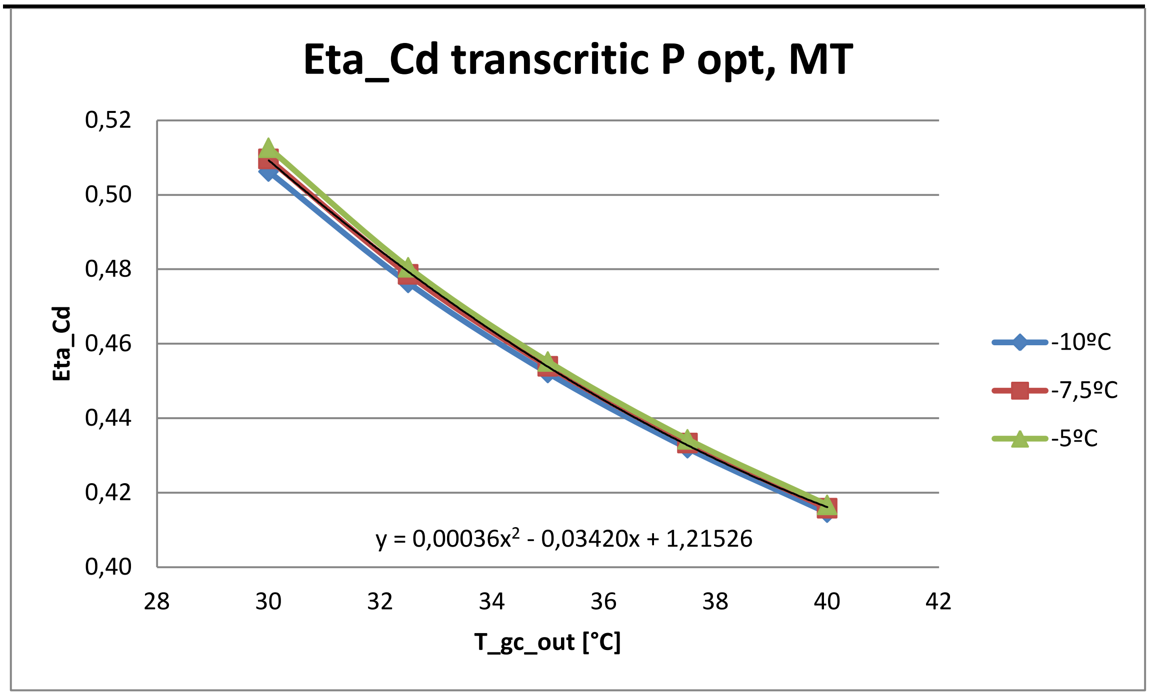


Figure 84: η_{Cd} CO2 medium temperature from optimum pressure, trans-critical regime

CO2 trans-critical with optimum discharge pressure for maximum COP. Medium Temperature
$\eta_{Cd} = 0.00036 \cdot T_{gc\_out\_opt}{}^{2} - 0.0342 \cdot T_{gc\_out\_opt} + 1.21526$

Table 16: Equation that correlates all the points from the Figure 84

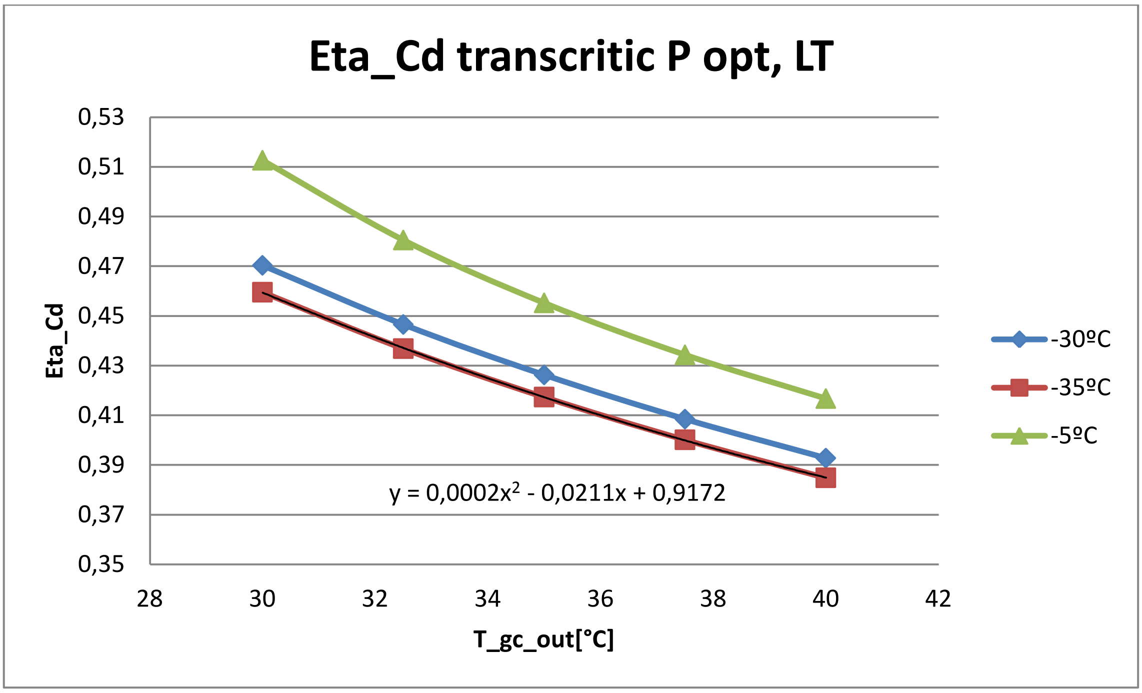


Figure 85: η_{Cd} CO2 low temperature from optimum pressure, trans-critical regime

CO2 trans-critical with optimum discharge pressure for maximum COP. Low temperature
$\eta_{Cd} = 0.0002 \cdot T_{gc\_out\_opt}{}^{2} - 0.0211 \cdot T_{gc\_out\_opt} + 0.9172 + 0.002 \cdot \left(T_{evap\_LT} - (-35)\right)$

Table 17: Equation that correlates all the points from the Figure 85

11.1.3. R404A

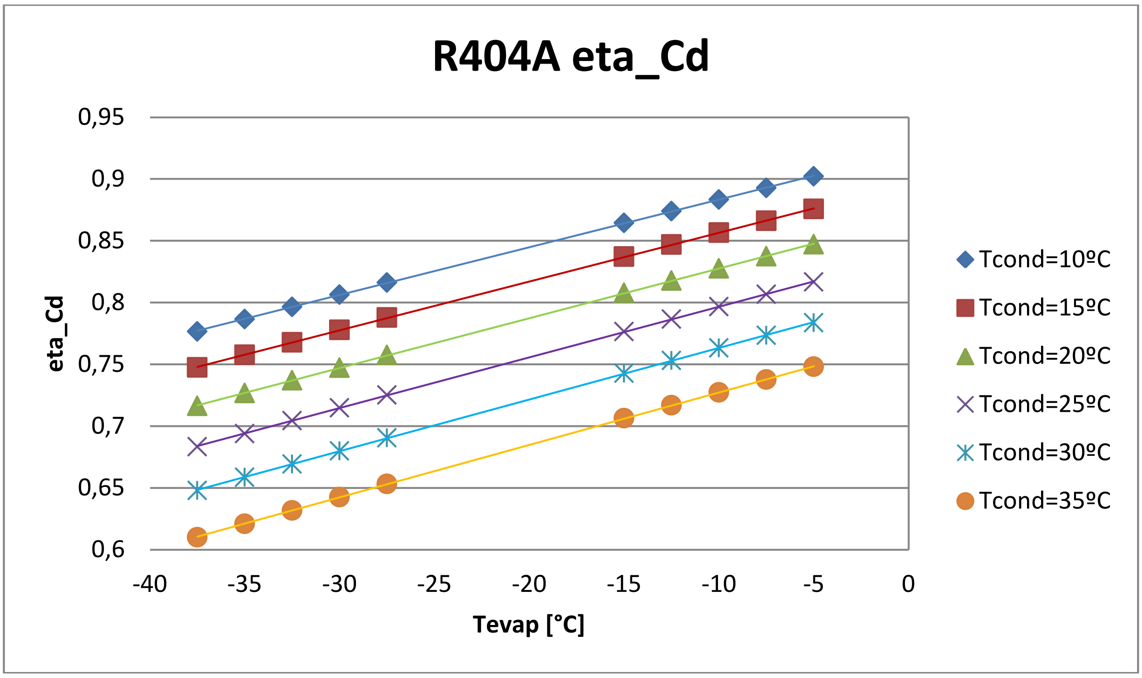


Figure 86: η_{Cd} R404A low and medium temperature

R404A. Carnot efficiency
$\eta_{Cd} = (0.004 \cdot T_{evap} + 0.922) - 0.006418 \cdot (T_{cond} - 10)$

