Field Measurements of Supermarket Refrigeration Systems. Part I: Analysis of CO_2 trans-critical refrigeration systems

Samer Sawalha, Mazyar Karampour, Jörgen Rogstam
Department of Energy Technology, The Royal Institute of Technology (KTH),
Brinellvägen 68, SE-10044 Stockholm, Sweden.
Tel: +46 (0)8 790 7889
Fax: Fax: +46 (0)8 204 161
E-mail: samer.sawalha@energy.kth.se

Keywords: Carbon dioxide (CO_2); Supermarket refrigeration; Trans-critical cycle; Field measurements

Abstract

This study investigates the refrigeration performance of three CO_2 trans-critical solutions based on field measurements. The measurements are carried out in five supermarkets in Sweden. Using the field measurements, low and medium temperature level cooling capacities and COP's are calculated for ten-minute intervals, filtered and averaged to monthly values. The results indicate that the systems using trans-critical booster system with gas removal from the intermediate vessel have relatively the highest total COP. The reasons are higher evaporation temperatures, lower internal and external superheat and higher total efficiency of booster compressors. Another important factor is gas removal from the intermediate vessel which leads to higher COP of low temperature level. Comparing the older and newer installed systems, a trend in energy efficiency improvement has been seen. The study shows this improvement originates from both changes in the system design (e.g. two stage expansion) and components efficiency improvement (e.g. higher total efficiency of compressors - lower internal superheat and higher evaporation temperatures of cabinets).

Keywords: Carbon dioxide (CO_2); Supermarket refrigeration; Trans-critical cycle; Field measurements

Nomenclature

$\dot{Q}_2$	Cooling capacity, kW
$\dot{V}_s$	Swept volume flow rate, m^3 s
$\dot{E}$	Power consumption, kW
$\dot{Q}$	Capacity, kW
$\dot{m}$	Mass flow rate, kg s^{-1}
Δh	Enthalpy difference, kJ kg^{-1}
BO	Booster
COP	Coefficient of performance
DX	Direct expansion
FR	Freezing/Low temperature level
h	Enthalpy, kJ kg^{-1}
HFC	Hydrocarbon
LR	Load Ratio
ML	Medium temperature level
TR	CO_2 trans-critical systems
Cp	Specific heat capacity, kJ kg^{-1} K^{-1}
SH	Super heat, K
n	Number of compressors, -
η	Efficiency, -
ρ	Density, kg m^{-3}

Subscript

amb	ambient
c	coolant

cab	cabinets
comp	compressor
diff	difference
el	electric
ev	evaporator
hs	high stage
ht	heat
in	inlet
is	isentropic
ls	low stage
oil	for oil
out	outlet
r	refrigerant
rat	rated
ratio	for ratio
sur	surroundings
tot	total
v	volumetric

1 Introduction

Natural refrigerants are seen as a potential permanent solution where CO_2 is the one that fits best in supermarket applications, mostly due to safety reasons, as it can be directly used in public areas. The three main solutions where CO_2 is applied in supermarket refrigeration are the indirect, cascade and the trans-critical systems. CO_2 was first used in supermarket refrigeration as a secondary working fluid in indirect system solutions for low temperature applications where pressure levels are reasonably low and conventional components can be used. The knowledge and experience gained from early research work on CO_2 and the early installations of CO_2 in commercial applications promoted its wider application in supermarkets with different system solutions. Cascade systems with CO_2 in the low stage and trans-critical solutions where CO_2 is the only working fluid have been widely applied in recent years, especially in mid-northern Europe where over 2800 installations of CO_2 trans-critical solutions have been reported [1].

The reason for the high number of CO_2 trans-critical system installations in northern Europe compared to warmer climates is that CO_2 trans-critical systems have similar or higher COP compared to conventional hydrofluorocarbon (HFC) system at ambient temperatures lower than about 25°C. The exact value is dependent on systems design and boundary conditions; for example in one study it is shown that CO_2 trans-critical system has higher COP than conventional R404A in ambient temperatures lower than 23°C [2], while in another study comparing trans-critical booster systems with R404A standard and optimized systems, the crossover point ranges between 21-25°C [3]. Therefore, it is essential to keep the condensing temperatures of the CO_2 systems as low as -possible by rejecting heat from the condenser/gas cooler directly to the ambient. This is true unless the system is controlled for heat recovery; in this case the discharge pressure is controlled to match the heating demand.

Different CO_2 system solutions for supermarket refrigeration have been extensively studied by computer simulation modeling and laboratory experimental evaluation and compared to conventional R404A system solution in Swedish supermarkets. The results prove the studied CO_2 systems have similar or higher efficiency compared to conventional solutions [2].

Similar studies have been done comparing other system solutions using CO_2. Cecchinato et al. compared the performance of various integrated and separated supermarket refrigeration and HVAC systems using HFC and CO_2 solutions for different climates [4]. The baseline system includes separate R404A direct expansion refrigeration and R410 air conditioning-heat pump system. Two of five alternative solutions use CO_2 in the refrigeration low temperature level (LT). It is shown that while annual energy consumption in the LT level of these two systems is reduced, the energy consumption is increased in the MT level as a cascade configuration is used. Extending the previous study, the baseline HFC system was compared to only natural-refrigerant solutions by Cecchinato et. Al [5]. The alternative HVAC systems use NH_3 with auxiliary condenser boiler or R290 while refrigeration systems use NH_3-CO_2, R290-CO_2 or only-CO_2 system solutions. It is concluded that up to 15% annual energy consumption can be decreased comparing with the baseline HVAC&R system. Sharma et al. [6] compared seven CO_2 booster or cascade/ secondary systems with a baseline R404A system. Using a bin analysis method for eight US climatic zones, it is found that the efficiency of the best CO_2 system design, a booster system with parallel compression, is equal to or more efficient than the baseline system in northern two-third of the country. Mikhailov and Matthiesen [7] compared the performance of a benchmark R404A system with various alternatives including CO_2 cascade, indirect and trans-critical booster systems. It is shown that for Quebec, representing cold climate, up to 18% annual energy can be saved using a CO_2 trans-critical booster system. For Atlanta, representing warm climate, cascade NH_3-CO_2 with CO_2 pumping in MT level and CO2 DX in LT level decreases the annual energy consumption by 11%.

Ge and Tassou compared a conventional R404A multiplex system with a CO_2 booster system [8]. It has been concluded that the two systems have similar performance in North England climate conditions. Furthermore, heat recovery from the CO_2 booster system is proved to satisfy 40% of space heating in the modeled supermarket. In another study based on laboratory measurements of three systems installed in Brazil, performance of a cascade R404A/CO_2 was compared with single stage multiplex R404A and R22 supermarket refrigeration systems [9]. The annual electricity use of the cascade system is measured to be 22.3% less than the R404A system and 13.7% less than the R22 system.

Computer simulation modeling and laboratory experimental tests are important to understand and predict the performance of refrigeration systems; so a performance map can be generated. They are also essential tools to compare different system solutions because it is quite difficult to compare real field

installations, where operating parameters and system requirements are not identical.
Nevertheless, it is important to study real installations where the systems are operated to fulfill the dynamic requirements of the supermarket. In field installations the system's analysis includes its response to the variable cooling and heating demands and varying boundary conditions. Such systems will also have different losses which are difficult to simulate in calculation or experimental models.

Among the researches with focus on field measurements of CO_2 supermarket refrigeration systems, Girotto et al. have evaluated the performance of a refrigeration system installed in Italy. The system is composed of three parallel CO_2 loops; two medium-temperature levels and one low-temperature level. The heat is rejected to the ambient through an indirect loop. The total annual energy consumption of the system is estimated to be 10% higher than a conventional DX R404A system solution [10]. Finckh et al. have studied the measurements of a limited set of parameters from several CO_2 trans-critical booster systems and compared the performance of a typical CO_2 booster system with a standard and an optimized R404A system. It has been shown that the CO_2 system has higher energy efficiency than standard and optimized R404A systems in ambient temperatures lower than 25°C and 21°C [3]. Rehault and Kalz have analyzed a project with the target to cut 30% energy consumption of a German supermarket chain standard system. Among other innovative energy saving measures, a CO_2 trans-critical booster system coupled with borehole sub-cooling is used. The measurements of the first year of operation have shown that the CO_2 system has similar energy efficiency as the supermarket chain conventional systems using synthetic refrigerants [11].

Despite the high number of CO_2 system installations in supermarkets, which are usually well equipped with measurements, detailed analysis of their performance is missing where losses and potential improvements in the system can be investigated. This study investigates the performance of three CO_2 trans-critical system solutions in five supermarket installations in Sweden.

2 Systems description

The two main CO_2 system solutions that are applied in supermarket refrigeration are the parallel and booster arrangements. As can be seen in Figure 1 the parallel solution consists of two separate circuits; one serves the medium temperature level cabinets and the other serves the freezers. Direct expansion (DX) is applied on both temperature levels. Two-stage compression is used for the low temperature level circuit. Since the temperature lift is relatively small on the medium temperature circuit then single stage compression is used.

In the booster system solution, similar to the units denoted as BO in Figure 2, the discharge of the booster compressor; i.e. low-stage compressor, merges with the superheated vapor exiting the medium temperature cabinets and the mixture enters the high stage compressor.

The systems analyzed in this paper are:

2.1 CO_2 trans-critical system 1 (TR1)

The TR1 supermarket is located in the north of Sweden and has been open since autumn 2007. The estimated maximum compressor cooling capacity is 230 kW for medium temperature level and about 60 kW for low temperature level.

Figure 1is a simple schematic of the TR1 system. As can be seen in the schematic, the heat rejection loop is connected to a heat pump for space heating in the supermarket.

