Energy Efficiency Evaluation of Integrated CO_2 Trans-critical System in Supermarkets; a Field Measurements and Modelling Analysis

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ABSTRACT

This paper investigates energy efficiency of an integrated CO_2 trans-critical booster system installed in a supermarket in Sweden. The supermarket has applied several features to improve energy efficiency including space and tap water heating, air conditioning (AC), and parallel compression.

Using field measurements data, the system performance is evaluated in a warm and a cold month. Furthermore, this integrated energy system concept is modelled and compared with stand-alone HFC-based energy systems.

The results show that the system provides the entire AC demands and recovers a great share of the available heat, both with high COP values. The comparative analysis shows that integrated CO_2 system uses about 11% less electricity than stand-alone HFC solutions for refrigeration (i.e. indirect HFC), heating and AC in North of Europe.

Energy efficiency analysis of the integrated CO_2 system proves that this system is an environmentally friendly all-in-one energy efficient solution suitable for cold climate supermarkets.

Keywords: CO_2 trans-critical booster system, Supermarket, Field measurements, Systems integration, Modelling

NOMENCLATURE			
Abbreviations			
AC	Air conditioning	$\dot{Q}$	Cooling or heating load, kW
COP	Coefficient of performance	T	Temperature, °C
DX	Direct expansion	$T_{gc,exit}$	Gas cooler exit temperature, °C
HFC	Hydrofluorocarbon	$T_{hpv,inlet}$	High pressure regulating valve inlet temperature, °C
HP	Heat pump	T_{hr2}	2nd de-superheater exit temperature, °C
HR	Heat recovery	η	Efficiency of compressor
HRR	Heat recovery ratio		
LR	Load ratio	***Subscripts***	
LT	Low temperature level	AC	Air conditioning
min	Minutes	amb	Ambient
MT	Medium temperature level	fan	Gas cooler fans
PC	Parallel compression	gc	Gas cooler
PPM	Parts per million	hpv	High pressure valve
Ref	Refrigeration or refrigerant	HR	Heat recovery
		is	Isentropic
Variables		LT	Low temperature level
$3WV_{gc}$	3-way by-pass valve before gas cooler	MT	Medium temperature level
E	Electrical energy, MWh	PC	Parallel compression
$\dot{E}$	Electric power, kW	ref	Refrigeration
h	Enthalpy per unit mass, kJ kg^{-1}	shaft	Compressor shaft
$\dot{m}$	Mass flow rate, kg s^{-1}	SPH	Space heating
P	Pressure, bar	tot	Total
P_{gc}	Gas cooler pressure, bar	TWH	Tap water heating
		xSH	External superheating

1. INTRODUCTION

Spread of using CO_2 trans-critical booster systems in supermarkets has been accelerating during the past years in Europe due to EU F-gas regulation (EU 517/2014, 2014). As of January 2016, more than 7000 supermarkets in the world have installed CO_2 trans-critical booster system, out of which 5500 in Europe (Shecco, 2016).

CO_2 trans-critical systems have similar or higher cooling COP compared to conventional hydrofluorocarbon (HFC) system at ambient temperatures lower than about 25°C (Finckh et al., 2011) (Sawalha et al., 2017). This is why they have been popular in new installations in the relatively cold climate of northern Europe. The CO_2 trans-critical systems offer another main advantage in cold climates, which is to recover all, or most, of the heat for space heating at relatively high efficiency (Reinholdt and Madsen, 2010) (Sawalha, 2013); therefore, it became a common practice in the past years to install CO_2 refrigeration systems with heat recovery.

The CO_2 refrigeration with heat recovery solution has been subject to few modifications aiming for a more energy efficient and multi-functioning system. One of the modifications in the system is to integrate air conditioning (AC) for warm season operation. The air conditioning integration is associated with parallel compression in the majority of the installed systems. According to authors communication with three major CO_2 refrigeration system manufacturers, there are about 10-15 CO_2 refrigeration systems with integrated heat recovery and air conditioning in Sweden. The more versatile becomes this integrated system solution, the more essential becomes understanding its characteristics for an energy efficient and reliable performance. This all-CO_2 and all-in-one multi-purpose energy system requires deeper investigations to make the most of its potential.

An all-in-one integrated CO_2 solution can be defined as an energy system which provides the entire or a significant share of cooling and heating demands in supermarkets. The only refrigerant used in this system is CO_2. The base function of this system is refrigeration, as the largest consumer of energy in supermarkets with a share of 35-60% in total energy use (Lundqvist, 2000) (NationalGrid, 2009) (Kauffeld, 2007). Integration of heating and air conditioning into the refrigeration system converts this single function system to a multi-function all-in-one solution. This integrated system combines all the thermal demands in a "plug & play" unit. Heating and air conditioning combined account for 10-30% of the total energy use in supermarkets, their exact shares are dependent on the climate conditions, building standards and conventional energy systems (NationalGrid, 2009) (Arias, 2005).

The all-in-one integrated CO_2 energy system has several advantages. First, the feed energy is electricity and the working medium is natural refrigerant CO_2, GWP equals to 1. The feed energy and the working medium can be compared with combustion-based heating and cooling systems (fossil fuels or biomass combustion in boiler, absorption chiller or district heating system), low-efficient electrical heaters, and high-GWP refrigerant based heat pump and air conditioning systems. The most conventional refrigerants in heat pump-air conditioning systems in Europe are R410A (GWP=2088), R407C (GWP=1784) and R134a (GWP=1430) (Nowak et al., 2014) (De Larminat and Wang, 2017). Using a natural-refrigerant based energy system can be considered as a long-term and sustainable solution avoiding the ever-restricting environmental regulations. Secondly, the integrated CO_2 system is a compact solution which saves space in the supermarket; few extra components are required to convert the single function CO_2 refrigeration system to a multi-function energy system. A separate heating and/or air conditioning unit requires all the stand-alone system components and extra space for installation. The third advantage is reducing the complexity of communications between various operation and maintenance entities responsible for running HVAC&R systems. For example, it is reported that the separation of responsibilities and independent control of refrigeration and HVAC systems in Swedish supermarkets resulted in significant decrease in the amount of heat recovered from the refrigeration systems (Arias and Lundqvist, 2006).

There are two challenges needed to be considered and addressed to witness the wide usage of this integrated CO_2 system solution. First, CO_2 components and systems have rather higher prices than conventional heat pump-AC components and systems. The compactness and the economies of scale of the CO_2 systems, on one side, and the increase in the operation and service costs of running systems with high GWP refrigerants, on the other side, are the factors reducing the investment and operation costs between the two systems. The second challenge is on the energy efficiency of the system. It should be shown in theory and in practice that the integrated CO_2 systems have higher or comparable energy efficiency, comparing with other refrigeration, heating, and AC stand-alone solutions. This is why the research like the one presented in this paper is essential to confirm or reject the energy efficiency superiority of this integrated CO_2 all-in-one concept over the stand-alone systems in different climate conditions.

The following literature review has been categorized based on the type of research (computer modelling and/or field measurements analysis) and the integrated function into the refrigeration system (heating and/or air conditioning).

Heating integration-computer modeling: Majority of the research works done on CO_2 integrated systems is based on computer modelling of heating integration into the CO_2 refrigeration system. Reinholdt and Madsen (2010) used computer simulation to investigate the heat recovery strategies from CO_2 booster systems. In one of the cases studied in this paper, it was shown that total COP (the ratio of "cooling power plus the recovered heat" to consumed electricity) can be increased by 25% when running on optimal pressure for heat recovery. The authors concluded that heat recovery is an appealing choice to increase the total energy efficiency of the supermarket. Tambovtsev et al. (2011) developed a bin temperature analysis method to compare the heat recovery from CO_2 booster system and electric heating in supermarkets. The authors showed that by using gas cooler by-pass and optimal control strategy a significant saving in supplying the heating energy is achievable.

Sawalha (2013) used computer simulation to investigate the performance of a CO_2 trans-critical system with heat recovery. A multi-step control strategy is recommended to maximize the efficiency of the CO_2 refrigeration system when recovering heat. Following the recommended control strategy, the study showed that the CO_2 trans-critical booster with heat recovery has 8% lower annual energy use in an average size Swedish supermarket compared to a conventional

R404A refrigeration system with a separate heat pump for heating needs. Using the suggested control strategy, Karampour and Sawalha (2014) showed that heat recovery from CO_2 trans-critical booster system has high seasonal performance factor (SPF) values of about 4, an SPF comparable to the majority of the commercial heat pumps available in the market. Polzot et al. (2017) compared the annual energy use of a CO_2 trans-critical booster system with heat recovery and a base system for three Northern Italy climate conditions. The base system consisted of an R134a/CO_2 cascade refrigeration and an R410A heat pump system. The authors concluded that in two of the three climate conditions the CO_2 integrated system saves 3.6% and 6.5% of the annual electricity use. Nöding et al. (2016) studied the effect of decoupling a CO_2 system heat recovery and heating demand by using hot water thermal storage tanks. Various high pressure side control strategies have been studied and the authors concluded that in the most realistic scenario, an optimum control strategy, using the tank storage, can reduce the daily electricity use by 8.5% comparing to a reference CO_2 system without a storage tank.