Table 18: Equation that correlates all the points from the Figure 86

11.2. Effect of sub cooling y1

11.2.1. Analysis liquid sub-cooling

The effect of the liquid sub cooling in the condenser outlet, done with a cooling medium (e.g. ambient air or water), produces a beneficial effect increasing the COP. The enthalpy of the liquid decreases, and therefore the quality of the mixture at the inlet evaporator also decreases, increasing the refrigerating effect. The sub cooling can occur at the end of the condenser or preferably in a special heat exchanger, called liquid sub cooler, placed after the condenser. We are going to analyze this effect in CO2 and R404A systems to obtain applicable conclusions to these general guidelines. The idea is to provide a coefficient or mathematical formula that gives us the COP with the condensing temperature and the sub cooling temperature.

The parameter to analyze this effect is defined in the Refrigerating Engineering book as “y1”.

$$y_1 = \left(\frac{h''_{2k} - h_s}{h''_{2k} - h_1'} - 1 \right) \cdot \frac{100}{t_1 - t_s} \left[\frac{\%}{^\circ C} \right]$$

Equation 41 (Equation 3.37 Refrigerating Engineering)

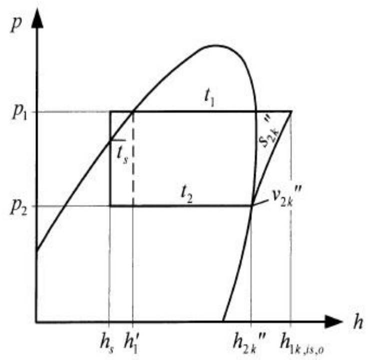


Fig. 3.37 Refrigerating Engineering Cycle with liquid sub cooling

y1 [%/°C] effect per degree of sub cooling in the COP

This factor defines the influence of the liquid sub cooling on COP respect to COP_{2d} (compression cycle with isentropic compression, without sub cool and superheat).

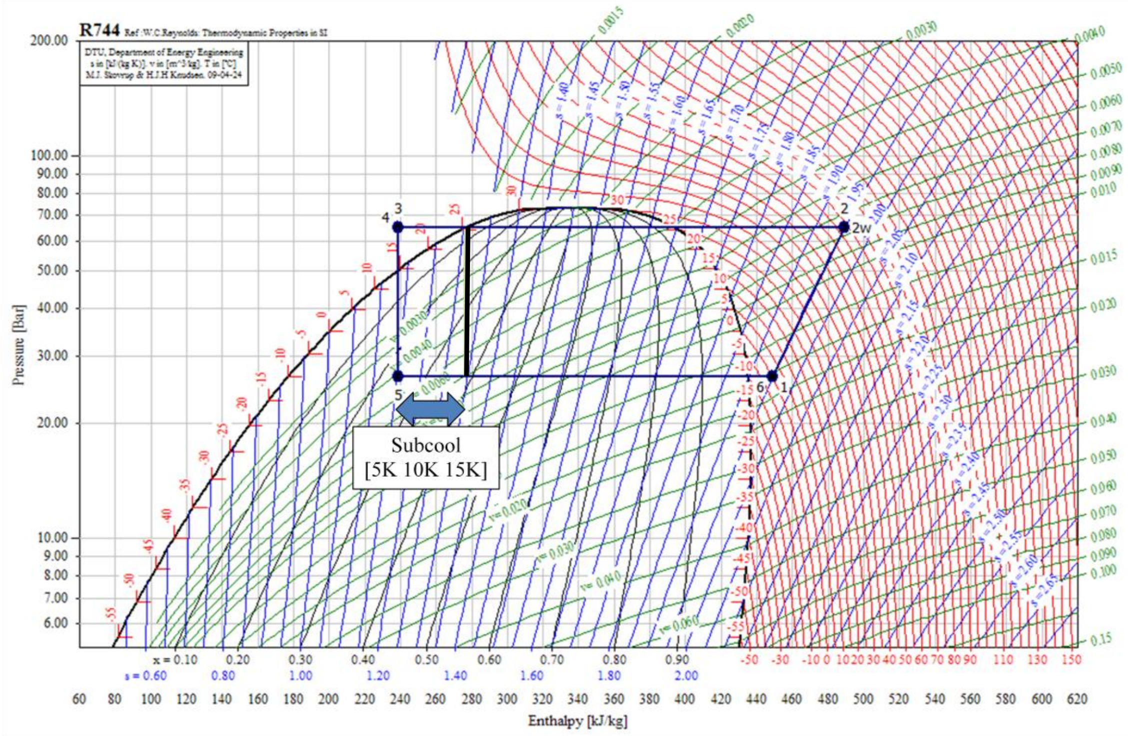


Figure 87: Sub-cooling analysis, scheme.

11.2.2. Analysis liquid sub-cooling in CO2

Due to the following reasons the simulation has used as a reference 10K as internal superheat in the inlet of the compressor.

1. In CO2 systems the internal superheat is about 10k, the expansion valve is set up to provide this superheat in the evaporator outlet.

2. The compressors manufacturer use to provide their information with 10k of superheat in the evaporator outlet.

3. We want the results to be applicable to real systems.

In addition has been analyzed the differences considering 5K, 10K and 15K of sub-cooling, and averaging after using this sub-cool. The results are showed in the next plot:

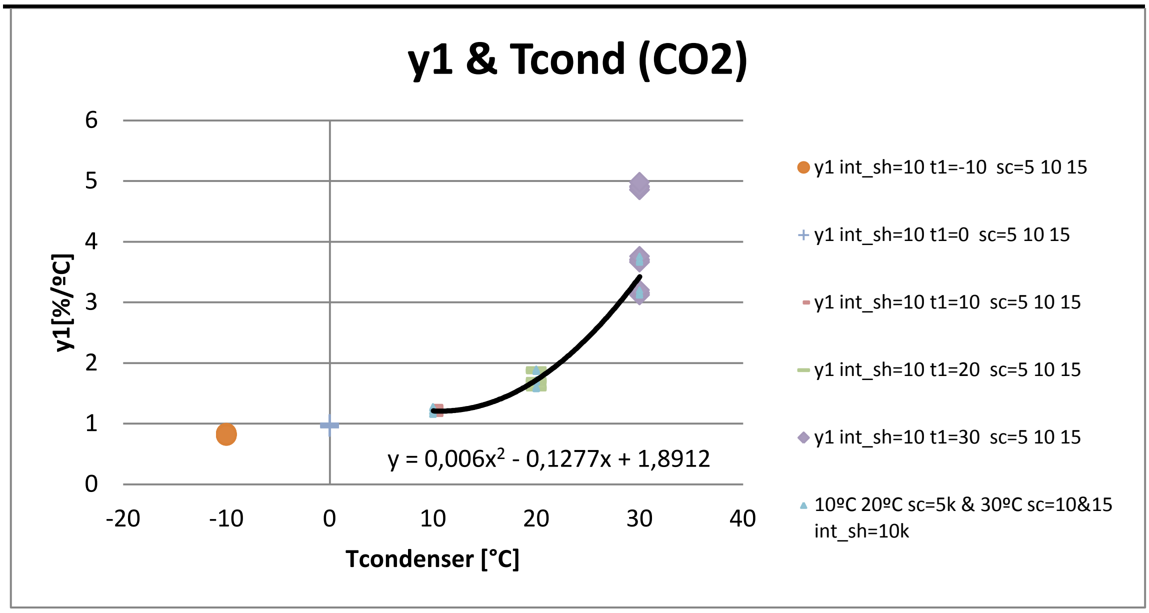


Figure 88: Effect in the COP per degree of subcooling y1, CO2, sub-critical regime