In this system there are four separated trans-critical units, two for the medium temperature cabinets and two for the low temperature. Each medium temperature unit is equipped with four single stage compressors. Each low temperature unit is equipped with 2 two-stage compressors.

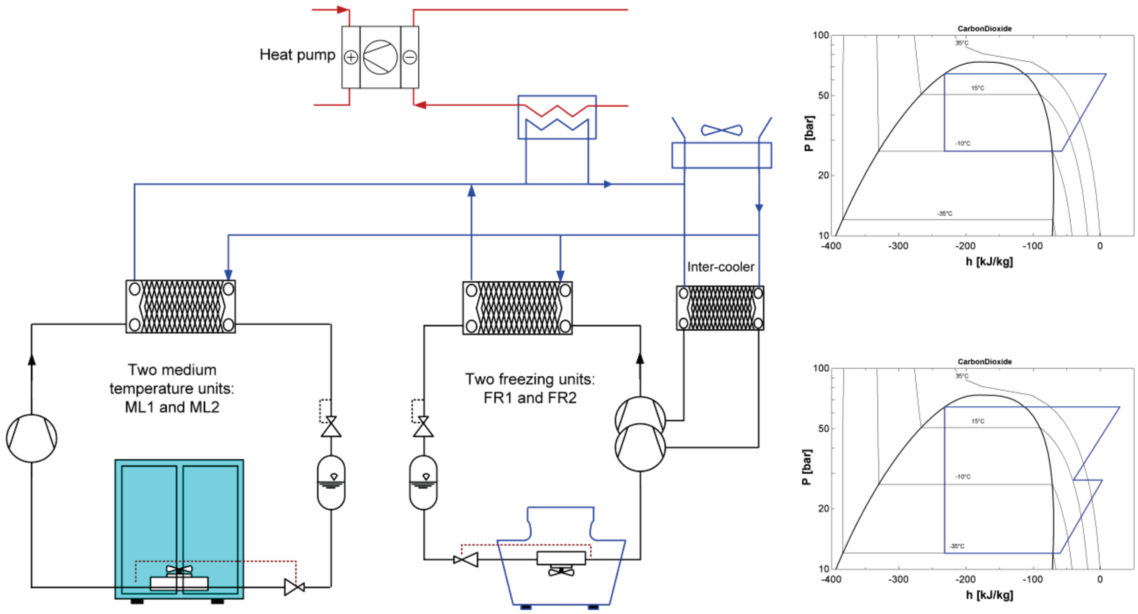

Figure 1: A simple schematic diagram of TR1 system (left) a sample P-h diagram for medium temperature stage (top-right) a sample P-h diagram for low temperature stage (bottom-right)

2.2 CO_2 trans-critical systems 2 and 3 (TR2 and TR3)

The supermarket TR2 is located in the south-western coast of Sweden and has been open since august 2008. The maximal compressor cooling capacity is about 210 kW medium temperature level and about 70 kW for low temperature level. Figure 2 is a schematic of the TR2 system. There are three separated trans-critical units, two booster types (denoted as BO1 and 2) for the medium and low temperature cabinets with a load ratio (i.e. ratio between cooling capacities at medium and low temperature levels) of about 2, and one single stage unit for the rest of the medium temperature cabinets. Each booster unit has three compressors at the medium temperature level and two booster compressors. The separate medium temperature unit has four single stage compressors.

As can be observed in the schematic in Figure 2, heat is recovered from the refrigeration system through the de-superheaters for the use in space heating. The refrigeration system is sub-cooled by being connected to a ground source heat pump loop. The rest of the heat from the system is rejected directly to ambient.

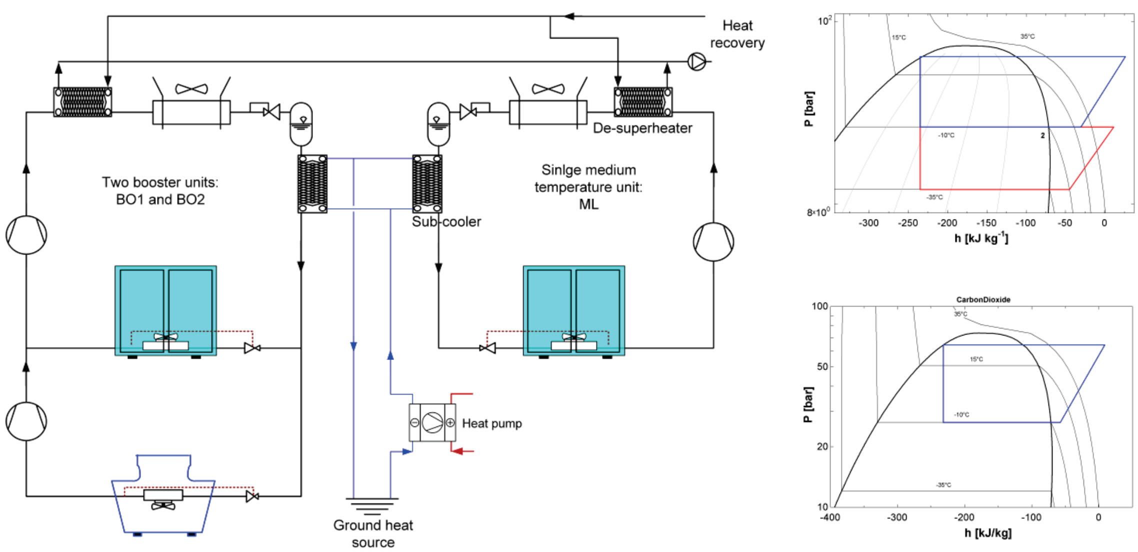

Figure 2: A simple schematic diagram of TR2 system (left) a sample P-h diagram for two stage booster unit (top-right) a sample P-h diagram for single medium temperature stage (bottom-right)

The supermarket TR3 is located in the south-western coast of Sweden and has been open since February 2010. The estimated maximum compressors cooling capacity is about 230 kW for the medium temperature level and about 60 kW for low temperature level. TR3 has two separate booster units similar to the units BO1 and BO2 in Figure 2, the system does not have ground source sub-cooling as in the case of TR2. One booster unit has four compressors at the medium temperature level and the other has five. Each unit has two booster compressors. As in the case of TR2 heat is recovered from the system via a de-superheater. The rest of the heat is rejected directly to the ambient.

2.3 CO_2 trans-critical systems TR4 and TR5

TR4 and TR5 are booster systems that are similar in design concept but mainly differ in capacity and location. A simple schematic of the booster solution applied in systems TR4 and TR5 is presented in Figure 3.

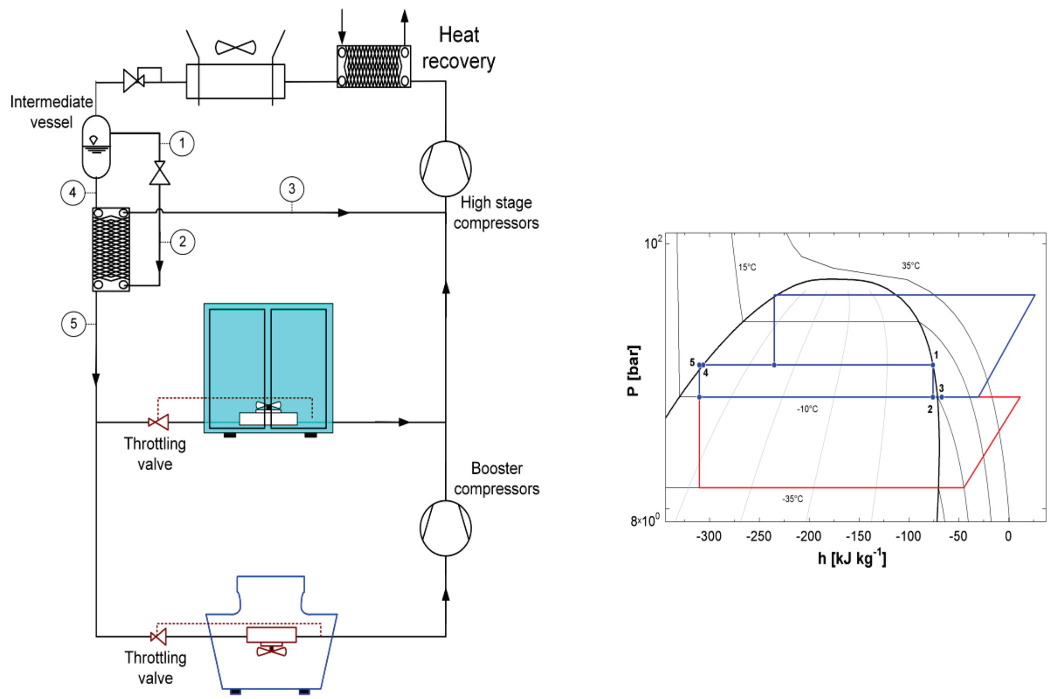

Figure 3: A simple schematic diagram of TR4 and TR5 systems (left) a sample P-h diagram (right)

The main difference between this solution and the booster system applied in TR2 and TR3 is the removal of the gas from the intermediate vessel which is then expanded in a heat exchanger to evaporate the liquid formed after the expansion, at the same time sub/further-cool the liquid before the expansion valves.

The removal of the gas from the intermediate vessel will reduce the throttling losses in the low stage cycle. The improvement on the cycle can be observed in the process plot in the P-h diagram in Figure 4a by the highlighted area. The booster system without the gas removal from the intermediate vessel will have the process plotted in Figure 4b.