Heating integration-field measurements analysis: Of the few works based on field measurements analysis, Tambovtsev et al. (2011) examined a proposed heat recovery strategy focusing on gas cooler by-pass in a German supermarket. It is indicated that using the gas cooler by-pass increases the total COP of the CO_2 system by 20%. Rehault and Kalz (2012) analyzed the heat recovery from CO_2 refrigeration system in another supermarket in Germany. The CO_2 system uses a parallel compressor connected to ground storage as the auxiliary heater, the primary heat source in the building is heat recovery from the CO_2 refrigeration systems. The study showed that up to 50% of the rejected heat is recovered in the de-superheater in the cold months. However, the detailed analysis of primary and auxiliary heating functions is missing in the study. Funder-Kristensen et al. (2013) presented a case study of a Danish supermarket replacing the gas heating system with heat recovery from a CO_2 trans-critical booster system. It is shown that the CO_2 system can provide the entire heating demands of the supermarket with 20% saving in running cost comparing with the gas heating system. The economic calculation was based on 0.14 €.kWh^{-1} for electricity and 1.4 €.m^{-3} for the fuel gas. Heat recovery from a Swedish supermarket CO_2 refrigeration system was studied by Abdi et al. (2014). The theoretical results show that the amount of heat that can be recovered from the refrigeration system is about 130% of the cooling demand in the system. However, the analysis of the field measurements shows that only 30-70% of the available heat for recovery is utilized, the rest is released outdoors where the district heating, in this case, is used in parallel to cover the heating demand. Comparing heating COP to COP_1 (heating COP) of a typical ground source heat pump (COP_1 between 3 and 4.5) shows that the refrigeration system can recover heat at rather high efficiency.

Air conditioning integration-computer modelling: Air conditioning (AC) integration has been another subject of research gaining attention in the past few years; the research work has also been done mainly using computer simulations. AC integration research is usually associated with some modified CO_2 system solutions to increase its energy efficiency in warmer climates. Two major system modifications are parallel compression and ejector technology. This modified integrated concept has been presented by different authors including Hafner and Banasiak (2016) and Minetto et al. (2014). Studying this concept, Gullo et al. (2017) presented and compared the performance of an AC integrated CO_2 trans-critical booster system equipped with multi-ejectors and parallel compression with a conventional DX R404A refrigeration system. As part of this research, the performance of the “air conditioning” function of the integrated CO_2 system is compared with a separated HFC-based system using R410A for three cities and three scenarios of small, medium and large AC demands. The authors concluded that in all the scenarios and the cities, the AC energy efficiency of the CO_2 integrated system is 15.6-26.2% higher than the separated HFC-based system.

Air conditioning integration-field measurements analysis: Hafner et al. (2016) presented a heating and AC integrated CO_2 system installed in an Italian supermarket. The system is equipped with a multi-ejector unit to recover part of the expansion work. Analysing the summer performance of the refrigeration and AC system, the authors concluded that using a multi-ejector unit results in 15-30% energy efficiency comparing with running the integrated system without ejectors.

Heating and air conditioning integration – computer modelling: Of the few research works modelled and studied the integration of all different cooling and heating systems in supermarkets, Cecchinato et al. (2010) used a seasonal dynamic building energy simulation to compare the performance of various stand-alone and integrated supermarket refrigeration and air conditioning systems using HFC and CO_2 solutions. The baseline system includes stand-alone R404A direct expansion medium temperature and low temperature refrigeration and R410A air conditioning system (i.e reversible heat pump/chiller). Two of the five alternative integrated solutions studied in the paper use CO_2 in the refrigeration at low temperature level (i.e. freezers) in a cascade arrangement. It is shown that while the annual energy use in the low temperature level of these two systems is reduced, the energy use is increased in the medium temperature level due to the use of cascade the configuration. Extending the previous study, the same baseline HFC system was compared to only natural-refrigerant integrated solutions by Cecchinato et al. (2012). The alternative HVAC systems use NH_3 with auxiliary condenser boiler or R290 while refrigeration systems use cascade NH_3-CO_2, cascade R290-CO_2 or only-CO_2 booster system solutions. The study concluded that annual energy use can be decreased by more than 15% comparing the integrated HVAC & cascade systems with the baseline stand-alone HVAC&R system. Furthermore, the study concluded that using CO_2 booster system integrated solutions resulted in equal or higher energy use than the reference R404A system. However, the layout of this system and the heat rejection/ heat recovery from it is to some extent different from what is known as standard booster and integrated systems nowadays.

In another computer modelling work, Karampour and Sawalha (2015) compared the air conditioning performance of an integrated CO_2 booster system with a conventional R410A air conditioning system and concluded that the AC function of the CO_2 system is more energy efficient than R410A system for ambient temperatures lower than about 25°C; therefore, it is a suitable solution for Central-Northern Europe climates. It has been shown that using parallel compression during the AC delivery will result in 15-20% higher total COP compared to AC delivery without parallel compression in warm ambient temperatures. The usage of parallel compression has been also investigated in the heat recovery mode in the winter period. It has been concluded that usage of parallel compression is an efficient solution (about 5% energy saving), but only for very low ambient temperatures of less than about -15°C. The calculated energy savings due to parallel compression usage are certainly dependent on the load ratios, the receiver pressure and parallel compressor size assumptions. On the optimization of parallel compression, Javerschek et al. (2016) compared the seasonal energy efficiency ratio (SEER) of a full integrated CO_2 system with constant and variable receiver pressures. The authors concluded that using a variable optimized receiver pressure results in about 4% energy saving for a warm climate city represented by Athens. Girotto (2016) presented a CO_2 integrated system which provides direct space heating and cooling for a small supermarket in Italy. The design concept entails using CO_2 directly in the heating and AC systems, instead of using secondary fluids or water. The outcomes of the modelling suggested that this integrated solution will result in about 15-20% energy saving and considerable cost reduction, comparing with the conventional integrated CO_2 system.

What is less discussed in depth in the literature and distinguishes this research work from the previous ones is a comprehensive analysis of integrated CO_2 systems based on real field measurements. This paper evaluates the key operating parameters and energy efficiency of an integrated CO_2 trans-critical booster system installed in a small-medium size Swedish supermarket. A main objective is to study the system operation and energy efficiency while providing refrigeration, heating and/or air conditioning simultaneously. To have a comparative analysis of the performance of this integrated CO_2 concept, a comparative modelling analysis of the annual energy use of integrated CO_2, stand-alone CO_2, and HFC based systems is presented in the paper.

2. SYSTEM DESCRIPTION

The integrated CO_2 trans-critical booster system is installed in a supermarket located in the center of Sweden. The supermarket has been in operation since September 2013. A simple schematic of the integrated CO_2 system is shown in Figure 1. The system provides the refrigeration, air conditioning, and a share of the heating demands. Heat recovery from the CO_2 system is the primary heating system and district heating is used as the auxiliary heating source.

The system can be explained following the numbering indicated in Figure 1. The system is composed of three compressor units: low temperature (LT) (1), medium temperature (MT) (2) and parallel (3) compressors. The hot gas discharge from the MT and parallel compressors mix and pass through two heat exchangers called de-superheaters via which heat is recovered. The first de-supeheater (4) provides tap water heating with a forward temperature of about 55-60°C. The second de-superheater (5) is for space heating providing the hydronic system of the space heating network with a forward temperature of 35-45°C, depending on the building's heating demand. This heat is used mainly in floor heating and radiators.

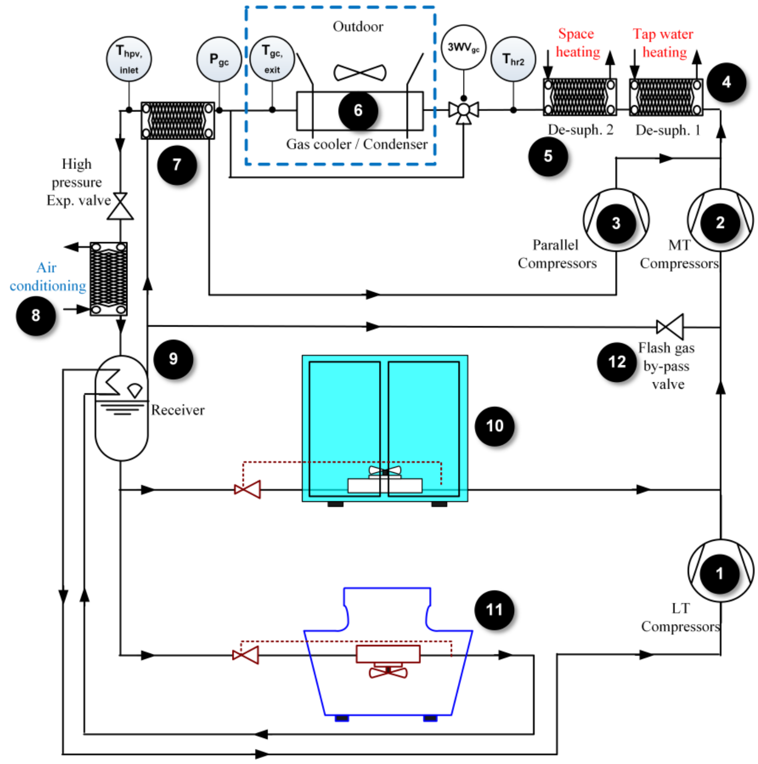


Figure 1: Schematic of the CO_2 integrated system

When the heat to be rejected from the system is higher than the heating demands, a gas cooler (6) on the roof of the supermarket is used to reject the excess heat to the ambient. A heat recovery control strategy is applied to regulate the gas cooler pressure and exit temperature of refrigerant from gas cooler. In this control strategy the gas cooler maybe by-passed using the 3-way valve; so all heat from the refrigeration system is recovered in the de-superheaters, usually when the ambient temperatures are very low. The gas cooler pressure P_{gc} follows an algorithm for the maximum COP in summer trans-critical mode. This mainly occurs when T_{amb} is higher than 25°C. Some parameters on the high pressure side are indicated in the schematic to be discussed further in section 4.1.1. These include gas cooler pressure P_{gc} [bar], gas cooler exit temperature $T_{gc,exit}$ [°C], high pressure regulating valve inlet temperature $T_{hpv,inlet}$ [°C] and gas cooler by-pass valve $3WV_{gc}$.