CO2 system y1[%/°C] sub-cooling effect in COP	
Y1: T condensation=[10 to 30]°C	$y1[\%/^{o}C] = 0.006 \cdot T_{cond}^{2} - 0.1277 \cdot T_{cond} + 1.8912$
Y1: T condensation=[-10 to 0]°C (Cascade system with CO2)	$y1[\%/^{o}C] = 0.95$

Table 19: Equations that correlates all the points from the Figure 88

When it is considered higher $\Delta T_{sub\text{-}cooling}$ the effect per degree is lower. When we have an external source to provide sub-cooling in the condenser the $\Delta T_{sub\text{-}cooling}$ may be around 15K. In the next plot is showed the external sub-cooling from ground heat sink. The sub-cool provides fluctuates between 5K and 17K (9).

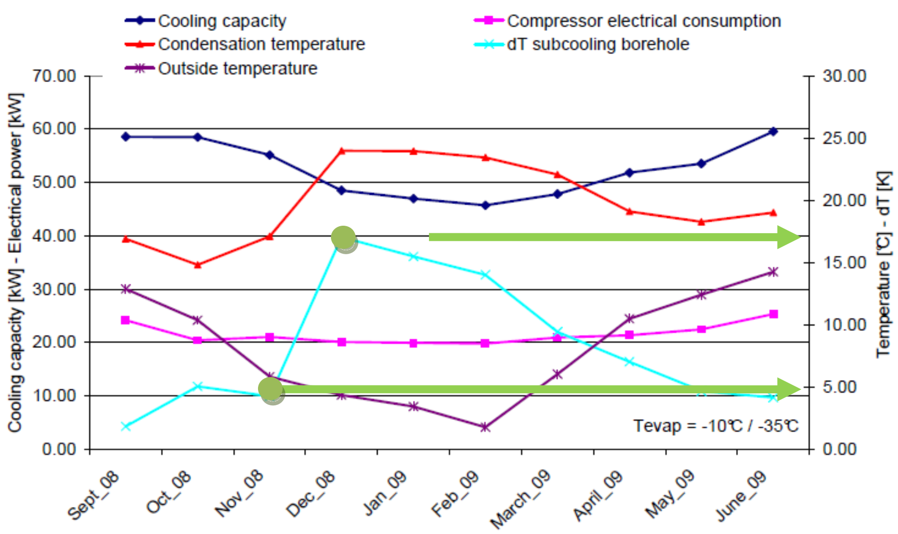


Figure 89: (Figure 6.5 in (9)) Different parameters plots for the KAFA1 unit during the whole period of study in the TR2 supermarket

In this case, we could use the correlation made with 10K of sub-cooling. The underground heat sink is not a conventional solution in supermarket refrigeration systems and for these reason we consider the effect per degree calculated for 5K of sub-cooling although we can see in the plot that the influence per degree when we consider a sub-cooling of 5K, 10K or 15K can be neglected in range between -10°C and 25°C, and we can consider the same effect per degree.

The biggest differences appear when we are closer than the critical point. This situation is given in summer, when the ambient temperature is higher and in winter when the systems is operating with heat recovery. In this two cases is avoid to work with condensing temperature between 26°C and 31°C due to the COP falls rapidly, and to provide the higher heat to the heat recovery system.

A refrigeration system without control to recover heat will operate in floating condensing where the condensing temperature follows the ambient temperature to a minimum condensing level which is usually 10°C. This is a guideline adapted in the refrigeration industry mainly to ensure proper function of the expansion valve and following the recommendations of the compressor manufacturers. In this case, the heating needs in the supermarket are covered by district heating (DH) or a separate heat pump system (SHP). (30) If we have installed a heat pump cascade for the HVAC in the coolant loop, there is a minimum temperature to supply to the heat pump around 10K requiring a condensing temperature around 16°C.

On the other hand if the system has a heat pump connected after the condenser/gas cooler, it extracts the necessary heat from the refrigeration system and provides sub-cooling to 7K. (30)

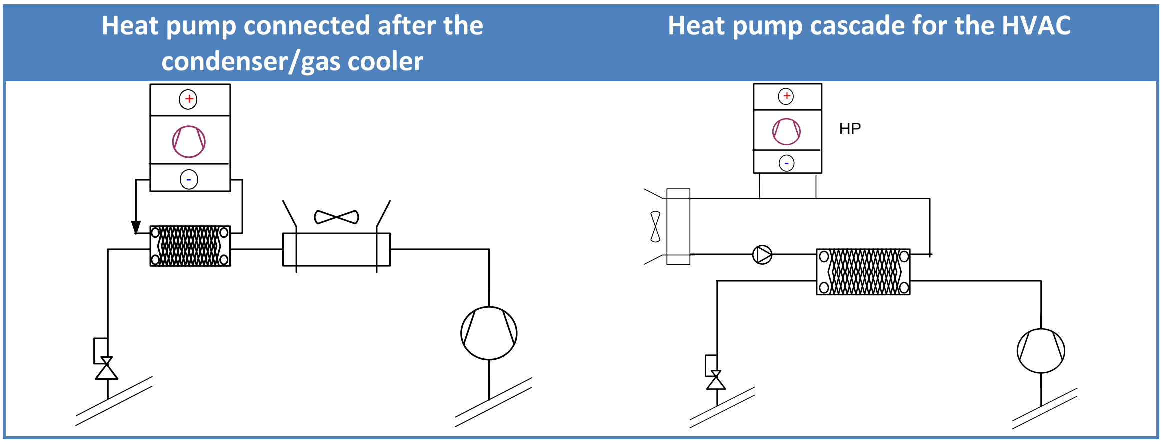


Figure 90: Different heat pumps couplings

All these comments justify the range in which the condensing temperature change in different systems and the correlation used to calculate "y1".

For the trans-critical region there is an optimal pressure $P_{opt} = 2.7 \cdot (T_{amb} + T_{gc,app}) - 6.1$ depending on the ambient temperature and the approach gas cooler temperature for the maximum COP in the refrigeration system. (7)

In the simulation is evaluated the influence in the COP per degree, for different gas cooler outlet temperature. With this point as a reference, and with the optimum pressure, is reduced 5K, 10K and 15K the gas cooler outlet temperature and is calculated "y1". For this zone is easier a correlation using the degrees of sub-cooling y1=f(SC).