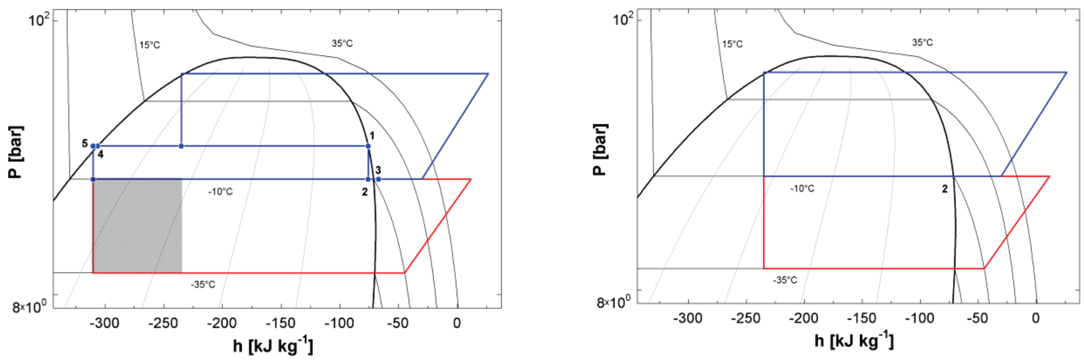

Figure 4: Plots of the processes of booster systems with (a) and without (b) gas removal from an intermediate vessel

It can be observed in Figure 4a that the achieved sub-cooling (point 4 to 5) is small, the calculation in this case is done for a Load Ratio (LR) of 3. The sub-cooling effect is expected to be small due to the relatively small vapor flow and the small temperature difference across the heat exchanger.

The two systems TR4 and TR5 are located in the southern part of Sweden, on the southwest and southeast coasts. Both systems have been in operation since the beginning of May 2010.

The maximal compressor cooling capacity in TR4 is about 25 kW for the medium temperature level and about 6 kW for low temperature level. TR5 is larger system with capacities of 130 and 30 kW at the medium and low temperature levels respectively. Each system consists of a single booster unit.

3 Measurements and evaluation method

The main parameters usually used for the evaluation of refrigeration systems are the cooling capacity and the COP. The COP of the system is the ratio of its cooling capacity and the electric power consumption of the system. The cooling capacity can be determined by measuring/estimating the mass flow rate of the refrigerant and the enthalpies at the inlet and exit points of the evaporators. The enthalpies are determined in the system by measuring the pressures and temperatures at the needed points. Performing a direct mass flow rate measurement is expensive and therefore usually not applied in field installations. Instead, indirect methods can be used such as extracting performance values from the compressor manufacturer data or performing energy balances around the compressor.

The following sub-sections explain in details the main measurements performed and the evaluations methods:

3.1 Pressure and temperature measurements

In order to obtain the thermo-physical properties of the refrigerant at any point of the refrigeration process two properties are needed, pressure and temperature are therefore measured.

A single pressure measurement is installed at each pressure level in each unit in the system. For instance, the medium temperature unit in TR1 has three pressure measurements; evaporation, condensation, and a measurement between the regulating valve (after the gas cooler) and before the evaporator's expansion valve. Compared to the medium units, the freezing units in TR1 have one additional pressure measurement in each, measuring the intermediate pressure between the two-stage compression.

In the booster units in the TR2 and TR3 systems four pressure measurements were installed in each. Additional measurement point is installed in the booster systems in TR4 and TR5 to measure the pressure in the intermediate vessel.

Pressure sensors give an absolute or relative pressure reading depending on their initial settings. The pressure reading is double checked with the corresponding measured evaporation temperature in order to ensure that the used value is accurate.

Temperature measurements have been installed before and after each of the main components in the systems. At some points it is difficult to measure the temperature, especially when the distribution and return lines split or merge. For example, for the calculation of the cooling capacity the internal superheat value is needed and it is difficult to obtain because the return lines of the different cabinets merge at different points of the piping network and may include considerable amount of external superheat.

The internal superheat can be estimated by extracting actual data from the control system in the cabinets. This will be lengthy and complicated process if to be done for each time step of the measurements; therefore, the internal superheat is estimated by selecting a number of cabinets in each system where the direct measurement is obtained. The resulting average internal superheat is then used for the whole system.

Outdoor temperature measurement is important to study its influence on the system's performance. The installed temperature measurement in the supermarket could be affected by different parameters such as its location from the sunlight and ventilation exhausts, etc. Therefore, the ambient temperatures were always compared to data from the closest Swedish Meteorological and Hydrological Institute [12] weather station in the region. Whenever a reasonable agreement is established the site measurement was directly used; otherwise, the weather station data was used.

3.2 Electrical power consumption

Electric power consumption measurement for each compressors group in a unit or at a certain pressure level should be done on a separate measurement channel. This is essential in order to perform the analysis of the system since the compressors can be used to estimate the mass flow rate of refrigerant and calculate the system's cooling capacity and COP. Therefore; other components such as fans, pumps, cabinets, etc. should be measured separately. This is unfortunately, not usually the case in field installations where several components could be connected with the compressors at a single measurement points.

In the system TR1 the electric power consumption for each unit was measured separately. In TR3, the booster compressors in each unit were measured separately; same is done to the high stage compressors.

In TR2 system, in one of the booster units the low and high stage compressors were measured at a single channel. The other booster unit in TR2 has separate measurements for the booster and high stage compressors. Since both booster units have the same design capacity then it was assumed that both units have the same division of power/capacity between the booster and high stage compressors.

In systems TR4 and TR5 was that the electric power consumption of the gas coolers fans was included in the measurements of the high stage compressors.

The running capacities of the gas coolers fans were recorded as percentage of their full capacity. The gas coolers manufacturer data was used to estimate the corresponding energy use to the recorded capacity percentage. This value is then deducted from the actual measurement and the energy use of the compressor is then estimated.

3.3 Mass flow estimation

The compressor's performance data provided by the manufacturer is used to calculate its total efficiencies at different operating conditions; i.e. pressure ratios. This can then be used to estimate the mass flow rate of the refrigerant at the compressors running conditions.

The total efficiency of the compressor can be calculated using the following equation:

$$\eta_{tot} = \frac{\dot{m}_r \cdot \left(h_{comp,in} - h_{comp,out,is}\right)}{\dot{E}_{el}} \qquad 1$$

where the mass flow rate of refrigerant is calculated using equation (2); it is the cooling capacity divided by enthalpy difference over the evaporator:

$$\dot{m}_r = \frac{\dot{Q}_2}{\Delta h_{ev}} \quad 2$$

The total efficiency is calculated and curve fitted for a matrix of evaporating and condensing temperatures/pressures.

To use this method to estimate the mass flow in the system, the suction pressure, suction temperature, discharge pressure, and the electric power consumption of the compressor must be measured. Then all the parameters in equation (3) are obtained and therefore the mass flow rate of refrigerant is estimated.

The uncertainty of the estimated mass flow rate can be calculated by knowing the temperature, pressure, power and total efficiency accuracies. As an example of the uncertainty analysis with typical instruments, a PT 1000-class A temperature sensor has a tolerance of ±0.15°C and an AK-2050 pressure transmitter has a max tolerance of ±0.8%. A typical power meter has 0.5% uncertainty. It is hard to know the exact accuracy of the total efficiency as the compressor manufacturer data rarely includes the accuracy information of the cooling capacities and powers. Assuming two cases of 0.5% and 2% tolerances for the total efficiency, estimated mass flow rate has an uncertainty ranging between 1.32% and 2.34%.

A main advantage in this method is that the number of compressors running is not needed for the estimation of the total mass flow rate; single electric power consumption measurements for all compressors in parallel can be used.

3.4 Data acquisition, synchronization and filtering

Two data acquisition systems have been used; IWMAC [13] for systems TR1, TR2 and TR3 and LDS [14] for TR4 and TR5.

IWMAC is a web-based software used for remote monitoring and control of mainly HVAC systems in buildings and refrigeration plants in retail industry [13]. Each measured value is stored in a database which can be accessed online where data can be downloaded to text files. Data scan in IWMAC is performed every few seconds for every parameter, the value at a certain time is recorded only if it is different than that of the previous time step. This resulted in the values of the different parameters being recorded at different time intervals. In order to put all the data at the same time steps, data values have been averaged every 10 minutes.

The data collected using LDS has a constant time interval of 2 minutes which is then averaged to every 10 minutes. In LDS [14] system for data acquisition the data is collected and saved on an on-site computer. Data can then be accesses from any computer by connecting to the logging computers via a modem, specifying which parameters to collect. In LDS, it is possible to write scripts, determining which data to be collected, for which period of time and when the collection should take place. This way, the data acquisition is made automatically on a regular basis. A disadvantage in this method is that the on-site logging computers have a limited capacity to store data, which can be lost if the download is not done in a good frequency or if the dial up fails. This resulted in data loss ranged from few minutes to few days in the worst case; however, the impact is expected to be rather small on the results because monthly averages have been used in the analysis of this study.

In the calculation templates the main parameters for the analysis, mainly cooling capacity and COP, are calculated for each time step. Then the data is filtered so the point with negative cooling capacity values or higher than the system's capacity are removed. Also points with negative or unrealistically high COP's, for instance higher than 7, are removed. The monthly averages are based on the filtered data.

3.5 Cooling capacity and COP calculations

The cooling capacity is calculated from equation (2). The enthalpy at the evaporator inlet is obtained from the measured temperature and pressure before the expansion valve. The enthalpy at the evaporator exit is calculated using the evaporation pressure and the assumed average internal superheat values, explained in section 3.1. Average internal superheat values for the medium and low temperature level cabinets are listed in Table 1.