CO_2 exiting the gas cooler flows in a sub-cooler (7) and is cooled by a cold vapour stream; suction line heat exchanger at the parallel compressors side. However, the sub-cooling effect has been found to be very insignificant. The sub-cooled liquid is expanded by a high pressure regulating valve and enters an air conditioning heat exchanger (8). This evaporator is connected to the air conditioning system with forward and return temperatures of about 7 and 12°C. CO_2 mass flow rate is too high in relation to AC demand in this heat exchanger, so complete evaporation is never achieved.

The two-phase CO_2 fluid leaving the air conditioning evaporator enters a receiver (9) where vapour and liquid are separated. The liquid stream is fed to the medium temperature cabinets (10) and low temperature freezers (11). The vapour in the receiver has two paths; the first path is expanding into the suction line of MT compressors through the by-pass valve (12), known as “flash gas by-pass” and it is the most commonly applied, the second path is direct to the parallel compressors (3) suction line. The refrigerant can go in either one of the paths or both paths simultaneously. The parallel compression path is more efficient because the parallel compressors suction pressure is higher than MT compressors suction pressure.

Managing the receiver vapour and regulating the receiver pressure are important parameters for the operation of the system since the AC forward temperature is affected by receiver pressure. Another ad-hoc solution used in this system is to condense part of the vapour in the receiver by a cooling coil from the LT return gas of the freezers. However, this cooling effect has been estimated to be small.

3. DATA COLLECTION AND EVALUATION METHOD

The real-time field measurements of the supermarket have been accessed via the web-monitoring interface “IWMAC” (IWMAC, 2015). The measurements are updated every 5-10 seconds but the data are synchronized and averaged to 15-min. intervals. The data used in this analysis are captured and processed for the months of January 2014 and July 2014, representing cold and warm months. These two months have been selected because the weather was rather mild in 2015 comparing to 2014 and one of the objectives in this study has been to investigate the performance of the system in providing heating and air conditioning under representative cold and warm climate conditions in Sweden.

A computer model in EES (Engineering Equation Solver) software (Klein, 2015) is used to evaluate the measured key operating parameters of the integrated system in order to find out how good the system performs while providing various cooling and heating demands. Using the field measurements, the performance of the integrated CO_2 booster system is analysed in the EES computer model. This includes cooling and heating loads and COPs of refrigeration, heat recovery and AC. The main steps in the calculations have been described in the following sub-sections.

3.1 Compressors electricity use, refrigeration, heating and air conditioning loads

Mass flow meters are rarely installed in supermarket refrigeration systems, so an indirect method is used to find the mass flow rate passing through the LT compressors, $\dot{m}_{LT}$ [kg s^{-1}], MT compressors, $\dot{m}_{MT}$, and parallel compressors, $\dot{m}_{PC}$. The method applied in this study is to find the compressors mass flow rates based on volumetric efficiencies and swept volume flow rates of compressors obtained from the manufacturer data, the method is explained by Sawalha et al. (2017).

The cooling and heating loads in the system; $\dot{Q}_{LT}$, $\dot{Q}_{MT}$, $\dot{Q}_{AC}$, $\dot{Q}_{TWH}$, $\dot{Q}_{SPH}$, and the heating power rejected in the gas cooler $\dot{Q}_{gc}$ [kW] are calculated using equation (1):

$$\dot{Q} = \dot{m}_{ref} * \Delta h_{heat\ exchanger} \qquad (1)$$

Where $\dot{m}_{ref}$ [kg s^{-1}] is the mass flow of refrigerant in the heat exchanger and $\Delta h_{heat\ exchanger}$ [kJ kg^{-1}] is the enthalpy difference across the heat exchanger. The enthalpy at the exit of LT and MT evaporators/cabinets [kJ kg^{-1}] are calculated using 11 K internal super-heating in LT level and 12 K in the MT level, the values have been extracted from field measurements analysis of the MT and LT cabinets. The enthalpy at LT and MT cabinets’ inlets corresponds to saturated liquid at the receiver pressure. Regarding the AC evaporator, the inlet enthalpy is equal to the high pressure regulating valve inlet, while the AC evaporator exit enthalpy is obtained using the mass and energy balance of the receiver.

During the analysis it has been observed that the electric power measurements of some MT compressors are not reliable; much lower values than expected have been recorded. For example, values between 7-15 kW were recorded for the entire year while about 15-25 kW were expected for T_{amb} higher than about 20-25°C, according to the compressors manufacturer data and with the same number of compressors running. Alternatively, it has been assumed that those compressors are performing according to the manufacturer specifications; hence, the power consumption of each compressor was estimated, according to equation (2), by using the total efficiency of the compressor extracted from the manufacturer data. This assumption has been evaluated on the LT compressors that have measured values close to expectations, i.e. direct power measurements and the power estimation based on "total efficiency" method have been compared for a set of 150 measurement points, and good accuracy of the total efficiency method has been confirmed; the deviations between the measured and estimated values were less than 10% for 80-85% of the points.

$$\dot{E} = \dot{m}_{ref} * \Delta h_{is} / \eta_{tot} \quad (2)$$

where η_{tot} [-] is the total efficiency of each compressor and Δh_{is} [kJ kg-1] is the isentropic enthalpy difference over each compressor unit.
It is worth mentioning that one compressor in each MT, LT or PC unit runs with variable speed and the other compressors are on/off controlled. There is a capacity-speed% measurement of each single compressor available and it is included in the mass flow rate and power calculations.

3.2 Coefficients of performances (COPs)
Having calculated refrigeration, heating and AC loads and measured/estimated LT, MT and parallel compressors power consumptions, different COPs of the integrated system can be calculated to evaluate its performance.

Total system COP [-] is defined as the ratio of the total provided demands to the total used electricity in the system, calculated using equation (3):

$$COP_{tot} = (\dot{Q}_{MT} + \dot{Q}_{LT} + \dot{Q}_{AC} + \dot{Q}_{SPH} + \dot{Q}_{TWH}) / (\dot{E}_{MT} + \dot{E}_{LT} + \dot{E}_{PC} + \dot{E}_{fan}) \quad (3)$$

where $\dot{E}_{fan}$ [kW] is the electricity consumption of gas cooler fans and the other parameters have been explained in section 3.1. $\dot{E}_{fan}$ is estimated to be 3% of the heat rejected in the gas cooler $\dot{Q}_{gc}$, according to communication with a major CO_2 gas cooler manufacturer.

The total refrigeration COP, referred to as COP_{ref}, is defined as the ratio of total provided refrigeration to the total electricity used only for the refrigeration function:

$$COP_{ref} = (\dot{Q}_{MT} + \dot{Q}_{LT}) / [(\dot{E}_{MT} + \dot{E}_{LT} + \dot{E}_{PC} + \dot{E}_{fan})_{only-refrigeration}] \quad (4)$$

The calculation of COP_{ref} [-] for summer operation includes the power consumption only for refrigeration function; i.e. the estimated power consumed to provide AC ($\dot{E}_{AC}$) is extracted from the total. For the winter operation, the system is run usually at 80-85 [bar] discharge pressure to recover heat. In order to eliminate the influence of heat recovery, the system was assumed to run at 45 [bar] (about 10°C), which is the minimum floating condensing pressure/temperature, and gas cooler exit temperature was assumed to be equal to T_{amb} + 5K (and not lower than 5°C), this is the expected running conditions when the outdoor temperature is lower than 0°C. Since the max T_{amb} in this month is about 4°C, the minimum condensing pressure kept the same for this narrow region of higher than 0°C.
Using this minimum floating condensing pressure, the power used for only-refrigeration function in winter operation is estimated.

To define the COPs for LT, MT and AC performances, the shares of their electric power demand in the total electricity use should be found. Two cases are considered to find these shares:

a) Without parallel compression: the CO_2 system can be considered as a simple refrigeration/heat pump unit with evaporation temperature equal to MT, as shown in Figure 2. In this case, the compressor suction pressure providing $\dot{Q}_{AC}$ load is corresponding to about -8°C, similar to MT evaporation temperature. The total load on the cold side $\dot{Q}_{cooling}$ [kW] is:

$$\dot{Q}_{cooling} = (\dot{Q}_{MT} + \dot{Q}_{MT,xSH}) + \dot{Q}_{AC} + (\dot{Q}_{LT} + \dot{Q}_{LT,xSH} + \dot{E}_{LT\_shaft}) \quad (5)$$

where $\dot{E}_{LT\_shaft}$ [kW] is the shaft power of the LT compressors. $\dot{Q}_{LT}$ plus $\dot{E}_{LT\_shaft}$ is the condensation load from the LT loop added to the high stage compressors. $\dot{Q}_{MT,xSH}$ and $\dot{Q}_{LT,xSH}$ [kW] are non-useful cooling (external super-heating) in the suction line of two compressor units. MT and LT external super-heating is calculated based on the temperature difference between compressors suction lines and evaporators outlets. The reason to include this non-useful cooling (external super-heating) in equation (5) is that $\dot{Q}_{cooling}$ represents "compressors cooling capacity", while equation (1) includes only useful "evaporators cooling capacity" for calculating $\dot{Q}_{MT}$, $\dot{Q}_{LT}$ and $\dot{Q}_{AC}$ and equation (6) includes only this useful "evaporators cooling capacity" to calculate COP_{ref}. In other words, the "compressors cooling capacity" is the sum of useful "evaporators cooling capacity" and non-useful heat gains between the evaporators outlet and compressors suction lines, known as external super-heating.