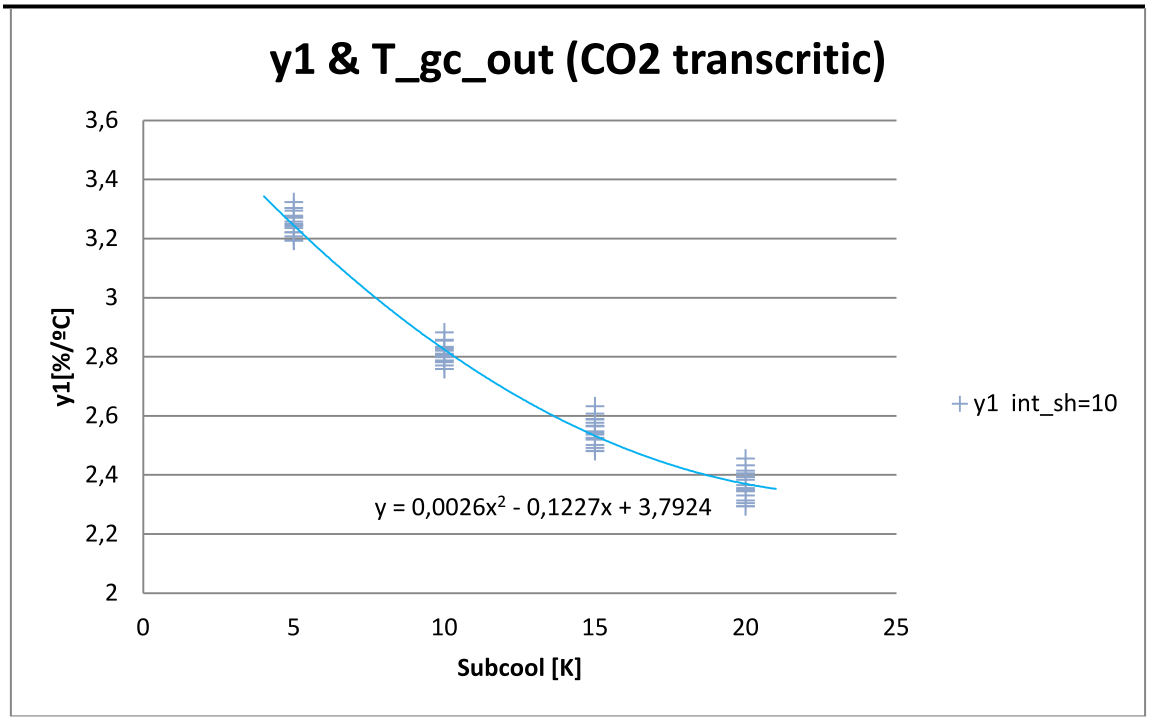


Figure 91: Effect in the COP per degree of subcooling y1, CO2, trans-critical regime

CO2 systems, subcooling effect in COP Trans-critical region	
Y1: gas cooler outlet temperature between [32 and 40]°C	$y1[\%/^{o}C] = 0.0026 \cdot SC^2 - 0.1227 \cdot SC + 3.7924$ $SC_{(subcool)} = T_{gc\_out} - T_{gc\_out\_opt}$

Table 20: Equation that correlates all the points from Figure 91

11.2.3. Analysis Sub cooling in R404A

The same analysis has been done with the refrigerant R404A. For the correlation has been used the T condensation between [10 and 30]°C, the values obtained with 10K as internal superheat, and with 5K and 20K as sub cooling.

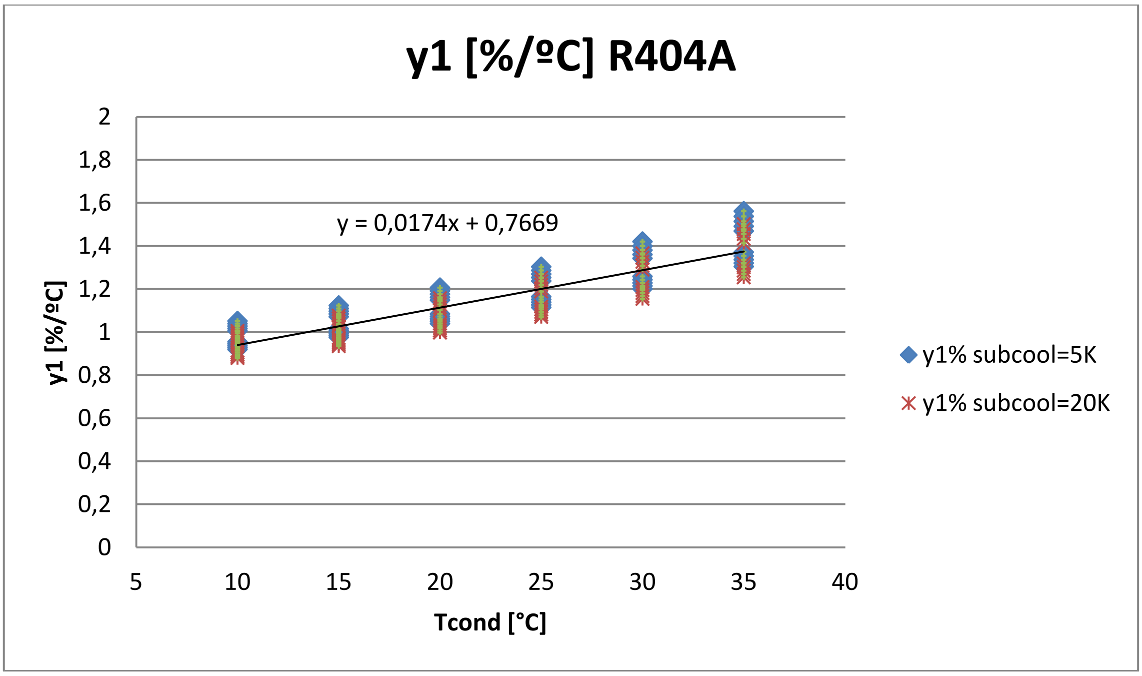


Figure 92: Effect in the COP per degree of sub-cooling y1, R404A

R404A systems, sub cooling effect in COP, T condensation=[5 and 35]°C	
Y1 equation	$y1[\% / ^{o}C] = 0.0174 \cdot Tcond + 0.7669$

Table 21: Equation that correlates all the points from Figure 92

11.3. Effect of external superheat y4

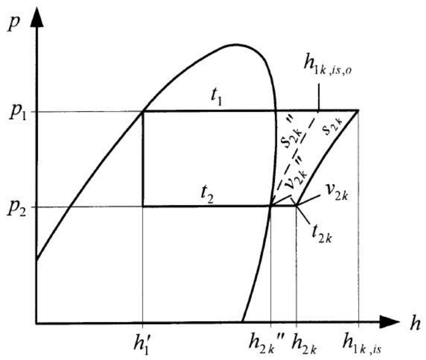
11.3.1. Analysis External Superheat

In the evaporation process the refrigerant is completely vaporized through the evaporator. As the cool refrigerant vapor continues through the evaporator, additional heat is absorbed to superheat the vapor. Under some conditions such pressure losses caused by friction or bad insulation between the evaporator and the compressor increase the amount of superheat. (31) In conventional systems the expansion valve is set up to ensure 7K as internal superheat while the internal superheat in CO2 systems is 10K. In addition we can find external superheat, that doesn't produce a useful effect. In recent measurements has been found superheat up to 20K and for this reason we are going to analyze this effect in CO2 and R404A systems to obtain applicable conclusions to these general guidelines.

The parameter to analyze this effect is defined in the Refrigerating Engineering book as "y4". This factor defines the influence of the external superheat on COP respect to COP_{2d} (compression cycle with isentropic compression, without sub-cool and superheat).

$$y_4 = \left(\frac{h_{1k,is,0} - h''_{2k}}{h_{1k,is} - h_{2k}} - 1 \right) \cdot \frac{100}{t_{2k} - t_2} \left[\frac{\%}{^{\circ}C} \right]$$

Equation 42 (Equation 3.40 Refrigerating Engineering)



y4 [%/°C] effect per degree of superheat in the COP.
Fig. 3.39 Refrigerating Engineering. Cycle with suction vapor superheat

This first analysis has been done in the book and the result has been showed in a table. But in this analyze one important parameter is the temperature t_{2k}. Although the change of COP is normalized by degree this temperature could influence. Above all it has been calculated $t_{2k} - t_2$ used in the table. The result has been 5k. After that has been simulated the same table using [10K 20K 30K] of superheat for subcritical and trans-critical area due to in the book is only analyzed the subcritical region.