Table 1: Average internal superheat values for medium and low temperature cabinets

	TR1		TR2		TR3		TR4		TR5	
	ML	FR	ML	FR	ML	FR	ML	FR	ML	FR
Internal SH (K)	10	13	13	11	10	8	6	6	8	8

The COP of the refrigeration system is the ratio of the cooling capacity to the electric power consumption as in the following equation

$$COP = \frac{\dot{Q}_2}{\dot{E}_{el}} \tag{3}$$

In this analysis only compressors electric power consumption is included in the COP calculations. Other parameters such as the fans of the gas coolers, the pumps, the cabinets, etc. were not included. Equation 3 is used to calculate the medium and low temperature COP's of the system TR1.

The medium temperature level COP for the booster system is calculated as

$$COP_{ML} = \frac{\dot{Q}_{ML,tot}}{\dot{E}_{hs}} \tag{4}$$

where

$$\dot{Q}_{ML,tot} = \dot{Q}_{ML,cab} + \dot{Q}_{FR} + 0.93 \cdot \dot{E}_{ls} \tag{5}$$

where $\dot{Q}_{MT,cab}$ is the cooling capacity at the medium temperature cabinets in kW. $\dot{E}_{el,ls}$ is the electric power consumption of the low temperature booster compressors (kW). The factor 0.93 is to account for 7% heat loss from the compressor to the surroundings [14].

The total high stage electric power consumption is assumed to be divided into two portions; one for the medium and one for the low temperature capacity

$$\dot{E}_{hs} = \dot{E}_{ML,cab} + \dot{E}_{hs,for,FR} \qquad 6$$

where $\dot{E}_{MT,cab}$ is the portion of high stage compressors power that covers the cooling capacity of the medium temperature cabinets. The rest, $\dot{E}_{hs,for,FR}$, is the power to cover the portion of capacity coming from the low temperature level to the medium temperature level; i.e. $\dot{Q}_{FR} + 0.93 \cdot \dot{E}_{ls}$. It is calculated as:

$$\dot{E}_{hs,for,FR} = \dot{E}_{hs} - \dot{E}_{ML,cab} = \dot{E}_{hs} - \frac{\dot{Q}_{ML,cab}}{COP_{ML}} \qquad 7$$

The low temperature COP for a booster unit is then calculated as

$$COP_{FR} = \frac{\dot{Q}_{FR}}{\dot{E}_{FR} + \dot{E}_{hs,for,FR}} \qquad 8$$

From any system in this study, the total COP is defined as the ratio of the total cooling capacity of the system, at the medium and low temperature levels, to the electric power consumption of all its compressors. Expressed as

$$COP_{tot} = \frac{\dot{Q}_{ML}+\dot{Q}_{FR}}{\dot{E}_{ML} + \dot{E}_{FR}} \qquad 9$$

This expression of the total COP is sensitive to the load ratio (LR) of the medium to the low temperature level, expressed in equation 10.

$$LR = \frac{\dot{Q}_{ML}}{\dot{Q}_{FR}} \qquad 10$$

Systems with the same medium and low temperature levels COP's but with different load ratios will have different total COP's. $\dot{Q}_{FR}$ in equation 9 can be substituted with its equivalent in equation 10.

$$COP_{tot} = \frac{\dot{Q}_{ML} + \frac{\dot{Q}_{ML}}{LR}}{\dot{E}_{ML} + \dot{E}_{FR}} \quad 11$$

For $\dot{E}_{ML}$ and $\dot{E}_{FR}$, the cooling capacities over the corresponding medium and low temperature levels COP's can be used.

$$COP_{tot} = \frac{\dot{Q}_{ML} + \frac{\dot{Q}_{ML}}{LR}}{\frac{\dot{Q}_{ML}}{COP_{ML}} + \frac{\dot{Q}_{FR}}{COP_{FR}}} \quad 12$$

$\dot{Q}_{FR}$ can be substituted from equation 10. Dividing by $\dot{Q}_{ML}$, the result is the following expression which is the total COP at certain load ratio:

$$COP_{tot,LR} = \frac{1 + \frac{1}{LR}}{\frac{1}{COP_{ML}} + \frac{1}{LR \cdot COP_{FR}}} \quad 13$$

Typical load ratio value in an average size supermarket system in Sweden is approximately 3.

4 Systems analysis

4.1 TR1

Data for the TR1 system has been collected for the period of January 2008 to June 2009. The two cooling units at the medium temperature level, ML1 and ML2, are identical. Same is true for the units at the low temperature level; FR1 and FR2.

Total cooling capacities in the cabinets at the medium and low temperature levels are plotted in Figure 5. Outdoor, condensing and evaporation temperatures are also plotted. Condensing temperatures presented in the plot are the average values of the condensing temperatures of all units, which had small deviations from the average value. Each point in the plot is the monthly average value of the corresponding parameter.

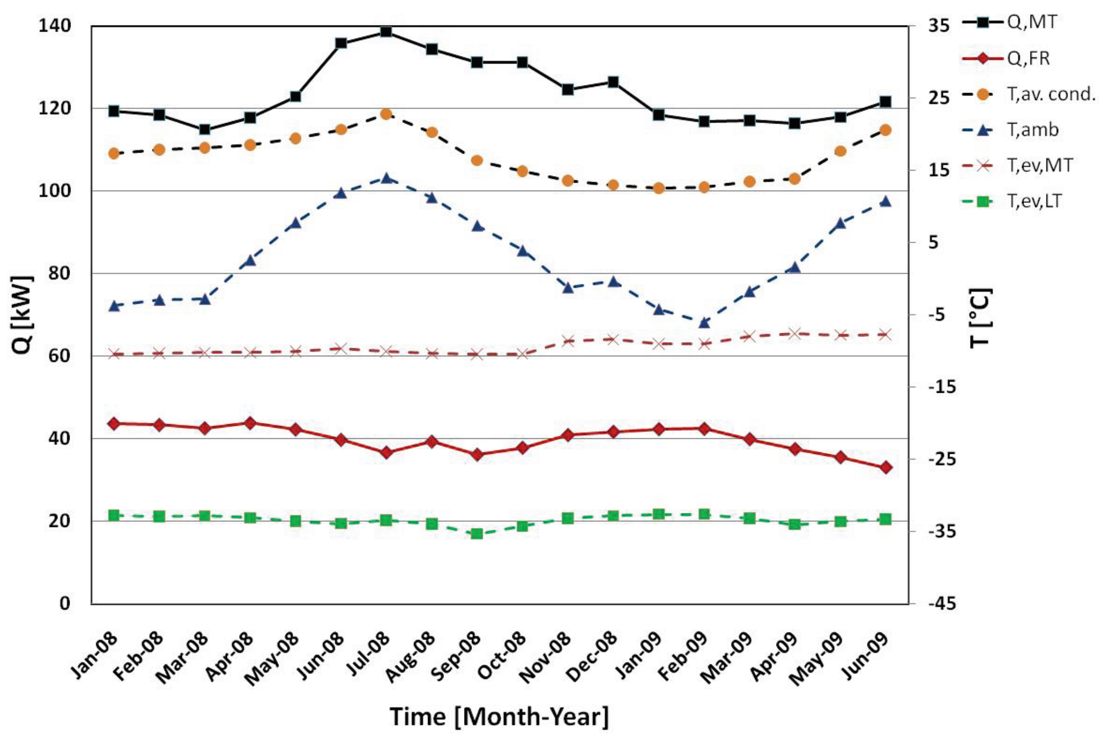

Figure 5: Monthly average values of total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures. The period is January 2008 to June 2009 for the system TR1

As can be observed in the plot, the evaporation temperatures at the medium and low levels are almost constant; about -9 and -34°C. Cooling capacity at the low temperature level did not vary much between winter and summer periods. This is mostly due to the use of lids that separates the cold air in the freezers from the supermarket environment, which has relatively higher moister content especially in the summer period.

However, it can be observed that the cooling capacity at the medium temperature level increases during the summer/warm period due to the higher humidity in the store area and the lack of the doors in most of the medium temperature cabinets, air curtains were applied instead. The cooling capacity at the medium temperature level is about 3 times higher than at the low temperature level.

It can be observed in the plot that the condensing temperature follows the ambient temperature but with relatively high temperature difference which is due to the use of the coolant loop between the refrigeration circuits' condensers/gas coolers and the dry cooler. The coolant loop was applied in order to couple the refrigeration system with the heat pumps, as can be seen in Figure 1. However, due to technical problems, which were not related to the solution itself, the heat pumps were not in operation during the study period and the system was controlled to operate at lower condensing temperature after the winter of 2008, from about 17°C to 13°C.

Medium and low temperature COP's of the system TR1 are presented in Figure 6. As can be observed in the plot the COP increases with decreasing condensing temperatures. It can also be observed in the plot that the system has generally low sub-cooling.