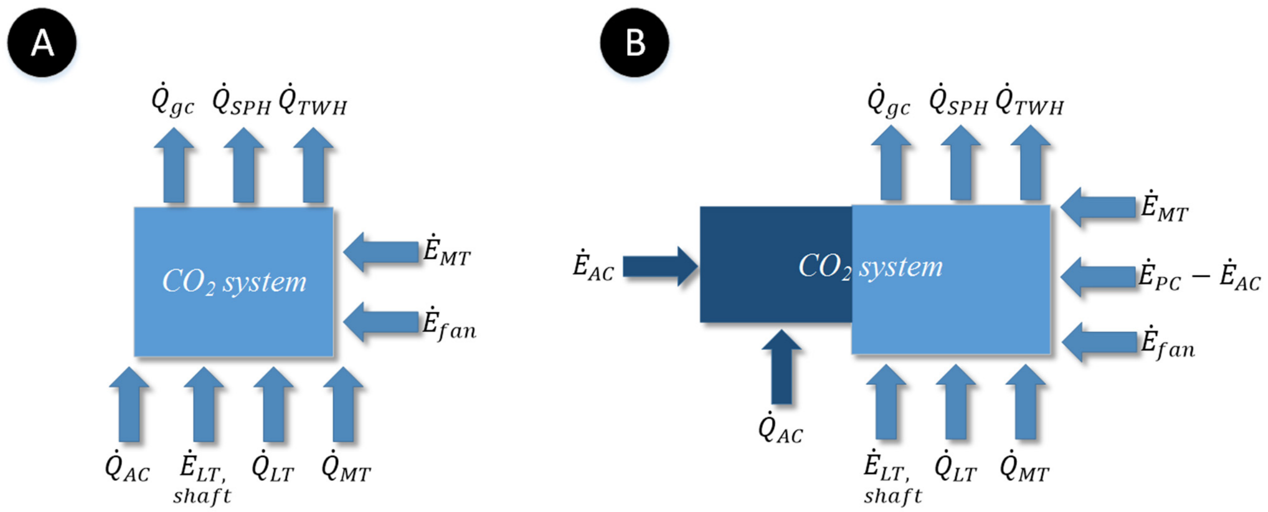


Figure 2: CO_2 system conceptual schematic (A) without parallel compression (B) with parallel compression

Using $\dot{Q}_{cooling}$ [kW] and the shares of each cooling load on MT compressors, *COP*s of the cooling side can be defined as:

$$COP_{cooling} = \dot{Q}_{cooling} \,/\, (\dot{E}_{MT} + \dot{E}_{fan}) \quad (7)$$

$$COP_{MT} = \dot{Q}_{MT} \,/\, [(\dot{Q}_{MT} + \dot{Q}_{MT,xSH}) \,/\, COP_{cooling}] \quad (8)$$

$$COP_{AC}(without\ PC) = \dot{Q}_{AC} \,/\, [\dot{Q}_{AC} \,/\, COP_{cooling}] \quad (9)$$

$$COP_{LT} = \dot{Q}_{LT} \,/\, [\dot{E}_{LT} + (\dot{Q}_{LT} + \dot{Q}_{LT,xSH} + \dot{E}_{LT\_shaft}) \,/\, COP_{cooling}] \quad (10)$$

COP_{MT} [-] and COP_{LT} [-] are the refrigeration COP at MT and LT respectively. For winter, COP_{MT} and COP_{LT} are calculated based on only-refrigeration function, as done for COP_{ref} and explained in equation (4).
Equation (11) calculates the total cooling COP; total compressors cooling capacity $\dot{Q}_{cooling}$ divided by the electricity use $\dot{E}_{MT} + \dot{E}_{fan}$. Equations (12), (13) and (14) calculate separated cooling COPs; each useful cooling capacity is divided by the share of its electricity use in the total electricity use.

b) With parallel compression: when parallel compression is used, $\dot{Q}_{AC}$ is provided at an evaporation level of about 5-6°C, as shown as a higher cold side pressure than MT level in Figure 2-B. The capacity of the parallel compressors is sufficient to ensure that the entire AC load is provided by parallel compressors. The compressor power which is used for providing AC load ($\dot{E}_{AC}$) is a fraction of parallel compressors total power use ($\dot{E}_{PC}$); the vapour flowing through the parallel compressors is a combination of the vapour flashing after the high pressure expansion valve and the vapour generated in the AC evaporator. $\dot{E}_{AC}$ can be found by comparing the ratio of parallel compressors mass flow rate and AC load required mass flow rate.
COP_{AC} in this case is calculated using the following equation:

$$COP_{AC}(with\, PC) \; = \; \dot{Q}_{AC} \; / \; \dot{E}_{AC} \tag{15}$$

The difference between $\dot{E}_{PC}$ and $\dot{E}_{AC}$ ($\dot{E}_{PC} - \dot{E}_{AC}$) is the electricity which is used for refrigeration function of the cycle; i.e. compression of the vapour flashing after high pressure expansion valve. The difference between $\dot{E}_{PC}$ and $\dot{E}_{AC}$ is added to the other power consumptions for refrigeration purpose ($\dot{E}_{MT} + \dot{E}_{LT} + \dot{E}_{fan}$) and included in the calculations of COP_{ref}, $COP_{cooling}$, COP_{MT} and COP_{LT}.

COP for heat recovery (COP_{HR} [-]) is defined as the total recovered heat over the extra electricity used only for heat recovery, as in the following equation:

$$COP_{HR} \; = \; (\dot{Q}_{SPH} \; + \; \dot{Q}_{TWH}) \; / \, [(\dot{E}_{MT} + \; \dot{E}_{LT} \; + \; \dot{E}_{PC} \; + \; \dot{E}_{fan})_{tot} \; - \; (\dot{E}_{MT} + \; \dot{E}_{LT} \; + \; \dot{E}_{PC} \; + \; \dot{E}_{fan})_{only-refrigeration}] \tag{16}$$

However, it has been observed that parallel compression is very rarely activated in winter conditions in this supermarket and it doesn't influence COP_{HR} calculations.

4. RESULTS ANALYSIS AND DISCUSSION

4.1 Evaluation of Key Operating Parameters

In order to establish a good idea on how the system performs in providing the different refrigeration, heating and AC loads, the key operating parameters of the system are evaluated in this section.

4.1.1 High pressure side analysis

The high pressure side of the cycle is analysed in order to evaluate how the system is controlled in summer floating condensing and winter heat recovery modes. The studied parameters include ambient temperature T_{amb} [°C], gas cooler pressure P_{gc} [bar], gas cooler exit temperature $T_{gc,exit}$ [°C], high pressure regulating valve inlet temperature $T_{hpv,inlet}$ [°C] and gas cooler by-pass valve $3WV_{gc}$. The last four parameters have been shown on the high pressure side of the system sketch in Figure 1. The summer observations show that $T_{gc,exit}$ [°C] and $T_{hpv,inlet}$ [°C] follow the ambient temperature T_{amb} closely. The reason for the small difference between $T_{gc,exit}$ [°C] and $T_{hpv,inlet}$ [°C] could be due to the under-sized capacity of the sub-cooler heat exchanger. The gas cooler approach temperature is 2-4K for sub-critical and 1-2K for trans-critical pressures. The system usually runs in the sub-critical floating condensing mode when T_{amb} < 25°C (80% of the time in July) and it follows an algorithm to control the discharge pressure to maximize the COP when the system runs in the trans-critical region.

Figure 3 shows the high pressure side key operating parameters for the January 2014. As explained earlier, the system provides space heating to the supermarket. To cover this demand, some key parameters on the high pressure side are regulated by the heat recovery control system. The recommendations on how to control the high pressure side key parameters for efficient heat recovery have been discussed in detail in some other research works (Sawalha, 2013) and (Madsen and Bjerg, 2016). In brief, the recommendation consists of a stepwise control strategy; the first step is to regulate the gas cooler pressure P_{gc} and keep the gas cooler exit temperature $T_{gc,exit}$ as low as possible by running the gas cooler at full capacity. The second step is to fix the gas cooler pressure at a maximum value P_{gc} and decrease the gas cooler capacity by reducing the fans speed or, ultimately, by-passing the gas cooler using by-pass valve $3WV_{gc}$.

The general trend of controlling the system agrees with the recommended strategy. Gas cooler pressure P_{gc} is fixed to 80-85 [bar], mainly when the ambient temperature T_{amb} is lower than 0°C. On the right Y-axis the opening or closing condition of gas cooler by-pass valve $3WV_{gc}$ has been shown. 1 means that the gas cooler is by-passed and 0 means that the refrigerant passes through the gas cooler to reject part of the heat to the ambient. This Opening-closing frequency has been averaged by time-weight and this explains the numbers between 0 and 1. In reality, the valve can be either fully closed ($3WV_{gc}$ = 0) or fully opened ($3WV_{gc}$ = 1, i.e. gas cooler by-passed).

As can be observed in the figure, the gas cooler is by-passed ($3WV_{gc}$ =1) for a long period in the middle of January in order to have full heat recovery from the refrigeration system, during this period $T_{hpv,inlet}$ is rather constant to a value of about 30°C, close to the 2nd de-superheater exit temperature T_{hr2} [°C]. For the other two periods in the beginning and end of January, $3WV_{gc}$ fluctuates between 0 and 1, this means that some heat is rejected in the gas cooler and this explains the fluctuations of $T_{hpv,inlet}$ between $T_{gc,exit}$ and T_{hr2}. During the periods when gas cooler is running, i.e. $3WV_{gc}$=0, the lowest reached gas cooler exit temperature $T_{gc,exit}$ is about 7°C. During the periods when gas cooler is by-passed, i.e. $3WV_{gc}$ =1, $T_{gc,exit}$ follows the ambient temperature since this sensor is installed on a pipe connection outdoors; therefore, it has no meaning to system performance during such conditions.