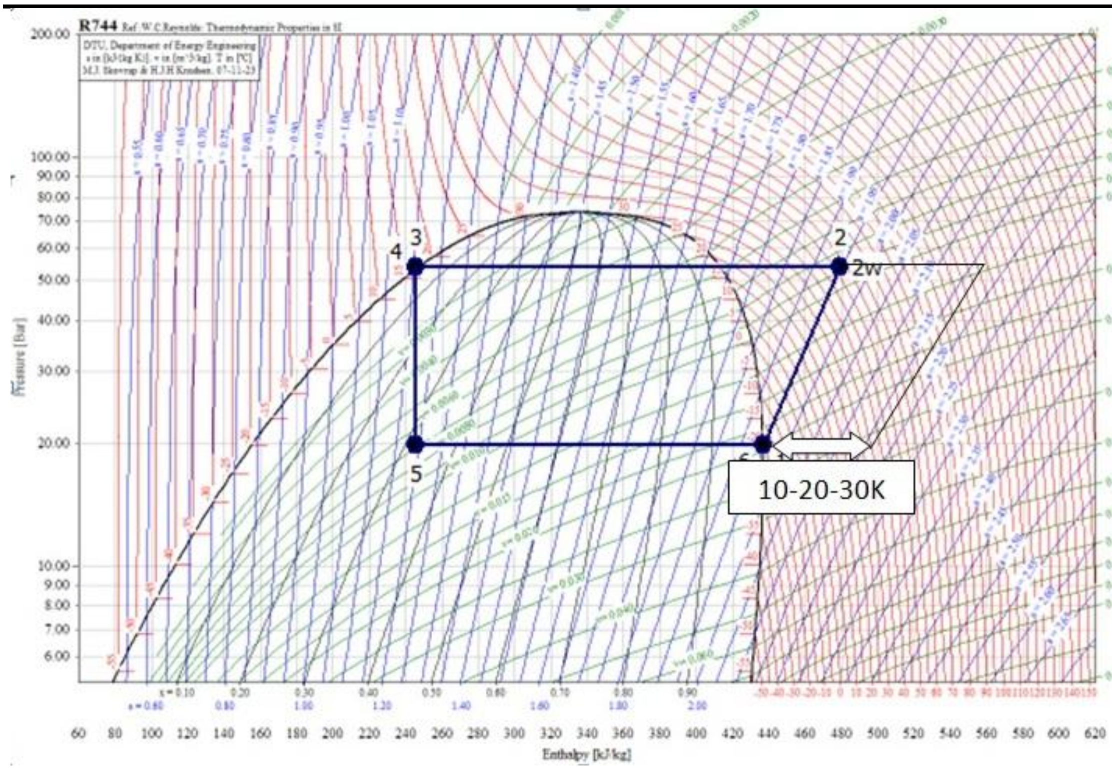


Figure 93: External superheat analysis, scheme

11.3.2. Analysis External Superheat in CO2

In the following plot is showed for each evaporator temperature and each superheat, “y4” for each condenser pressure but the points are overlapping in trans-critical and subcritical mode. Considering isentropic compression the influence of the superheat only depends on the evaporation temperature. It is represented for each evaporation temperature the different PR tested, but it is not appreciate the differences for each ΔT analyzed.

The result for CO2 is showed in the next plot.

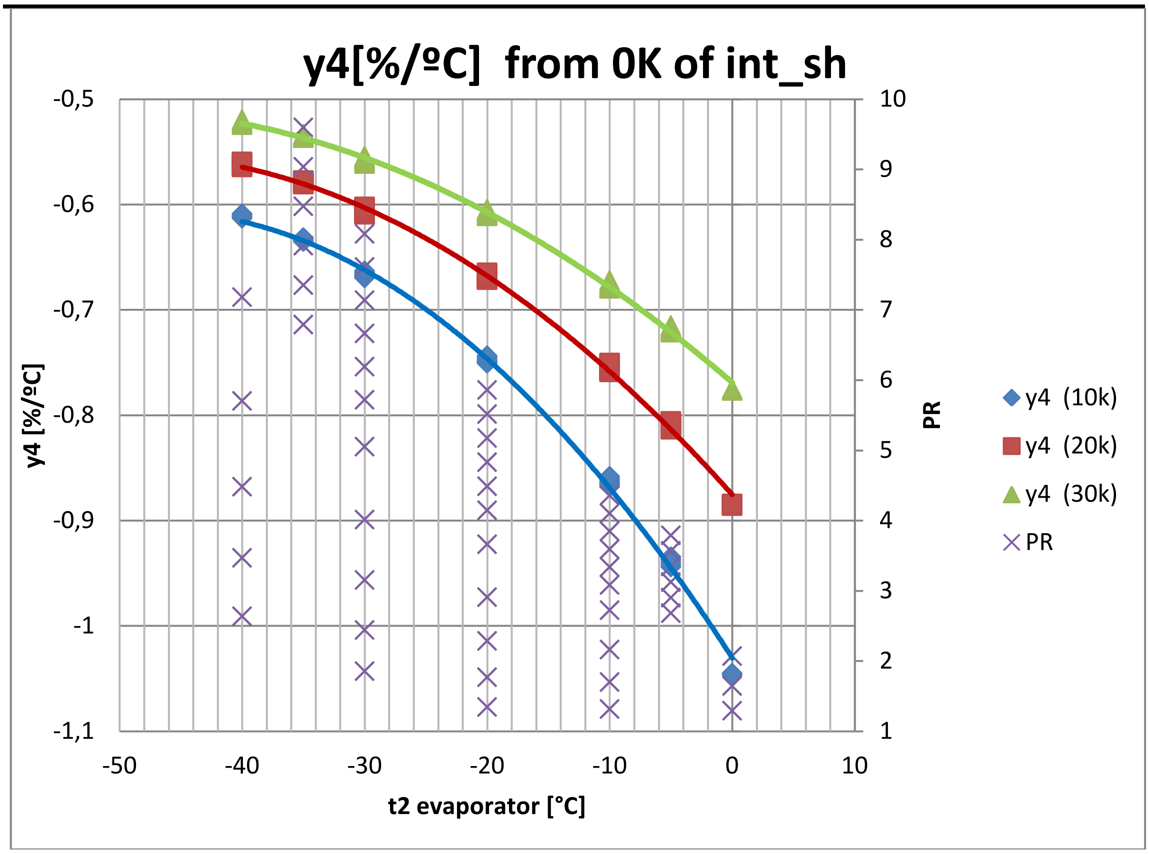


Figure 94: y4[%/°C] coefficient, increase in the COP per degree of external superheat from saturated conditions, CO2

Other important aspect is the ΔT superheat considered in each case. When it is increased the superheat, the effect per degree is lower. For the subcritical region the intervals for the evaporator temperature have been chosen as in the book, but for the trans-critical region has been plotted the temperatures -35°C and -5°C because we are more interested in evaporator temperatures around -35°C and -10°C due to they are the usual temperature used in supermarkets.

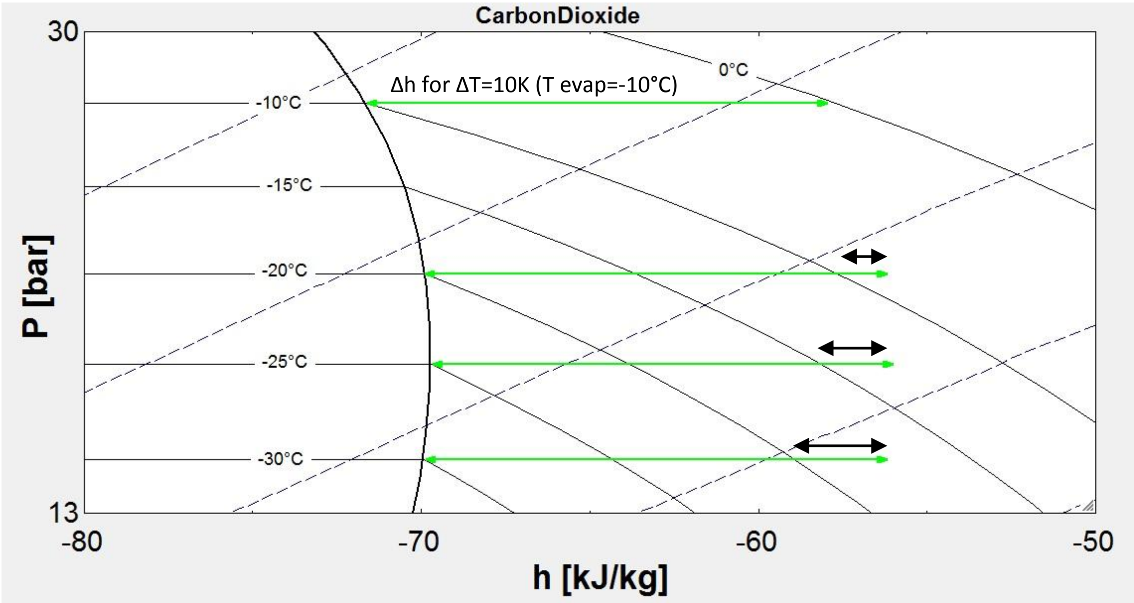
T2(evaporator)	ΔT=5K	ΔT =10K	ΔT=20K	ΔT =30K
-10°C	-0.93[%/°C]	-0.87[%/°C]	-0.76[%/°C]	-0.68[%/°C]
-30°C	-0.71[%/°C]	-0.67[%/°C]	-0.61[%/°C]	-0.56[%/°C]