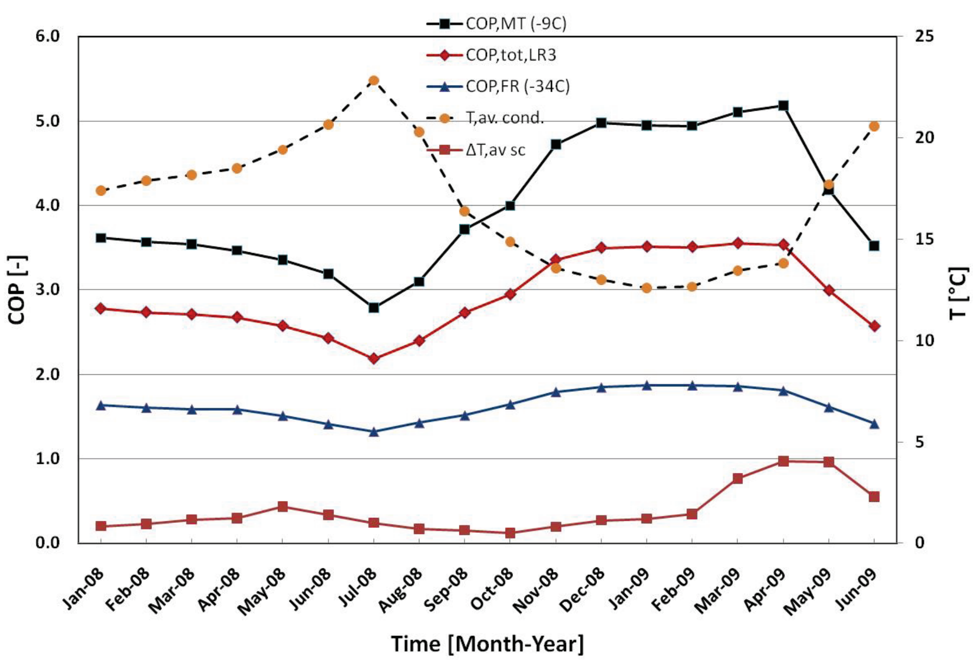

Figure 6: Medium temperature level COP, low temperature level COP, total COP, condensing temperature, and sub-cooling temperature difference. The period is January 2008 to June 2009 for the system TR1.

The total COP plot is for a load ratio of 3 using the expression in equation 13.

4.2 TR2

Data for the TR2 system has been collected for the period of September 2008 to June 2009. The two booster units, BO1 and BO2, are identical. The third unit in the system, ML, serves cabinets at the medium temperature level.

Similar parameters are plotted for TR2 as for TR1 in the previous section, presented in Figure 7 and Figure 8.

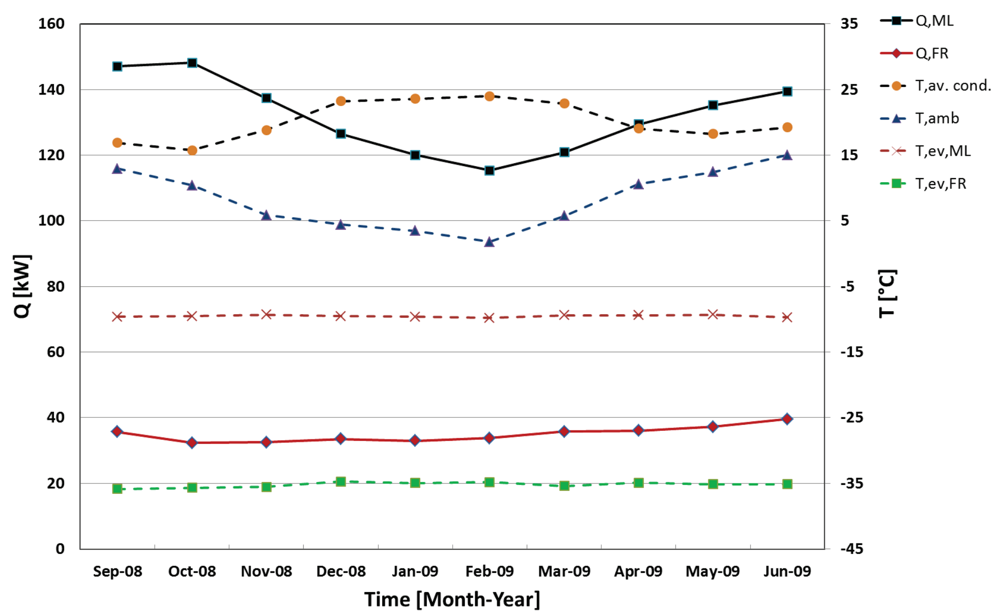

Figure 7: Monthly average values of total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures. The period is September 2008 to June 2009 for the system TR2.

It can be observed in Figure 7 that similar trends in the parameters of the system TR1 are also observed in TR2, except that the condensing temperature is relatively high at low outdoor temperatures. This is due to the need to increase the discharge pressure of the refrigeration system to recover heat in the de-superheater at low outdoor temperatures.

Figure 8 is a plot of the system's COP's, condensing temperature and sub-cooling temperature difference provided by the borehole loop.

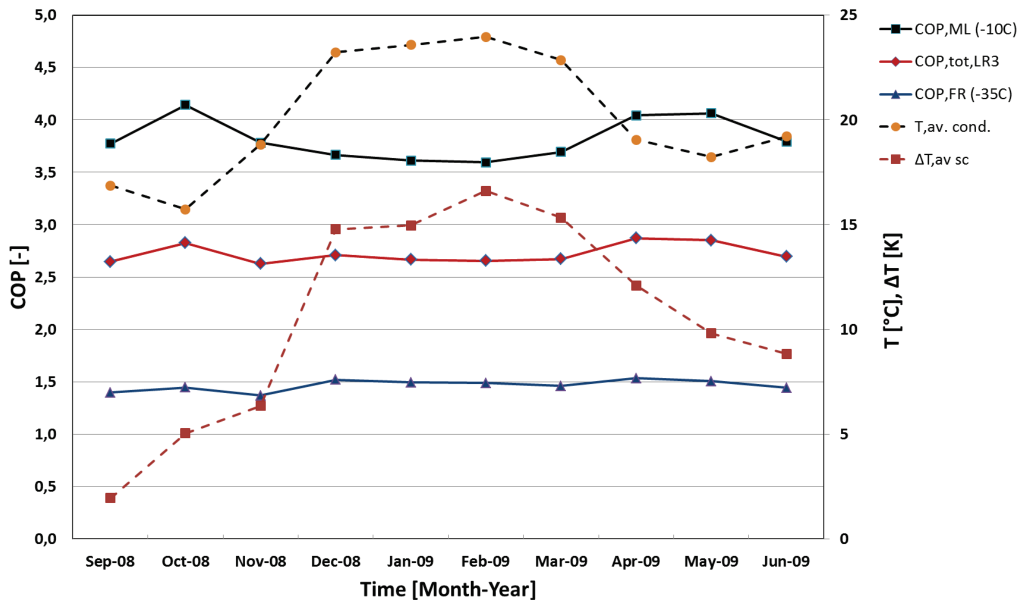

Figure 8: Medium temperature level COP, low temperature level COP, total COP, condensing temperature, and sub-cooling temperature difference. The period is September 2008 to June 2009 for the system TR2.

As can be observed in Figure 8 the COP does not decrease with increasing the condensing temperature, this is due to the higher sub-cooling provided by the borehole circuit, up to 17K, which is coupled to a heat pump for space heating and operated during the winter period, see Figure 2. The borehole sub-cooling is not operated during the summer period.

4.3 TR3

Data for the TR3 system has been collected for the period of February to November 2010. Similar trend in the parameters in Figure 9 is observed in TR3 compared to TR2.

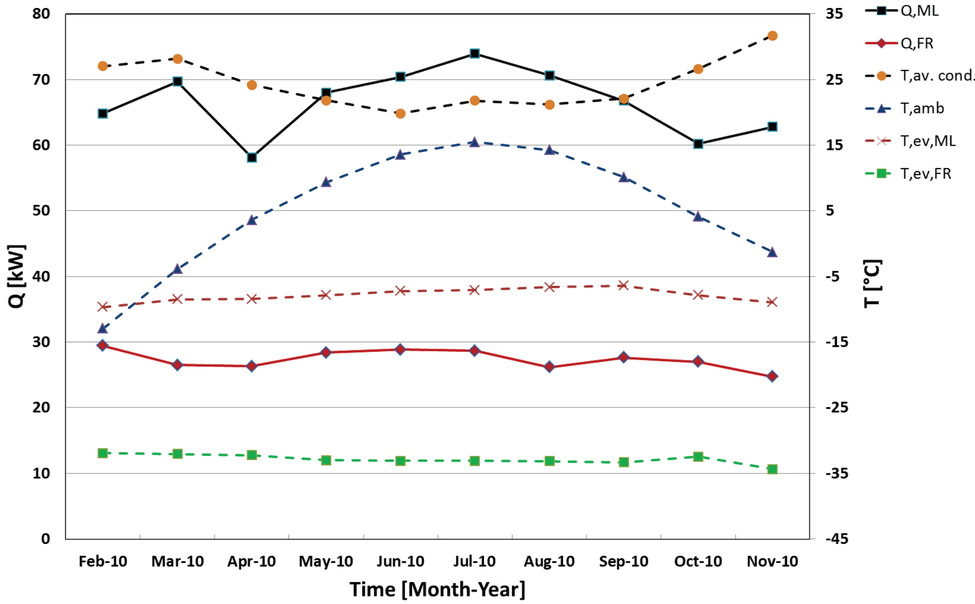

Figure 9: Monthly average values of total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures for the system TR3. The period is February to November 2010.

The condensing temperature plot in Figure 9 indicates the system control for heat recovery, similar to the case of the system TR2. The condensing pressure is kept high at low outdoor temperatures when heating is needed.

Figure 10 is a plot of the system's COP's, condensing temperature and sub-cooling temperature difference provided by the condenser/gas cooler.