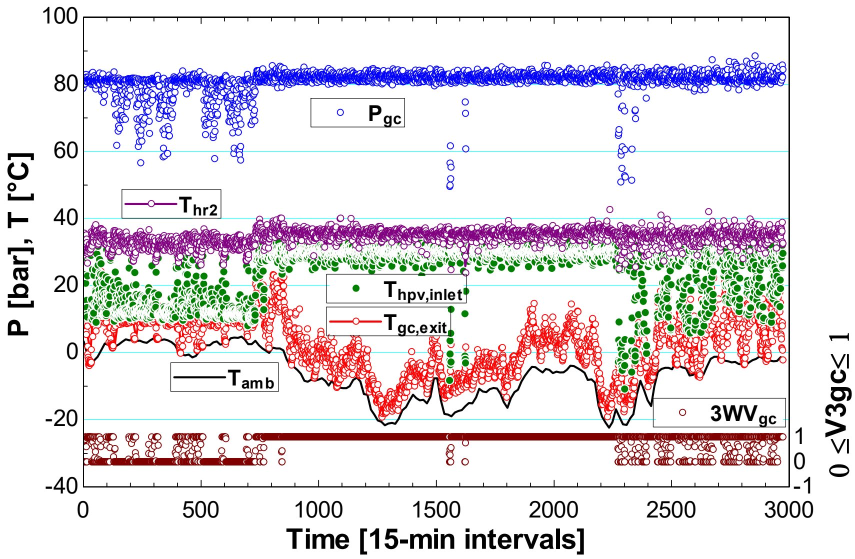


Figure 3: Key high pressure side operating parameters in January 2014

4.1.2 Delivered temperatures for different functions of the systems

The delivered temperatures for refrigeration, heating, and AC by the CO_2 integrated system are studied to make sure that the temperature requirements are properly fulfilled. The results are summarized in Table 1. What can be concluded from this "key operating parameters" evaluation is that the supplied temperatures seem reasonable for refrigeration, heating, and AC. A minor exception to this statement is for the space heating forward temperature; the CO_2 system heating forward temperature for space heating in extreme cold winter temperatures of lower than -15°C is not sufficient and district heating should be used as auxiliary. This can be seen in the table comparing space heating "water forward temperature" delivered by the CO_2 system (40-45°C) and 44-50°C demand for ambient temperatures in the range of -15°C to -26°C.

Table 1: Key operating parameters – functions, deliveries and demands

Function	Key Operating Parameter	Winter	Summer	Notes on Demand
Refrigeration at MT level	CO_2 evaporation temperature	$-8 < T_{MT} < -5$°C* Average = -6.4°C**	$-8.5 < T_{MT} < -7.3$°C* Average = -7.9°C**	Lower evaporation temperature is needed during summer
Refrigeration at LT level	CO_2 evaporation temperature	$-34 < T_{LT} < -27$°C* Average = -31.2°C**	$-35.2 < T_{LT} < -30.7$°C* Average = -33**	Lower evaporation temperature is needed during summer
Tap water heating	Supplied tap water temperature	55-60°C	55-60°C	Constant annual demand
Space heating	Water forward temperature	35- 40°C for $T_{amb} > 0$°C 40- 45°C for $T_{amb} < 0$°C	No space heating demand	Forward temperature demand curve governing the heating system (CO_2 heat recovery + district heating) as a function of T_{amb}: 36°C @ 0°C T_{amb} 44°C @ -15°C T_{amb} 50°C @ -26°C T_{amb}
Air conditioning	CO_2 evaporation (receiver saturation) temperature	No air conditioning demand	4-7°C	High demand period when T_{amb} >15°C
	Secondary fluid forward and return temperature		Forward: 7-8°C Return: 12-13°C	

*The range is truncated discarding 10% of the highest and lowest values.
**Averaged for the entire month.

4.2 Energy Performance Analysis

In the following sub-sections, energy efficiency of the system including the provided cooling and heating loads and COPs are evaluated.

4.2.1 Cooling and heating loads

Cooling and heating loads of the integrated CO_2 system and ambient temperature (T_{amb} [°C]) are shown in Figure 4. Figure 4-A and C are hourly-averaged loads for January and July 2014 respectively and Figure 4-B and D are daily-averaged values for the same periods. The cooling and heating loads are shown as negative and positive values, respectively. The hourly- and daily-averaged loads are used to investigate the performance of the system in day-night and summer-winter periods. The loads trends are discussed individually in the following text. It is worth pointing out that Y-axis scales of summer-winter cases are different:

- $\dot{Q}_{LT}$ *(light-blue)*: the LT refrigeration load varies in a narrow range of 10-15 kW in both months, independent of hour-of-the-day and T_{amb}. The main reason for this steady performance is the glass lids on the freezers which significantly reduce the impacts from indoor humidity fluctuations throughout the year.

- $\dot{Q}_{MT}$ *(dark-blue)*: $\dot{Q}_{MT}$ is about 5-10 kW higher in July than January, comparing Figure 4-B (January) and Figure 4-D (July). During the cold period $\dot{Q}_{MT}$ is in the 20-25 kW narrow range according to Figure 4-B. Figure 4-D shows some increases during the warm days of July, 8-11th and 21-30th, but the variations are not so significant, comparing to conventional supermarkets. As an example, a previous study showed that $\dot{Q}_{MT}$ was about twice higher in warm summer days than cold winter ones (Freléchox, 2009). The reason for this rather stable behaviour is that many of the MT cabinets in modern supermarkets are also equipped with glass doors.
As shown in Figure 4-A and Figure 4-C, there is a clear increase of $\dot{Q}_{MT}$ in the working hours 8:00-22:00 mainly due to the presence of customers, relatively higher outdoor temperature, higher indoor temperatures, and higher indoor

humidity. This increase can't be seen in $\dot{Q}_{LT}$ values. A major cause for this difference could be that the cold air is let out easier while opening the door of vertical MT cabinets comparing with opening the glass lid of horizontal LT freezers.

- $\dot{Q}_{AC}$ *(purple)*: the values of AC load in July shows that the system doesn't provide any AC load when the ambient temperature is lower than 10-12°C. The reason for AC demand in relatively low ambient temperatures of 10-15°C could be that specific HVAC zones in the supermarket including bakery might have different demands. The clear relation between $\dot{Q}_{AC}$ and T_{amb} can be seen in both Figure 4-C and Figure 4-D. It can be seen that when T_{amb} increases, Q_{AC} increases. In addition to the "higher outdoor/indoor temperature" factor, presence of the customers in the supermarket during the opening hours increases CO_2 PPM in the air and more ventilation of the conditioned air might be required, specifically in the afternoon rush hours. This can be observed in Figure 4-C.

- $\dot{Q}_{TWH}$ *(red)*: this load doesn't change significantly throughout the year and is independent of T_{amb} as can be seen in all four figures. However, in the daily profile in Figure 4-A and Figure 4-C it can be observed that the early-morning food preparation and late-night cleaning in the supermarket resulted in peaks of $\dot{Q}_{TWH}$ up to 5 kW.

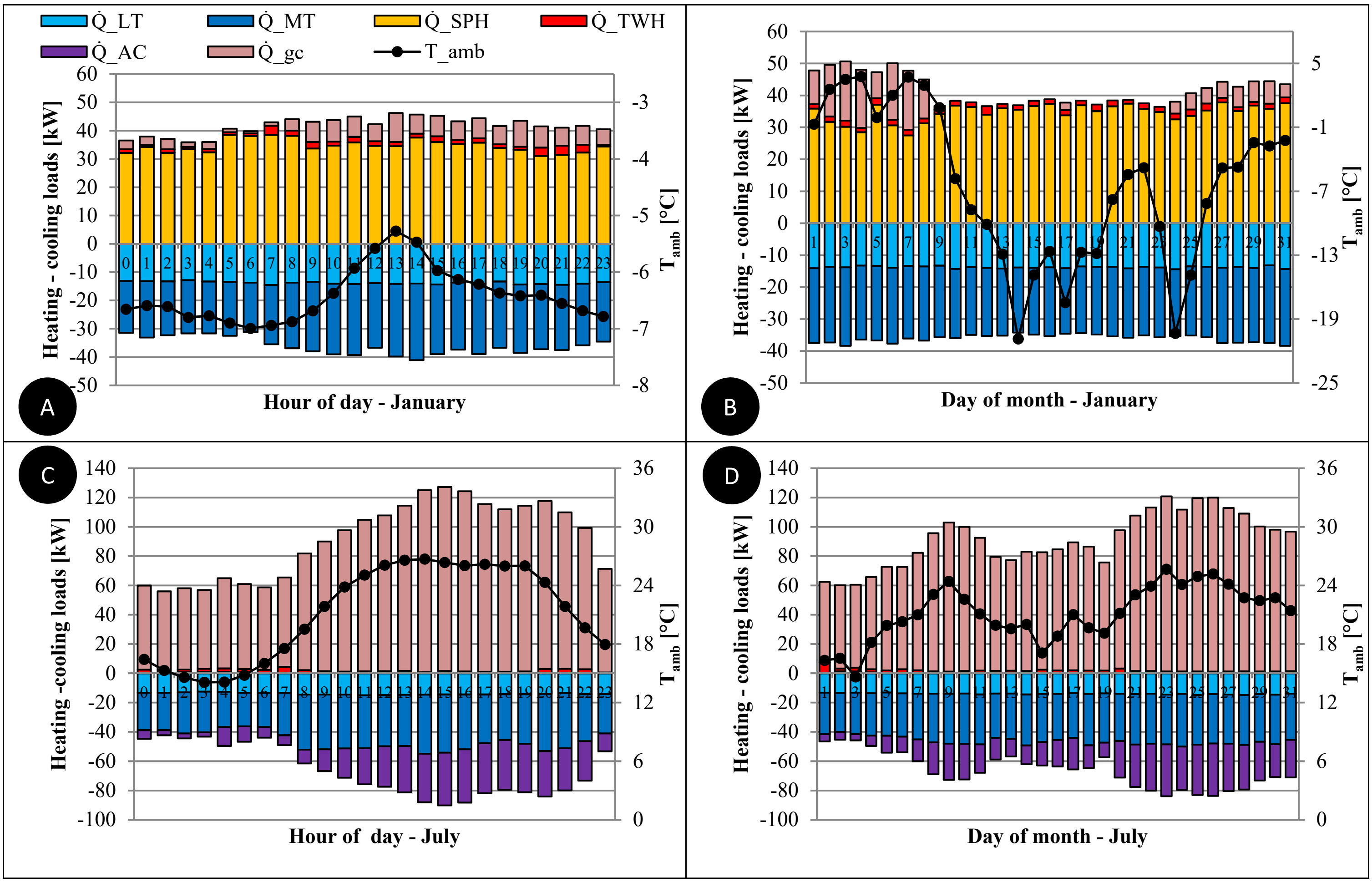


Figure 4: Hourly-averaged cooling and heating loads for (**A**) January 2014 (**C**) July 2014. Daily-averaged cooling and heating loads for (**B**) January 2014 (**D**) July 2014

- $\dot{Q}_{SPH}$ *(orange)*: Analysing the field measurements for longer period than presented in Figure 4, it is observed that the space heating load is zero for T_{amb} >15°C and it is very small for 10< T_{amb}<15°C. The hourly- and daily-averaged winter values of Q_{SPH} are within the 30-40 kW range and rather independent of T_{amb} when it is lower than 0°C. This indicates that district heating is used as supplementary heating source at sub-zero ambient temperatures.