For example, for -10°C as evaporator temperature, and with 22K of superheat, if we use the value obtained with ΔT=5K, we will have a reduction of 20.5% in COP, when in fact this value should be 16.7%. This behavior is explained as follows:

If we are considering isentropic compression the influence in the COP is due to the change of slope of the isentropic lines. This slope is lower when we move this point, at the same pressure, to the right.

The first isotherms curves, following the isobaric evaporation line in the superheat region, change more their slope, and this effect is higher the higher evaporator temperature. Therefore, when it is increased the ΔT of superheat the effect of the first isothermal curves is attenuated by the following isotherm curves, that are more parallel, and the result when we average is less effect per degree.

When it is increased the evaporator temperature we are closer to the critical point and the slope of the isotherm lines closer the saturation zone changes more. For the same ΔT of superheat, with lower evaporator temperature, the enthalpy difference Δh is lower, the differences between the isentropic lines that we compare is also lower. The condenser temperature doesn't affect considering isentropic compression.



In the before plot is showed the Δh for ΔT=10K, and the same Δh has been translated to lower evaporation temperatures. For lower temperature the Δh that corresponds to ΔT=10K is lower, and due to the differences between the cycle form saturated conditions and with external superheat comes from the change in the isentropic slope, this change is lower and the effect per degree is also lower.

The before analysis has been done using isentropic compression from saturation conditions, but there are an internal superheat that the expansion valve ensures in the evaporator exit. The usual value for CO2 compressors is around 10K. The analysis has been repeated by changing the reference conditions in the book (superheat 0K), using now 10k as a base.

The next plot shows the results:

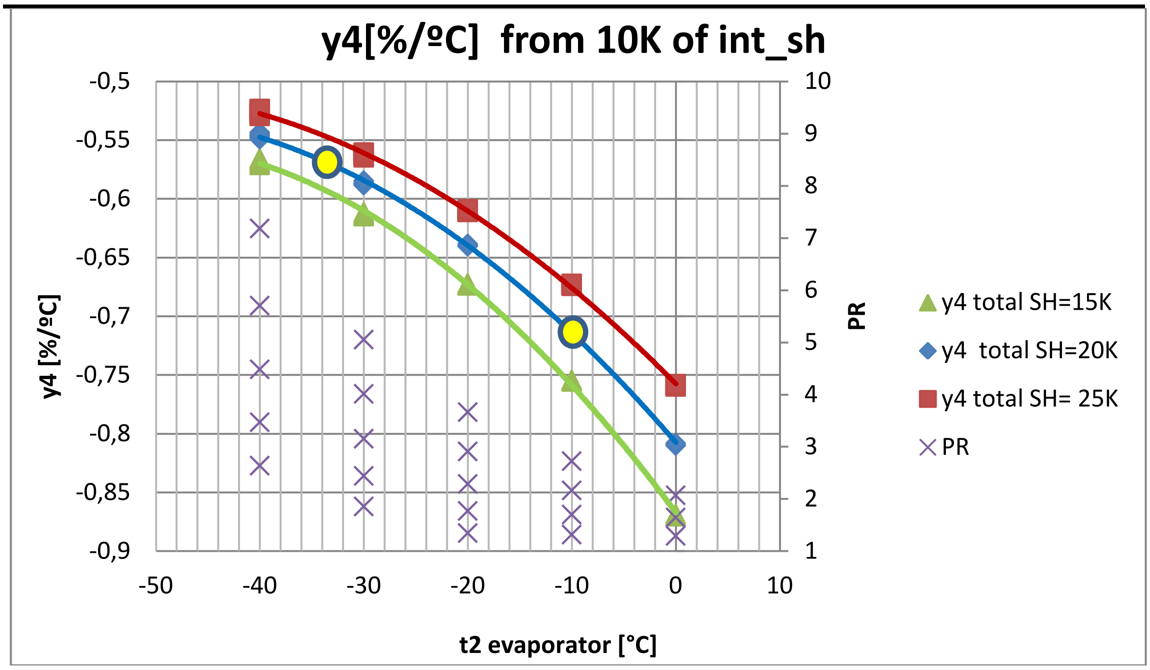


Figure 95: y4[%/°C] increase in the COP per degree of external superheat from 10K as internal superheat, CO2

The next table compares the effect in the COP for each degree of external superheat, form 10k as internal superheat. It is showed the results for the two evaporation temperatures used in CO2 supermarkets systems and considering two values for total superheat.

T2	ΔT=15K total SH	ΔT =25K total SH
-10°C	-0.76[%/°C]	-0.67[%/°C]
-35°C	-0.58[%/°C]	-0.54[%/°C]

Finally the value adopted has been:

CO2 systems, external superheat effect in COP	
Chiller (medium temperature)	$y4[\%/^{o}C]_{Chiller} = -0.72$
Freezer (low temperature)	$y4[\%/^{o}C]_{Freezer} = -0.57$

Table 22: y4[%/°C] coefficients for CO2

11.3.3. Analysis External Superheat in R404A

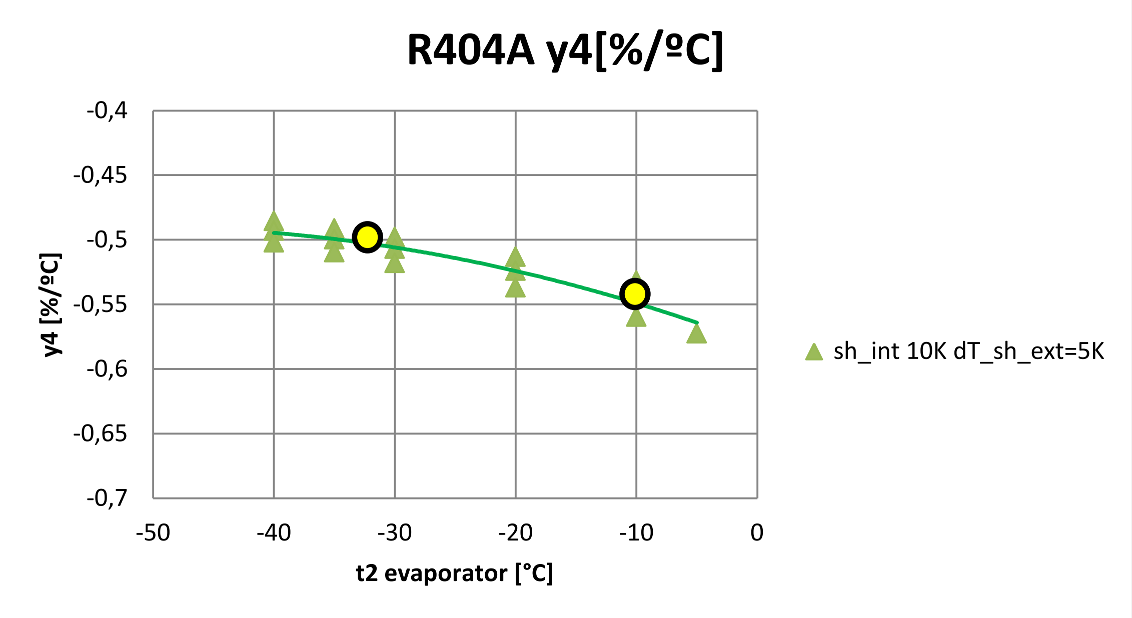


Figure 96: y4[%/°C] increase in the COP per degree of external superheat from 10K as internal superheat, R404A