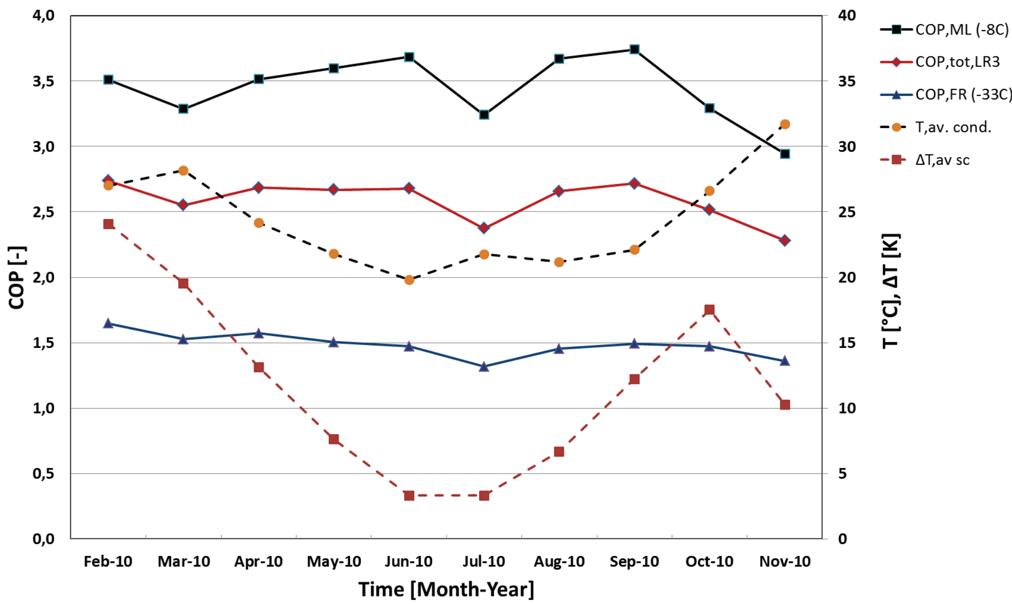

Figure 10: Medium temperature level COP, low temperature level COP, total COP, condensing temperature, sub-cooling temperature difference. The period is February to November 2010 for the system TR3.

It can be observed in Figure 10 that the system's COP's do not always decrease with increasing the discharge pressure and do not vary much between winter and summer months. In the winter period the discharge pressure is increased to provide the required heating capacity, at the same time the condenser/gas cooler is operated to provide sub/further cooling using the cold outdoor temperatures. This can be observed in the sub-cooling plot in Figure 8 which has a value up to 25K in the month of February. It gives similar effect as the borehole does in TR2 system.

4.4 TR4 and TR5

Data for the TR4 and TR5 systems have been collected for the period of June to September 2010. Both systems have been discussed in the same section since they have similar design and data collection period. The following figures, Figure 11 and Figure 12, display plots of the total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures for systems TR4 and TR5.

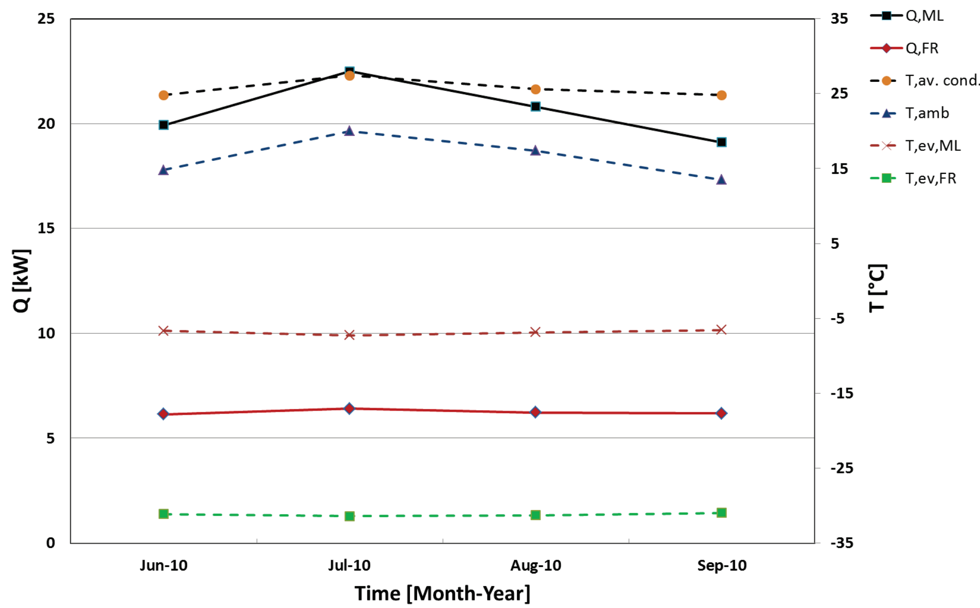

Figure 11: Monthly average values of total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures for the system TR4. The period is June to September 2010.

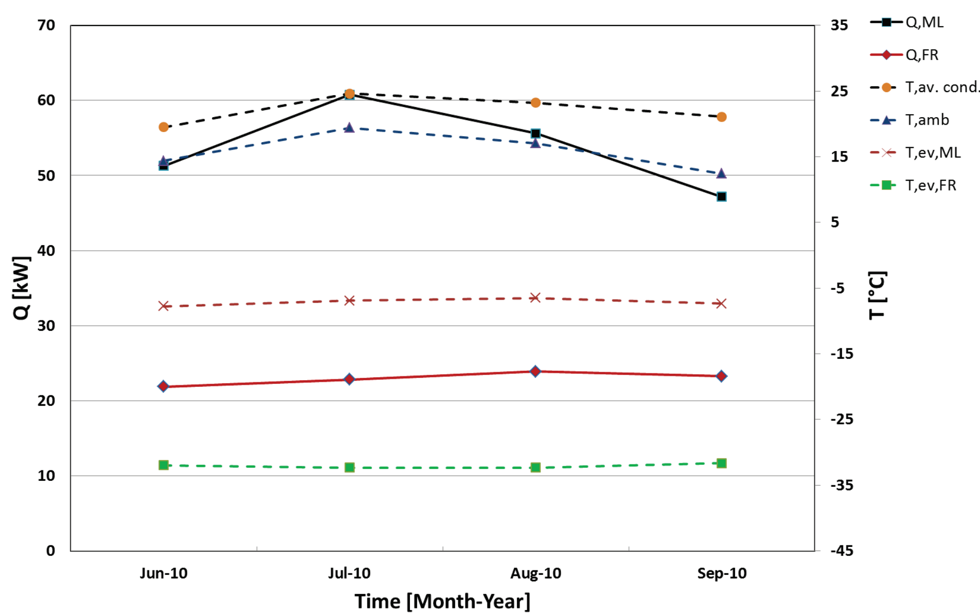

Figure 12: Monthly average values of total cooling capacities in the cabinets at the medium and low temperature levels, outdoor, condensing and evaporation temperatures for the system TR5. The period is June to September 2010.

It can be observed in the preceding two figures that temperatures for both systems are comparable where the evaporation temperatures for TR4 and TR5 are relatively higher than the TR1, TR2 and TR3. It can also be observed from the figures that the system TR5 has much higher capacities than TR4.

Figure 13 and Figure 14 present plots of the COP's, condensing temperature and sub-cooling temperature difference for the systems TR4 and TR5.

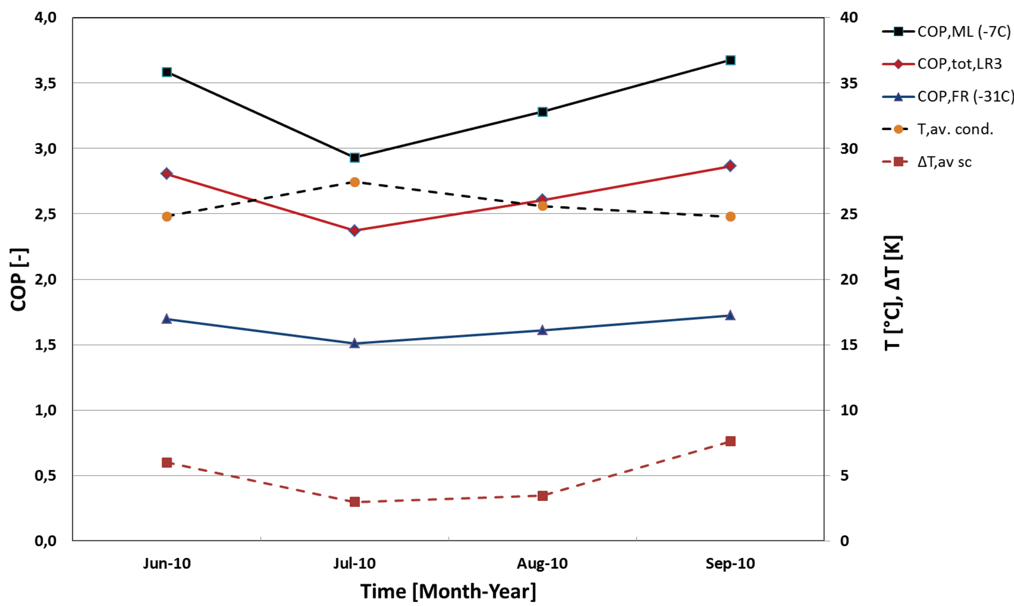

Figure 13: Medium temperature level COP, low temperature level COP, total COP, condensing temperature, sub-cooling temperature difference. The period is June to September 2010 for the system TR4.