It can be observed in both Figure 4-A and Figure 4-B that $\dot{Q}_{SPH}$ has the same order of magnitude as total refrigeration load, i.e "$\dot{Q}_{MT}$ + $\dot{Q}_{LT}$". This means that all the refrigeration loads provided by the CO_2 system is recovered for heating demands. The average heat recovery ratio HRR, the ratio of the recovered heat "$\dot{Q}_{SPH}$ + $\dot{Q}_{TWH}$" to cooling demand on high stage compressors [i.e. $\dot{Q}_{cooling}$ in equation (5)] is 85-105% at around mid-nights and 75-85% during opening hours of the supermarket. In some of the authors older studies, it has been shown that by using proper heat recovery control strategy, higher HRR (potentially up to 150%) can be expected from the CO_2 system with high heating COP (Sawalha, 2013) (Abdi et al., 2014).
It is worth mentioning that there are always some heat losses to the ambient on the high pressure side of the system. This is the reason that the heat recovery ratio HRR can't reach the highest potential values. The temperatures before and after the high pressure side heat exchangers are not exactly the same even if the three-way by-pass valves are operating. For example, this can be observed comparing the temperatures before and after the gas cooler (T_{hr2} and T_{hpv}) in Figure 3, when the gas cooler is fully by-passed in mid-January. The authors speculate that the valves are not completely tight to have the by-pass effect complete.

- $\dot{Q}_{gc}$ *(pink)*: while all the provided refrigeration and AC loads are rejected outdoors through the gas cooler during the summer (Figure 4-C and D), only a minor portion of these loads is rejected during winter (Figure 4-A and B). As can been seen in Figure 4-B, gas cooler is by-passed completely (i.e. zero $\dot{Q}_{gc}$) for very low ambient temperatures in the 9th to 23rd of January period (except for the 17th) to recover the entire heat available from the refrigeration system.

4.2.2 Coefficients of performances (COPs)

Figure 5 shows the 15-min. averaged total system COP (COP_{tot}), total refrigeration COP (COP_{ref}), medium and low temperature refrigeration COPs (COP_{MT} and COP_{LT}) for January and July 2014. The COPs have been discussed for winter and summer cases separately in the following text.

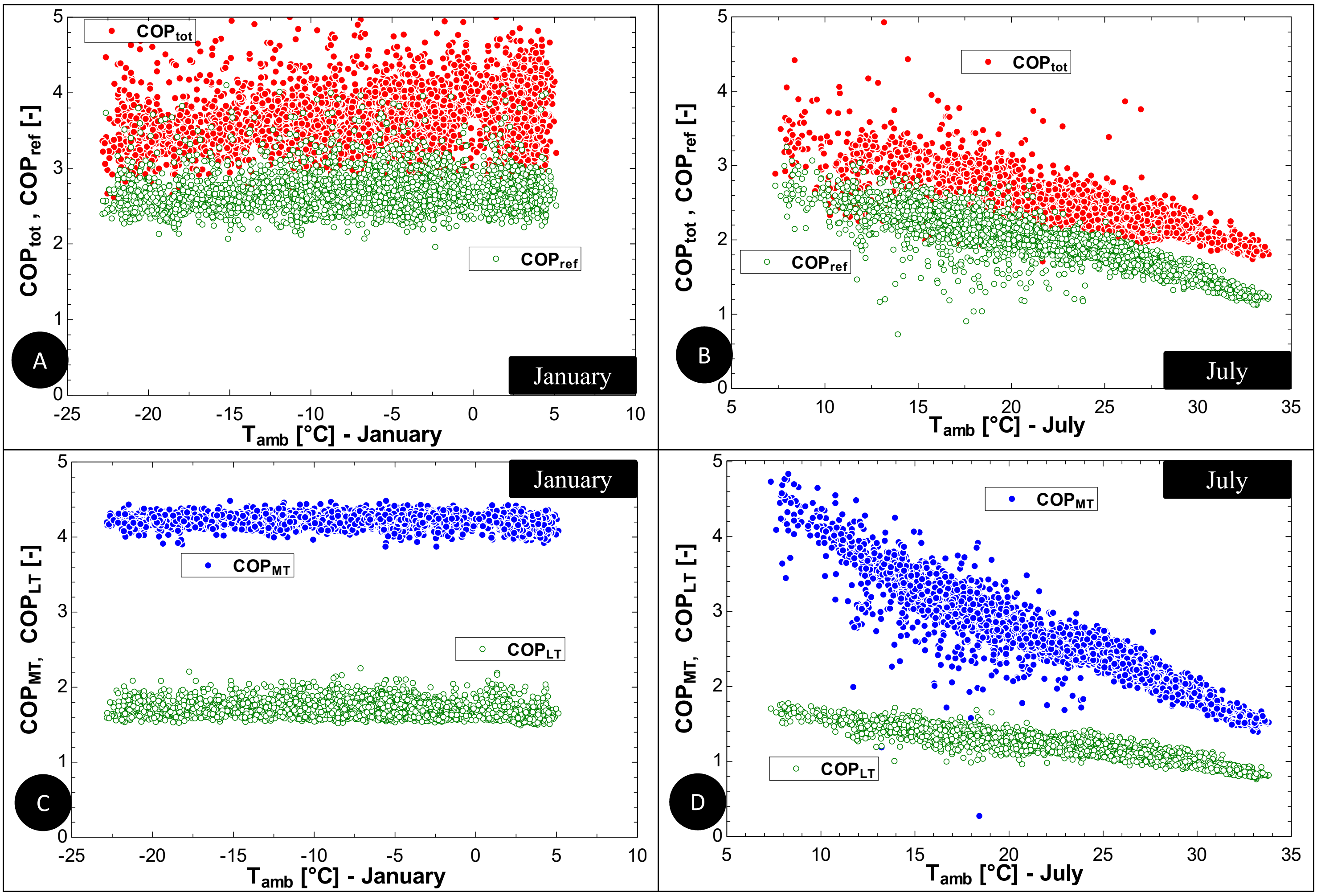


Figure 5: 15-min. averaged total system COP (COP_{tot}) and total refrigeration COP (COP_{ref}) for (**A**) January 2014 (**B**) July 2014, 15-min. averaged medium temperature COP (COP_{MT}) and low temperature COP (COP_{LT}) for (**C**) January 2014 (**D**) July 2014

- ***Winter case***: as shown in Figure 4, most of the loads including $\dot{Q}_{MT}$, $\dot{Q}_{LT}$, $\dot{Q}_{TWH}$ and $\dot{Q}_{SPH}$ have narrow range of variations in winter. Furthermore, the gas cooler pressure P_{gc} is fixed to 80-85 [bar] for ambient temperatures lower than 0°C. The minimum floating condensing pressure is also fixed to 45 [bar]. The exit temperature from the de-superheat for space heating (Thr_2 in Figure 1) is rather constant at about 35°C and the inlet temperature to the high pressure control valve ($T_{hpv,inlet}$) is about 30°C. The gas cooler is by-passed frequently in this period. All these narrow changes in loads and boundary conditions of the system lead to a horizontal trend in performance, as can be observed in the plots of COP_{tot}, COP_{ref}, COP_{MT} and COP_{LT} in Figure 5-A and Figure 5-C.

- ***Summer case***: when T_{amb} is lower than 15°C, the integrated CO_2 system provides refrigeration, heating and AC simultaneously. This explains the difference between COP_{tot} and COP_{ref} points at the left end in Figure 5-B. The reason for simultaneous heating and AC demands in T_{amb} region of 10-15°C could be the different demands and set points in

different HVAC zones of the supermarket. At T_{amb} lower than 15°C the system runs in floating condensing mode with low P_{gc} of 50-55 [bar], corresponding to 14-18°C condensation temperature. This region is where relatively high values for COP_{ref}, COP_{MT} and COP_{LT} are reached; occurring mainly during occasional cold July nights.
For ambient temperatures between 15-25°C, parallel compression is used frequently. When T_{amb} is higher than 25°C, mainly in summer mid-days, the system operates in trans-critical mode. Parallel compression is the only vapour management method used in this region. The reason for the clear difference between COP_{tot} and COP_{ref} in this region is that AC load is provided at higher evaporation temperature comparing -8°C without PC and 5°C with PC. This increases the total COP of the system, comparing with the refrigeration COP.

The refrigeration COPs (COP_{ref}, COP_{MT} and COP_{LT}) discussed here are in good agreement with authors previous study on "field measurements of CO_2 refrigeration systems" (Sawalha et al., 2015). For example, the average COP_{ref} of this system in July 2014 is 2 at average P_{gc} pressure of 67 [bar]. This matches well with COP_{ref} of the improved CO_2 systems studied in the other paper, considering that this system operates 20% of the time in trans-critical mode and the other systems rarely operate in trans-critical mode. As discussed in the other paper, these COPs are normalized for a load ratio ($\frac{\dot{Q}_{MT}}{\dot{Q}_{LT}}$) of 3. This implies that the efficiency of refrigeration was not compromised to provide the heating and AC demands.