R404A systems, external superheat effect in COP	
Chiller (medium temperature)	$y4[\%/^{o}C]_{Chiller} = -0.5$
Freezer (low temperature)	$y4[\%/^{o}C]_{Freezer} = -0.55$

Table 23: y4[%/°C] coefficients for R404A

11.4. Effect of internal superheat y5

This coefficient is not defined in the book. It evaluated the effect in the COP per degree of internal superheat different than 10K. In the refrigerating engineering book (29) is defined the coefficient y3. It evaluates at the same time, the increase in the cooling capacity, and the increase in the specific compression work. But in this analysis it has been treated separately. The y5 coefficient evaluates the increase in the COP (due to increase in the cooling capacity) if the internal superheat is different than 10K, and the y4 coefficient evaluates the reduction in the COP (increase in the compression specific power) if the exit of the evaporator presents more than 10K of superheat.

11.4.1. Analysis Internal Superheat in CO2

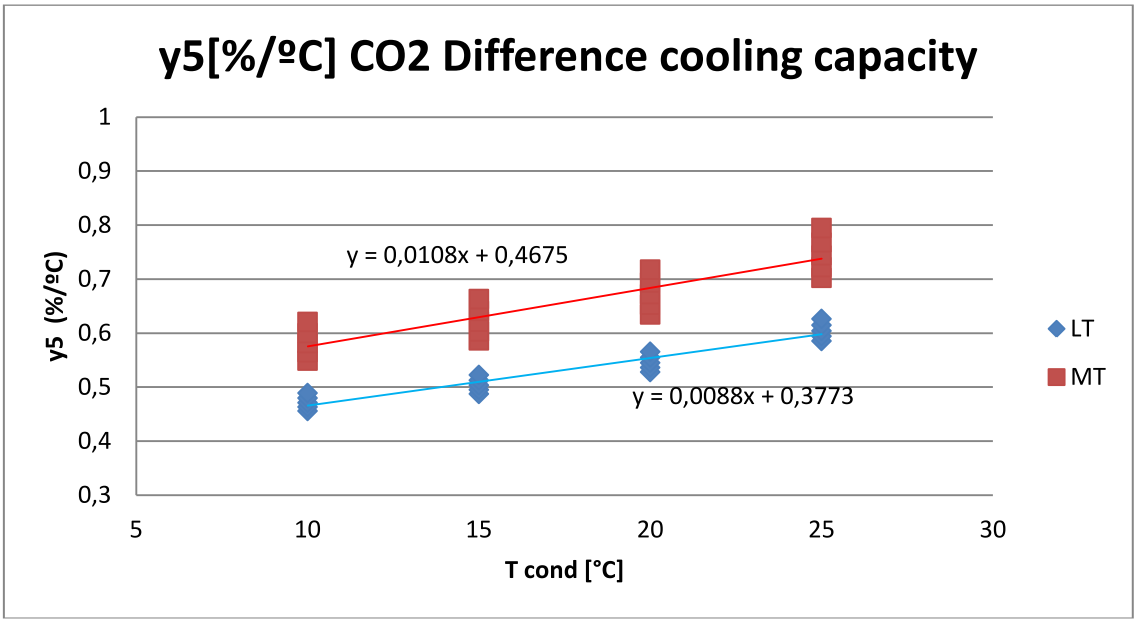


Figure 97: y5[%/°C] increase in the COP per degree of internal superheat higher than 10K. Low and medium temperature (LT&MT), CO2

CO2 systems, internal superheat effect in COP	
Chiller (medium temperature)	$y5[\%/^{o}C]_{Chiller} = 0.0108 \cdot T_{cond} + 0.4675$
Freezer (low temperature)	$y5[\%/^{o}C]_{Freezer} = 0.0088 \cdot T_{cond} + 0.3773$

Table 24: y5[%/°C] coefficients for CO2

11.4.2. Analysis Internal Superheat in R404A

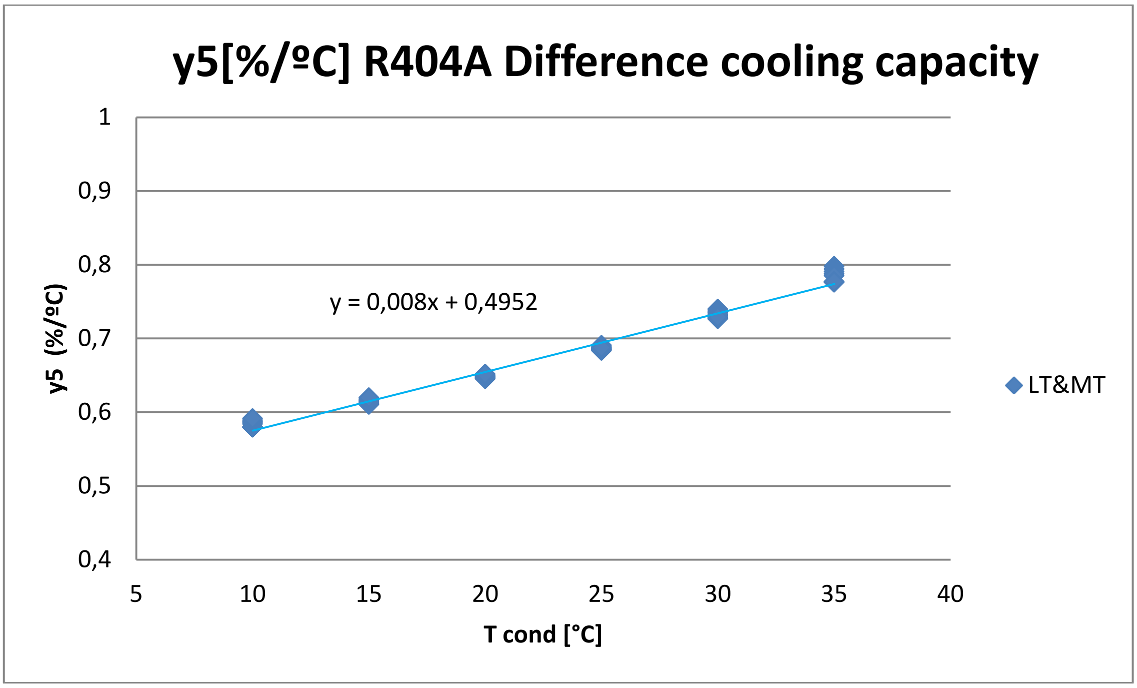


Figure 98: y5[%/°C] increase in the COP per degree of internal superheat higher than 10K. Low and medium temperature (LT&MT), R404A

R404A systems, internal superheat effect in COP	
Chiller and freezer (medium & low temperature)	$y5[\%/^{o}C] = 0.008 \cdot T_{cond} + 0.4952$

Table 25: y5[%/°C] coefficients for R404A

11.5. Sum up, validation and conclusion

A new method for the supermarket analysis has been introduced. The method implicitly uses the total efficiency method although it does not obtain the mass flow directly. It calculates the cooling capacity from the electric power once the total COP has been evaluated. All the equation has been obtained after plot the different cycles for different evaporation and condensing temperatures. After the cycle calculation, the coefficients and equations were obtained.

The advantage for this method is the input parameters. These parameters are pressures and temperatures instead of enthalpies. This has as a result complicated assumptions and difficult equations in order to correlate all the points properly.