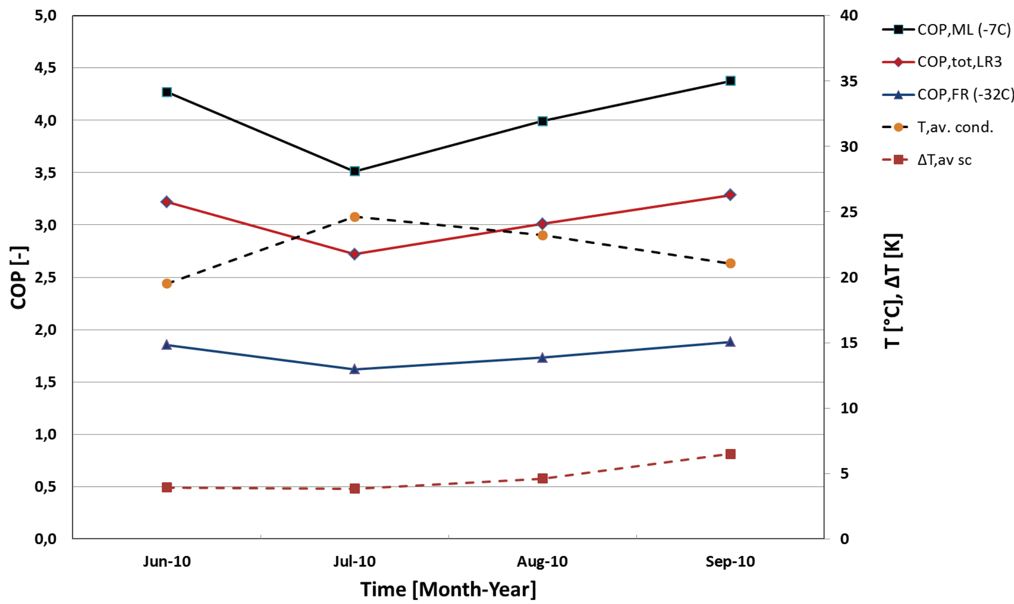

Figure 14: Medium temperature level COP, low temperature level COP, total COP, condensing temperature, sub-cooling temperature difference. The period is June to September 2010 for the system TR5.

It can be observed in the plots that the sub-cooling temperature differences in both systems are relatively small compared to the other systems. This is because the data have been collected during the summer period where discharge pressure should be kept at as low as possible and there is no need for heat recovery.

5 Systems Comparison

The COP's of the different systems are plotted against the condensing temperatures, in Figure 15, where the evaporation temperature for each system is indicated at the corresponding legend. Comparing the systems at the same condensing temperature will eliminate the influence of the temperature difference between condensing temperature and outdoors at the heat rejection side. System TR1 is the only system that has a coolant loop at the heat rejection side, as seen in Figure 1, which means that it will have higher condensing temperature than the rest of the systems when the systems are not controlled for heat recovery.

If the systems are controlled to recover heat then the discharge pressure and the corresponding condensing temperature at low outdoor temperatures will depend on the heating capacity needed to be extracted from the system, which will influence the cooling COP of the system. This will make it difficult to compare the cooling COP of the different systems based on the outdoor temperatures, this is why they condensing temperature is used instead of outdoors temperature.

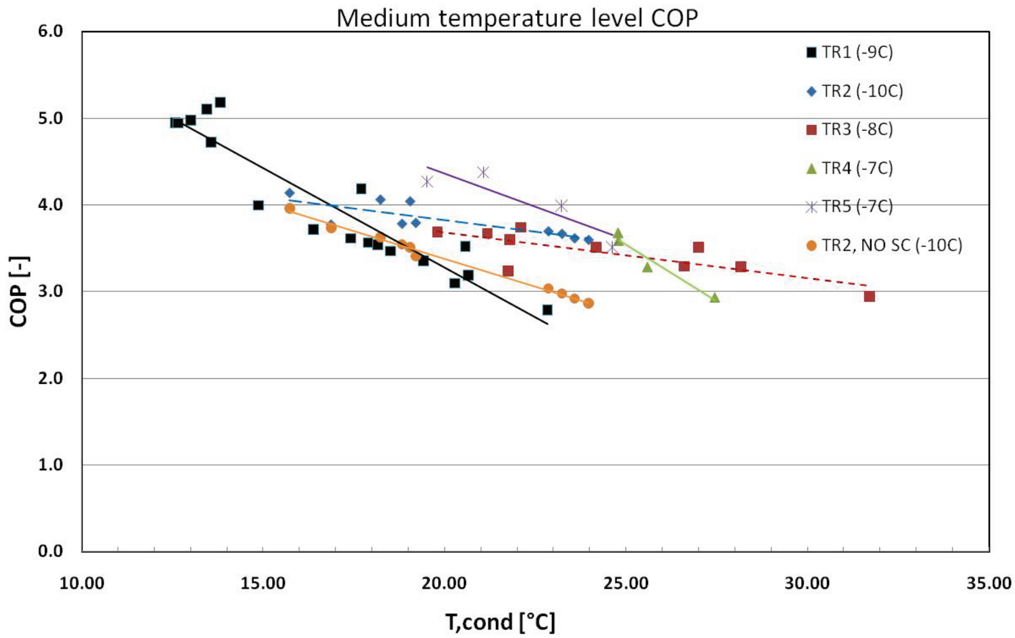

Figure 15: Medium temperature level COP’s at different condensing temperatures for all systems. Evaporation temperature for each system is indicated at the plot’s legend.

Based on the plots in this figure and on the design of the systems and their performances, they can be divided into three groups; “TR1”, “TR2 and TR3”, and “TR4 and TR5”. It can be observed that the system TR1 operates in a relatively low condensing temperatures though it has coolant loop at the warm side, this is mainly because it is installed in a location in the north of Sweden which is relatively cold and it is also not controlled for heat recovery.

Systems TR4 and TR5 on the other hand have been analyzed during a relatively warm period of the year, June to September, and they are located in the south of Sweden not far away from each other. This can be observed in the outdoor temperatures plots in Figure 11 and Figure 12 which shows that the values are quite similar for both systems.

The difference in condensing temperatures between TR4 and TR5, observed in Figure 15, is due to better performance of the condenser/gas cooler in TR5; approach temperature difference is 1-2K compared to around 4K in case of TR4.

TR4 and TR5 have relatively high medium temperature COP compared to TR1. This can be attributed to several factors such as the lower internal superheat, as can be seen in Table 1, about 10% higher total medium temperature compressor efficiency (60% for TR1 compared to 70% for TR4 and TR5), and higher evaporation temperature for the systems TR4 and TR5. External superheat is comparable between these three systems; around 8-9K. Table 2 is a list of the external superheat values at medium and low temperature levels for all systems.

Table 2: Average external superheat values at medium and low temperature levels

	TR1		TR2		TR3		TR4		TR5	
	ML	FR	ML	FR	ML	FR	ML	FR	ML	FR
External SH (K)	8	31	18	16	15	20	8	12	9	14

As can be observed in Figure 15, the systems TR2 and TR3 have comparatively high COP at high condensing temperatures, i.e. the systems do not have the same drop in COP as in the cases of TR1, TR4 and TR5 with increasing condensing temperature. The reason is mainly due to the use of the borehole sub-cooling in system TR2, see Figure 8, and sub-cooling in the condenser/gas cooler in TR3, see Figure 10. This is done in both systems when operated in the heat recovery mode. If the system TR2 is calculated without including the influence of the borehole sub-cooling then the plot labeled as “TR2, NO SC (-10C)” in Figure 15 can be observed to be at similar level to TR1 system.

Sub-cooling the supermarket refrigeration system by using a borehole is not a conventional solution therefore the improvements in COP in TR2 compared to the other systems can be argued to come from special system boundaries and not related to the refrigeration system solution. However, the system TR3 uses the cold ambient air to sub-cool the refrigerant in the heat recovery mode; up to 25K of sub-cooling can be observed in Figure 10. This control strategy of TR3 in heat recovery mode results in comparable COP to the system TR2 with borehole sub-cooling.

Similar trends of the medium temperature level COP's can be observed for the low temperature level COP's in Figure 16. The system TR1 has higher low temperature level COP than TR2 system without the influence of the borehole sub-cooling; this is mainly due to the heat removal from the inter-cooler in TR1 at the intermediate pressure level in the two-stage compressor, that can be seen in the schematic of units FR1 and FR2 in Figure 1.

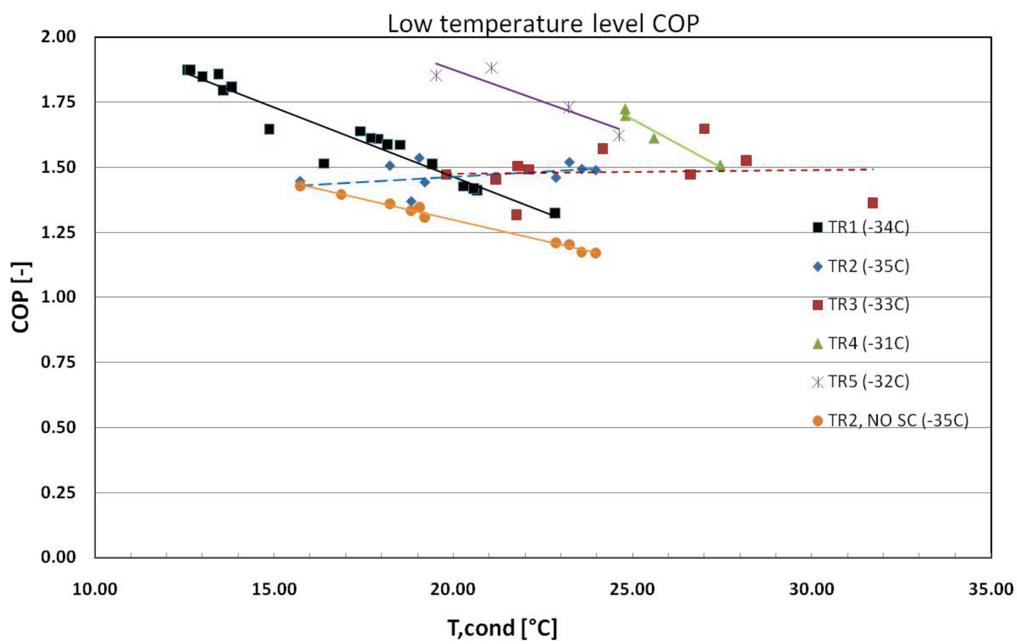

Figure 16: Low temperature level COP's at different condensing temperatures for all systems. Evaporation temperature for each system is indicated at the plot's legend.