Figure 6 shows 15-min. averaged heating COP_{HR} in the winter (left) and air conditioning COP_{AC} in the summer (right).

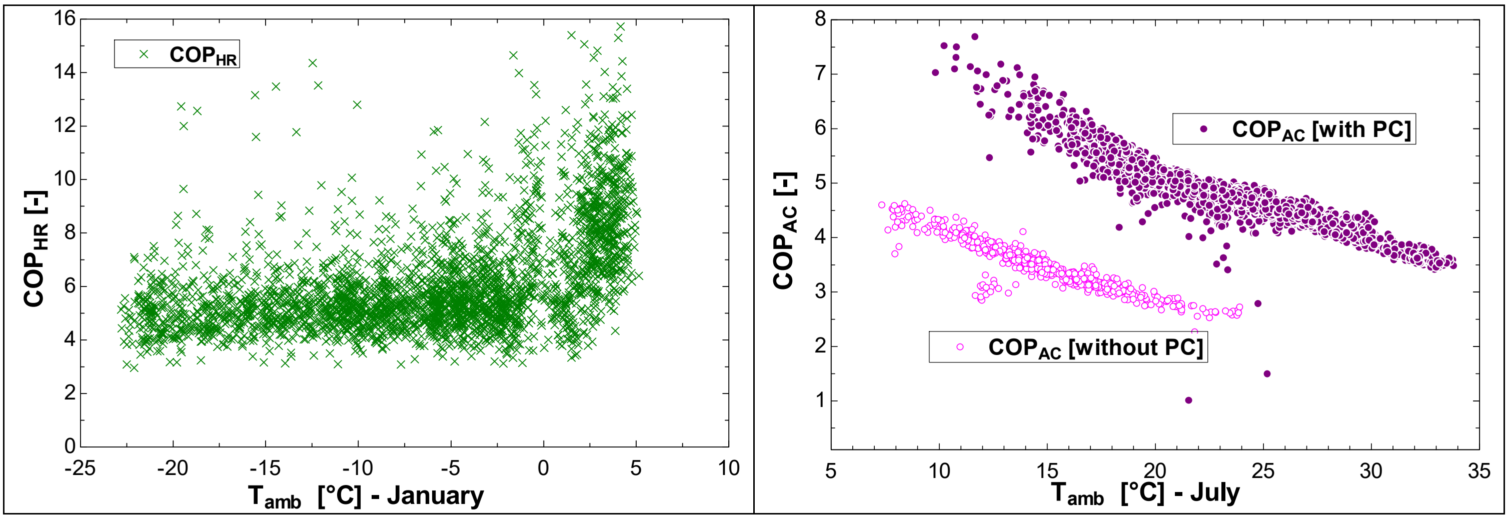


Figure 6: 15-min. averaged heating COP (COP_{HR}) (left) - Air conditioning COP (COP_{AC}) with and without parallel compression (right)

- COP_{HR}: heat is recovered with COP_{HR} ranging between 4 and 6 for ambient temperatures lower than 0°C. The definition of COP_{HR} in equation (16) considers Q_{SPH} and Q_{TWH} combined and as a single heating demand. This makes it directly comparable to heating COP of a heat pump (referred to usually as *COP1*) since the CO_2 system heat recovery control is mainly governed by space heating demand. This range of COP_{HR} is as, or more, efficient than the majority of the commercial heat pumps in the market. As can be observed in Figure 6, COP_{HR} has much higher values for the ambient temperatures within the 0-5°C range, because the heat is almost recovered for free from the system; i.e. discharge pressure is slightly increased to recover heat.
- COP_{AC}: COP_{AC} values for July are shown in Figure 6-right. To investigate the importance of using parallel compression (*PC*) when air conditioning is supplied by the integrated CO_2 system, COP_{AC} values for the cases "with-" and "without parallel" compression, abbreviated as "with PC" and "without PC", have been set apart by colour. The zero values show the time intervals where the system doesn't provide air conditioning, mainly summer mid-nights. Based on an earlier study, COP_{AC} of the CO_2 integrated system is expected to have higher values than HFC one for the ambient temperatures lower than 25°C (Karampour and Sawalha, 2015). This makes the system suitable in mild summers of Northern Europe. As can be seen in this figure, PC is rarely used for $T_{amb} < 12$°C and it is always used for $T_{amb} > 24$°C. The set-point to start the parallel compression is a certain by-pass valve opening degree, defined in the control system (Danfoss, 2017).

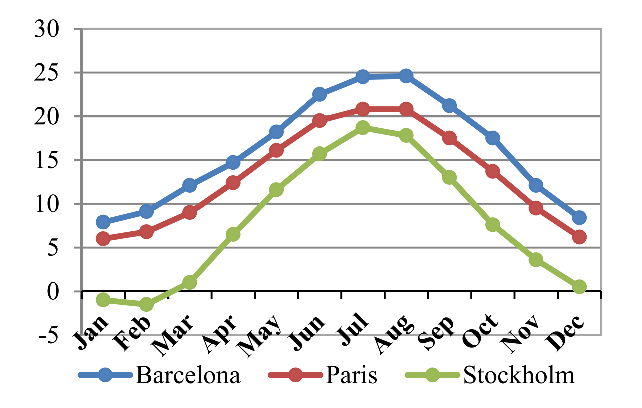
This defined value in the control system is based on the gas cooler and the receiver pressure transmitters readings. For ambient temperatures between 12 and 24°C, the comparison shows that the average COP_{AC} "with PC" is about 65% higher than "without PC" case. Consequently, COP_{tot} is also 7% higher when parallel compression is used during the air conditioning delivery of the system; this is not shown in this plots but extracted from the corresponding field measurement data. The reason PC is not used continuously in the warm season is the minimum capacity of parallel compressors; they require a minimum flow rate, and for the cases which there is not enough vapour in the receiver, PC compressors can't be switched on.

4.3 Annual electricity use comparison study

The integrated- or stand-alone energy systems in supermarkets are evaluated and compared using computer modelling in Engineering Equation Solver (EES) software (Klein, 2015). Most of the inputs to the models are adapted from field measurements analysis presented in this paper and a recent study by Sawalha et al. (2017). However, larger cooling and heating loads, compared to section 4.2, are used to represent better a medium size supermarket demands. These load assumptions are explained later in this section. Barcelona, Paris, and Stockholm are selected as three cities representing warm, mild and cold European climate conditions. The monthly average climate data have been extracted from Meteonorm software (Meteotest, 2016) and shown in Table 2. Three energy system solutions for Barcelona and Paris and four systems for Stockholm are compared. The energy system solutions are as follow, summarized in Table 2:

- System solution 1 (S1): an integrated CO_2 system similar to the system presented in this paper which provides refrigeration, heating, and air conditioning. Parallel compression is used for summer case when T_{amb} >12°C.
- System solution 2 (S2): a standard CO_2 trans-critical booster system with heat recovery and flash gas by-pass (no parallel compression). Air conditioning is provided by a stand-alone R410A system. The system main boundary conditions are summarized in Table 3.
- System solution 3 (S3): an R404A direct expansion with separate MT and LT units for refrigeration. This refrigeration system is not conventional in Sweden due to large HFC charge needed. Heating is provided by an air source R407C heat pump system and air conditioning by a stand-alone R410A system. This solution is considered as the reference system in this energy use comparison study since its refrigeration system is the most conventional system solution in many European supermarkets.
- System solution 4 (S4) – only for Stockholm: standard conventional HFC refrigeration system in Sweden is an indirect R404A system as presented by Sawalha et al. (2017) and shown in Figure 7. This refrigeration system is composed of two R404A loops; low temperature (LT) loop in the freezers is a DX R404A unit while medium temperature (MT) loop consists of an indirect secondary fluid (for example, propylene glycol or ethylene glycol) cycle connecting the MT cabinets to an R404A refrigeration unit. The refrigerant after the LT condenser is subcooled by a portion of the flow in MT brine loop. The R404A exit temperature from this sub-cooler is assumed to be 10 K higher than the medium level R404A evaporation temperature. On the high pressure side, the heat from MT and LT condensers are rejected to the ambient by indirect secondary fluid cycles connected to a dry cooler. The secondary fluid circulation pumps power is assumed to be 20% of the MT compressor electricity use, based on field measurements (Sawalha et al., 2017). The heating and AC systems in S4 solution are similar to S3.