The major problem using this method is when it is used for a booster system or cascade system. In these systems is important to know the heat rejected from the low temperature unit to the medium temperature unit, because the electric power consumption from the medium temperature compressors will be distributed between the two units according the cooling capacity in the medium temperature, and the condenser capacity rejected from the low temperature unit. It is necessary a temperature sensor in the low temperature unit compressor outlet. With the instrumentation proposed and the method based on coefficients to determinate the COP, it is only know the electric power, and the isentropic compression, but not the real heat that goes to the refrigerant and will be transferred to the medium temperature unit.

Many assumptions and definitions are required in order to obtain correlations in CO2 trans-critical regime. The possible differences between the method and the calculations using enthalpies come from accumulation of errors in each set of curves. In the following table is showed the differences in the COP using enthalpies or the coefficients for two CO2 basic cycle. It is showed the input temperature and pressure, the calculation of each coefficient and the COP value.

CO2 refrigeration system		
	Medium temperature trans-critical	**Low temperature Sub-critical**
T evaporation	-10°C	-30°C
P condenser	77Bar (Trans-critical)	64.5Bar (T condenser=25°C)
T gas cooler out	27°C	20°C
Internal SH	10K	10K
External superheat	0K	0K
COP	3.61	2.65
	Using the coefficients	
T condenser	$T_{gc\_out\_opt}$=30.8°C	25
Sub-cool	3.8K	5K
COP_{2C}	6.45	4.42
η_{Cd}	0.503	0.53
y1[%/°C]	3.364	2.449
COP	3.66	2.63

Table 26: Differences in the COP using the coefficients method.

The differences are lower than 2%. It is possible to conclude that the alternative method using the coefficients works with reasonable accuracy. It is due all the correlations proposed fits good with all the condensing and evaporator temperatures analyzed.

12. References

1. **J, Arias.** *Energy Usage in Supermarkets – Modelling and Field Measurements.* Energy Technology, KTH Stockholm. 2005.

2. **Berglof, Klas.** http://www.acr-news.com/news/news.asp?id=801&title=Refrigeration+Controls+an+Instrumentation%3B+The+system+shrink. [En línea]

3. **Tamilarasan, Manickam Louis.** *Field Measurements, Evaluation and Comparison of Supermarket Refrigeration Systems.* 2009.

4. **Y.T. Ge, and S.A. Tassoua.** Thermodynamic analysis of transcritical CO2 booster refrigeration systems. *Energy Conversion and Management.* 2011, págs. 1868-1875.

5. **PAVELS.** Field Measurements, Evaluation and Comparison of Supermarket Refrigeration Systems. 2011.

6. **Danfoss.** Transcritical CO2 system in a small supermarket. Dec de 2008.

7. *Theoretical evaluation of trans-critical CO2 systems in supermarket refrigeration. Part I: Modeling, simulation and optimization of two system solutions.* **Sawalha, Samer.** 2007.

8. **Iwmac.** Centralised operation and surveillance, by use of WEB technology. [En línea] 2011. http://www.iwmac.no/english/.

9. **Frelechox, D.** Field measurements and simulation of supermarkets with CO2 refrigeration systems. *Master of Science thesis project in Energy Technology, KTH Stockholm.* 2009.

10. **DANFOSS.** *Automatic Controls for Industrial Refrigeration Systems.* 2006.

11. **Dorin CO2, compressor website.** http://www.dorin.com/documents/Download/19/CO2-0909.pdf. [En línea]

12. **Bock, compressors website.** www.bock.de/. [En línea]

13. **Per Bäckström, Pentronic AB.** Personal contact.

14. **Kimray, Liquid flow meters.** http://www.kimray.com/LinkClick.aspx?fileticket=qOCSQItnW7U%3D&tabid=192&mid=749. [En línea]

15. http://www.cadillacmeter.com/water/cmag.php#pricelist. [En línea]

16. http://www.globalw.com/products/FM500.html#Options. [En línea]

17. **Anders Leidermark, Siemens AB.** Personal contact.

18. **Juan Carlos García, OCHOA maquinaria SL.** Personal contact.

19. *Asercom 2003, Definition of Total Heat Rejection for compressors.*

20. *Climate Change and the Power Industry, Price Water House Coopers, 2005.*

21. **Prakash, John.Arul Mike.** *Energy Optimisation Potential through Improved Onsite Analysing Methods in Refrigeration.* 2006.

22. **Anwar, Zahid.** *Experimental Investigation of heat recovery in CO2 heat pump system.* Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration, Royal Institute of Technology. Stockholm, Sweden : s.n., 2010.

23. **Cottineau, Vincent.** *Optimization and calculation of supermarket refrigeration systems.* 2011.

24. European Standard EN12900. 2005.

25. European Standard EN13771. 2003.

26. **Dabiri A. E., Rice C. K.** *A compressor simulation model with corrections fort he level of suction gas superheat.* s.l. : ASHRAE Transactions, Vol. 87, Part 2, p. 771-782, 1982.

27. *Influence of the superheat associated to a semihermetic compressor of a transcritical CO2 refrigeration plant.* **D. Sánchez, E. Torrella , R. Cabello, R. Llopis.** s.l. : Applied Thermal Engineering 30 302–309, 2010.

28. *Energetic evaluation of an internal heat exchanger in a CO2 transcritical refrigeration plant using experimental data.* **E. Torrella, D. Sanchez, R. Llopis, R. Cabello.** s.l. : International Journal of Refrigeration, 2011, Vols. Volume 34, Issue 1, Pages 40-49.

29. **Granryd, E., et al.** *REFRIGERATING ENGINEERING.* Stockholm : s.n., 2009.
30. **Cheng, Y. y Sawalha, S.** *Investigations of Heat Recovery in Different Refrigeration System Solutions in Supermarkets.* 2010.
31. http://www.refrigers.com/content/view/31654/28/. [En línea]
32. http://www.scribd.com/doc/2634854/MODELOS-MATEMATICOS-DE-SISTEMAS-FISICOS. *Modelos matematicos de sistemas físicos.* [En línea]
33. *Commercial and industrial refrigeration, Application of carbon dioxide as a secondary fluid with phase change, in the low temperature cycle of cascade systems and direct expansion systems with transfer of heat into the environment.* **Girotto, S.** Milano : Politecnico of Milano, 2005. XI European conference on technological innovations in air conditioning and refrigeration.
34. http://www.comeval.es/pdf/cat_tec/intercambiadores/intercambiadores_A4_esp.pdf. [En línea]
35. http://energianow.com/Articulos/sistema_refrigeracion_eficiencia.pdf. [En línea]
36. **Knut Bakken, Askim, Norway.** *Refrigeration system with closed circuit circulation. 6,112,532* 5 de Sep de 2000.
37. **Ibrahim Dincer, Mehmet Kanoglu.** *Refrigeration Systems and Applications .* http://books.google.es/books?id=WnEqJLuI-hMC&pg=PA109&lpg=PA109&dq=pump+in++refrigeration+systems+function+and+description&source=bl&ots=5Yk3WwZmFF&sig=RJiKFMbhFu0_1elTYvM38540TRE&hl=es&ei=Rt5wTZjJN87usgaPscyGDw&sa=X&oi=book_result&ct=result&resnum=7&ved= : s.n.
38. *Optimum flow rates in indirect systems.* **Granryd, E.** Beijing : s.n., 2007. International Congress of Refrigeration.
39. **Department of Energy Technology KTH.** *Refrigerating Engineering.* Stockholm : s.n., 2009.
40. **KP110, Bitzer compressors tandem.** http://pdf.directindustry.com/pdf/bitzer/semi-hermetic-recip-compressors-tandem-kp-110-3/17562-114281-_2.html. [En línea]
41. *Naumburg P-H, Experience of performance tests on medium size hermetic compressors, Scan Ref 4/87.*
42. **RDM.** Centralised operation and surveillance, access through modem line. [En línea] 2009. http://www.resourcedm.com/.
43. **Berglöf, Klas.** *METHODS AND POTENTIAL FOR ON-SITE PERFORMANCE VALIDATION OF AIR CONDITIONING, REFRIGERATION, AND HEAT PUMP SYSTEMS.* 2005.