The system solution applied in TR4 and TR5 shows the highest low temperature level COP. Some of the reasons that can explain the relatively higher COP are: the higher evaporation temperature, low internal and external superheat, listed in Table 1 and Table 2 , and the higher total efficiency of the booster compressors compared to the ones installed in TR2 and TR3; about 60% compared to 45%.

Another main reason that contributes to the higher low temperature level COP in the solution applied in systems TR4 and TR5 is the removal of the gas from the intermediate vessel which reduces the throttling losses in the low stage cycle, explained in section 2.3.

In the systems TR2 and TR3, the sub-cooling effect on increasing the low temperature level COP at high condensing temperatures in heat recovery mode can be observed.

Combining the medium and low temperature levels COP's in a total COP using the expression in equation 13 for a system with load ratio of 3 results in the plots presented in Figure 17.

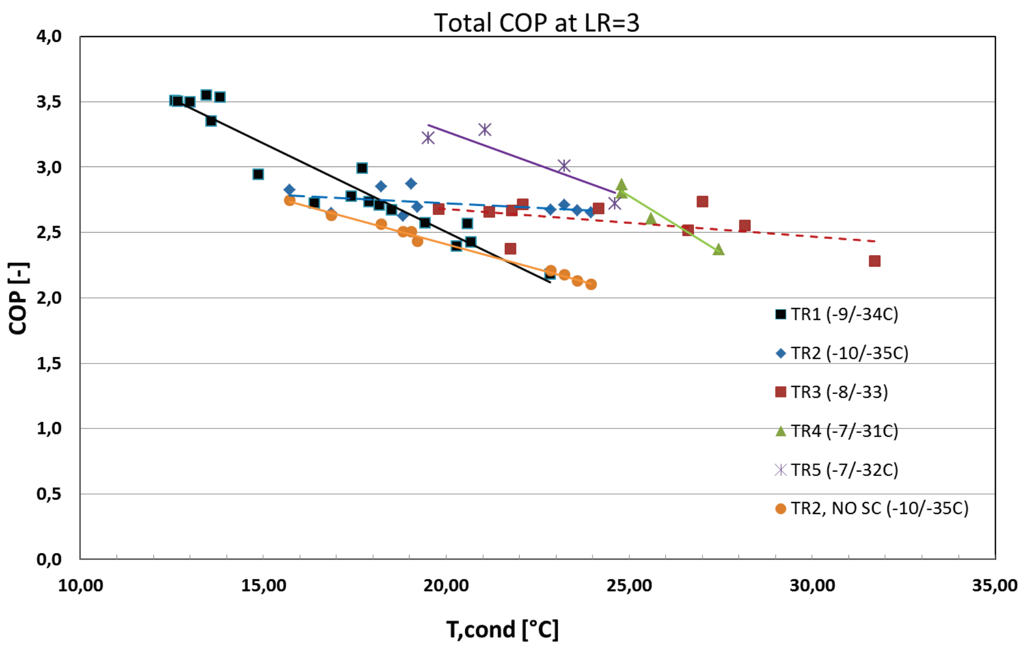

Figure 17: Total COP's at different condensing temperatures for all systems. Evaporation temperatures at medium and low levels for each system are indicated at the plot's legend.

It has to be observed that the systems have different average evaporation temperatures which are indicated at the plots' legends. The solution applied in systems TR4 and TR5 has the highest total cooling COP at most of the operating range; however, TR3 has higher COP at high condensing temperatures (i.e. discharge pressure) where high degrees of sub-cooling are achieved using the cold ambient air. It has to be noticed that the systems TR4 and TR5 were not tested during the winter period where heat recovery will be needed in relatively high capacities and then the sub-cooling effect on system improvement can be observed.

6 Conclusions

Field measurements of five supermarkets using three different CO_2 trans-critical solutions were analysed for periods of 4 to 18 months. The essential parameters to evaluate the system performances were discussed including temperatures, pressures and compressors electricity consumptions. As typically no mass flow meter is installed in the supermarkets, the total efficiency of the compressor was used to estimate the mass flow rate of refrigerant in the cycle. Performance data sheets from compressor manufacturers were used to generate the total efficiency curve at different pressure ratios.

Monthly averages of cooling demands ($\dot{Q}_2$), low temperature level COP (COP_{FR}), medium temperature COP (COP_{ML}) and total COP (COP_{tot}) were plotted for the five supermarkets. For most of the cases, the cooling capacities and COPs of the low temperature level were less fluctuating than at the medium temperature level as glass lids installed on the freezers provide isolation from fluctuating indoor humidity.

It has been observed that the running mode of the systems (floating condensation or heat recovery) and the amount of sub-cooling have a significant impact on system performance. Higher sub-cooling provides higher cooling COP at certain cooling capacity. This can be clearly seen in the results of systems TR2 and TR3 in running in the heat recovery mode.

Systems TR4 and TR5 using transcritical booster system with gas removal from the intermediate vessel have the highest total COP. For example, at 20°C condensing temperature, TR5 has a COP value of 3.3 while TR2 and TR3 have a COP value of 2.7. Gas removal from the intermediate vessel leads to higher low temperature level COP. Other reasons for this higher energy efficiency are 1-3K higher evaporation temperatures, lower internal and external superheat and 10-15% higher total efficiency of booster compressors compared to TR2 and TR3. These parameters highlight the importance of suitable control for internal superheating, good insulation to reduce external superheating and using compressors with highest efficiency available.

Analyzing the performance of these five CO_2 supermarket refrigeration systems indicates a trend of improvement in energy efficiency by up to 25-35% higher total COP comparing TR4-TR5 with TR2-TR3 or TR1, as TR1 is the oldest system installed (2007) and TR4 and TR5 are the newest ones (2010).

In part two of this paper, the performance analysis of three HFC systems will be discussed and compared to the five CO_2 systems analyzed in this paper. Furthermore, using computer simulation, a detailed parametric analysis of the different system solutions is included in the second part. This will clarify better the main contributors to improvement in systems' efficiency and supplement the study in paper 1.

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8 References

[1] Shecco, Guide 2014: Natural refrigerants continued growth & innovation in Europe, Brussels, Belgium, 2014. http://publications.shecco.com/publications/view/2014-guide-natural-refrigerants-europe.

[2] S. Sawalha, Carbon dioxide in supermarket refrigeration, Doctoral Thesis, Royal institute of technology (KTH), Stockholm, Sweden, 2008.

[3] O. Finckh, R. Schrey, M. Wozny, Energy and efficiency comparison between Standardized HFC and CO2 transcritical systems for Supermarket applications, in: IIR/IIF, Prague, Czech Republic, 2011: p. ID: 357.
[4] L. Cecchinato, M. Corradi, S. Minetto, Energy performance of supermarket refrigeration and air conditioning integrated systems, Applied Thermal Engineering. 30 (2010) 1946–1958. doi:10.1016/j.applthermaleng.2010.04.019.
[5] L. Cecchinato, M. Corradi, S. Minetto, Energy performance of supermarket refrigeration and air conditioning integrated systems working with natural refrigerants, Applied Thermal Engineering. 48 (2012) 378–391. doi:10.1016/j.applthermaleng.2012.04.049.
[6] V. Sharma, B. Fricke, P. Bansal, Comparative analysis of various CO2 configurations in supermarket refrigeration systems, International Journal of Refrigeration. 46 (2014) 86–99. doi:10.1016/j.ijrefrig.2014.07.001.
[7] A. Mikhailov, H.O. Matthiesen, System efficiency for natural refrigerants, ASHRAE Journal. (August 2013).
[8] Y.T. Ge, S.A. Tassou, Performance evaluation and optimal design of supermarket refrigeration systems with supermarket model “SuperSim”. Part II: Model applications, International Journal of Refrigeration. 34 (2011) 540–549. doi:10.1016/j.ijrefrig.2010.11.004.
[9] A. da Silva, E.P. Bandarra Filho, A.H.P. Antunes, Comparison of a R744 cascade refrigeration system with R404A and R22 conventional systems for supermarkets, Applied Thermal Engineering. 41 (2012) 30–35. doi:10.1016/j.applthermaleng.2011.12.019.
[10] S. Girotto, S. Minetto, P. Neksa, Commercial refrigeration system using CO2 as the refrigerant, International Journal of Refrigeration. 27 (2004) 717–723. doi:10.1016/j.ijrefrig.2004.07.004.
[11] N. Rehault, D. Kalz, Ongoing Commissioning of a high efficiency supermarket with a ground coupled carbon dioxide refrigeration plant, in: International Conference for Enhanced Building Operations (ICEBO), Manchester, England, 2012.
[12] SMHI, Swedish Meteorological and Hydrological Institute (2011): Available at : www.smhi.se, (2014).
[13] IWMAC, IWMAC, Centralized operation and surveillance by use of WEB technology, (2011). www.IWMAC.eu.
[14] LDS, Long Distance Service, Computer software for handling data from field measurements, (2011).
[15] K. Berglöf, Methods and potential for on-site performance validation of air conditioning, in: 8th International Energy Agency Heat Pump Conference, Las Vegas, USA, 2005: pp. P2–3.