Table 2: Energy systems and ambient temperatures for Barcelona, Paris, and Stockholm

	Refrigeration	Heating	AC	Monthly average ambient temperature [°C]
S1: Barcelona-Paris-Stockholm	CO_2 booster Ref + HR + AC + PC			
S2: Barcelona-Paris-Stockholm	CO_2 booster + HR		R410A AC	
S3: Barcelona-Paris-Stockholm	R404A DX	R407C HP	R410A AC	
S4: Stockholm	R404A indirect	R407C HP	R410A AC	

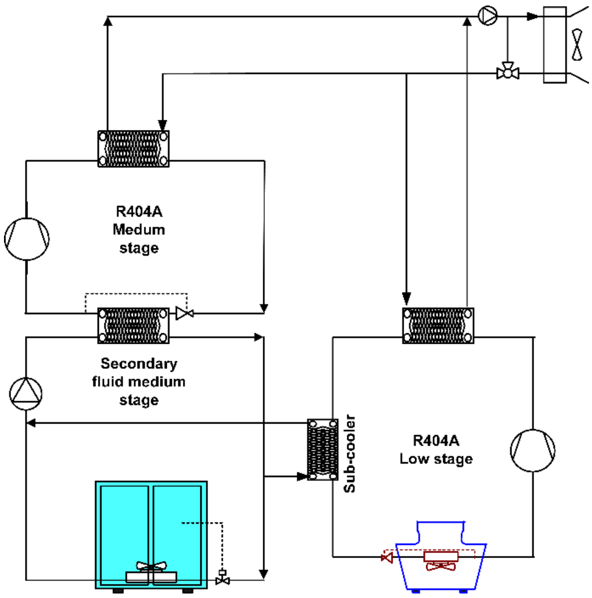


Figure 7: the S4 refrigeration solution, an indirect R404A system in medium temperature level and DX R04A in low temperature level

To represent a medium size supermarket, the supermarket modelled in this section has larger cooling and heating loads comparing with the CO_2 integrated system presented in the previous sections of this paper. A medium size supermarket represents better a facility where within all different energy systems solutions are competing. The refrigeration demand at MT level is dependent on T_{amb}; assumed to be 200 kW at 35°C and reduced linearly to 100 kW at 10°C, below which it remains constant. The LT level freezers are typically covered by glass doors and field measurement analysis (Sawalha et al., 2017) has shown that it remains almost constant throughout the year. 35 kW of LT refrigeration demand is assumed in this modelling. The heating demand for a medium size Swedish supermarket is obtained by the program CyberMart. Detailed descriptions and calculations of the program CyberMart can be found in the Doctoral Thesis of Arias (2005). The main heating demand in supermarkets is space heating and the set point to start supplying the heat is 10°C ambient temperature. It is estimated that at 10°C the heating demand is 40 kW for a medium size supermarket and it increases linearly to 190 kW at -20°C ambient temperature. The heat is supplied at a constant forward temperature of 40°C. The air conditioning demand is estimated based on personal communication with a Swedish supermarket chain. The supermarket chain has three typical small, medium and large supermarket sizes with design AC capacities of 60, 100 and

200-250 kW, respectively. The medium size supermarket is selected for this study has an AC load of 100 kW at 32°C and assumed to drop to zero at 10°C. It should be mentioned that the assumed heating and cooling loads for Barcelona and Paris are similar to Stockholm in this computer model. However, large differences are expected in reality depending on the relative humidity, building design and insulation, services in the supermarket, load ratio, etc.

The majority of temperature boundaries in the systems S1-4 are kept similar. Some of the minor differences between the boundaries include 5 K difference between condensing and outdoor temperatures in CO_2 gas coolers and HFC DX condensers while it is assumed to be 10 K for indirect HFC system. Furthermore, AC evaporation temperature for the CO_2 system is kept as 6°C in AC flooded evaporator but it is assumed 0° in DX HFC evaporators. A summary of main assumptions and boundary conditions can be seen in Table 3.

Table 3: Main boundary conditions and assumptions for the annual electricity use comparison study

Boundary condition	Assumed value	Notes/References
Medium stage evaporation temperature	-8°C for all systems	According to field measurement observations reported in this paper, Sawalha et al. (2017) and Sawalha et al. (2015)
Low stage evaporation temperature	-32°C for all systems	According to field measurement observations reported in this paper, Sawalha et al. (2017) and Sawalha et al. (2015)
Minimum condensation temperature	+10°C for all systems	According to field measurement observations reported in this paper, Sawalha et al. (2017) and Sawalha et al. (2015). HFC compressors minimum condensation temperature is according to Bitzer (2017) and Dorin (2017) semi-hermetic compressors.
Condenser/gas cooler approach temperature and sub-cooling	5 K for S1, S2, and S3 10 K for S4 (indirect) No sub-cooling for any system.	According to field measurement observations reported in this paper, Sawalha et al. (2017) and Sawalha et al. (2015)
Evaporator approach temperature and superheating	5 K for R407C HP system. 10 K internal superheating for all systems. No external superheating for any system.	According to field measurement observations reported in this paper, Sawalha et al. (2017) and Sawalha et al. (2015)
Air conditioning Evaporation temperature	6°C for CO_2 systems 0°C for R410A systems	CO_2 flooded evaporation results in higher degrees of evaporation in comparison with HFC DX evaporation temperatures.
Compressors efficiencies	As a function of pressure ratios, developed by using manufacturers data, semi-hermetic compressors.	All the total efficiencies are in the similar ranges of 55-60% for LT, 65-70% for MT, AC and HP compressors and slightly higher than 70% for parallel compressors.
Auxiliary powers	Fans power: 3% of the air-cooled condenser/gas cooler load in all systems. Pumps power: 20% of compressors power in the S4 solution.	Fan power according to communication with a major gas cooler/condenser manufacturer Pump power according to Sawalha et al. (2017)

The results of the annual electricity use comparison are shown in Figure 8. On the left y-axis, the annual electricity used for refrigeration, heating, and AC can be read. On the right y-axis, total electricity use ratio of different solutions can be compared with 100% of S3 reference system solution. For example, when 112% is read on the right y-axis for S1 in Barcelona it means that it uses 12% more electricity compared to S3 reference system.

As can be observed in the figure, CO_2 integrated system (S1) is the best alternative solution in Stockholm. CO_2 S1 and S2 solutions consume about 11% and 8%, respectively, less electricity than the S4 solution, which is the conventional

HFC solution in Sweden. CO_2 S1 and S2 solutions use 9% and 15% higher electricity than HFC S3 system in Paris, respectively, and 12% and 20% higher electricity in Barcelona.

Comparing AC electricity use of CO_2- and R410A-based systems, CO_2 S1 integrated system consumes about 19% less electricity in Stockholm, 17% less in Paris and 12% less in Barcelona. This implies that integration of AC into the CO_2 refrigeration system results in a decrease in AC electricity consumption. This advantage of the CO_2 integrated system versus HFC solutions is more significant in mild-cold climates and it has been shown that the AC function of the CO_2 integrated system has higher COP values than HFC systems for ambient temperatures lower than about 25°C (Karampour and Sawalha, 2015).

The electricity use of CO_2 S1 and S2 systems for heating is 5% less than HFC heat pump in Stockholm while it is 14% higher in Paris and 11% higher in Barcelona. The reason that HFC heat pump is more efficient than heat recovery from the CO_2 system in Paris and Barcelona is that this heat pump is air source and in mild winter conditions it has higher heat source temperature comparing with the CO_2 heating system.

To conclude, what can be interpreted from this comparison study is that CO_2 integrated solution (S1) offers an energy-efficient, environmentally- friendly and compact all-in-one solution for cold climates; however, some new modifications in this technology is required to increase its energy efficiency in warmer climates.

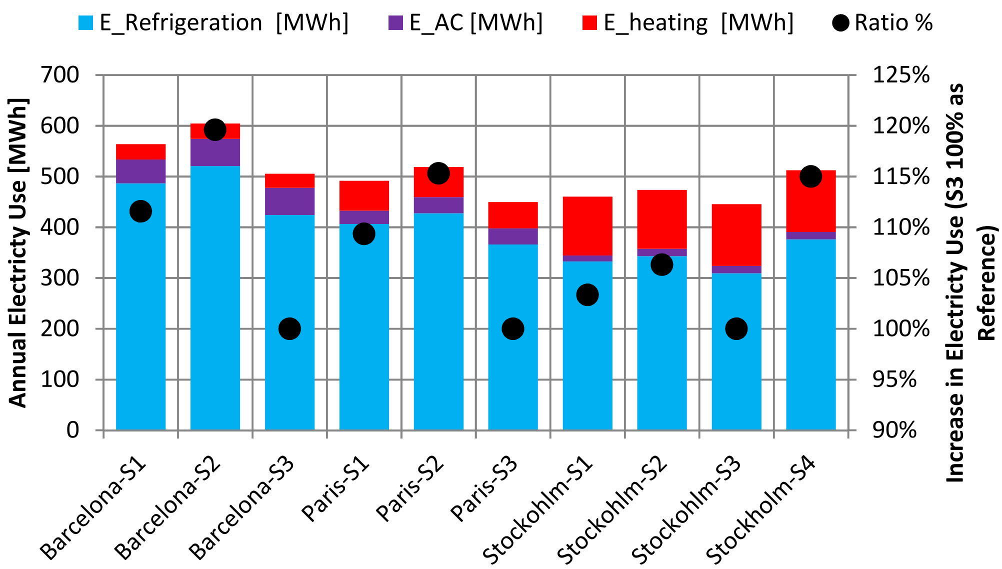


Figure 8: Annual electricity use comparison of energy systems S1-3 in Barcelona, Paris, and Stockholm. S4 is only in Stockholm.

The results presented in Figure 8 are based on the assumptions and boundary conditions presented in Table 3. The minimum condensing temperature for HFC systems is assumed as 10°C in this paper, similar to CO_2 systems. This value was assumed as 20-25°C in some other references (Girotto et al., 2004) (Mikhailov et al., 2011) (Royal, 2010) (Gullo et al., 2017). The authors didn't indicate that they had access to R404A DX field measurements and didn't motivate this assumption. It is only Girotto et al. (2004) who stated that this assumption has been adapted based on common practice; probably related to the operating envelope of HFC compressors and HFC thermostatic expansion valves. However, modern HFC compressors can provide minimum condensing temperatures as low as 5-10°C (Dorin, 2017) (BITZER, 2017) and electronic expansion valves can operate in a wider range of pressure differences comparing with thermostatic expansion valves.
This is a key assumption that will influence the results; therefore, if the minimum condensing temperature for R404A DX is assumed as 20°C in this paper, the annual electricty use difference between the CO_2-based (S1 and S2 solutions) with the reference HFC-based (S3 system) would be less than what is shown in Figure 8:

- +7% and +14% in Barcelona (compared to +12 and +20% in the reference case, refer to Figure 8)
- +1% and +7% in Paris (comapred to +9 and +15% in the reference case)
- and -9% and -7% in Stockholm (compared to +3 and +6% in the reference case).

This shows that CO_2 systems will perform more comparable to HFC solutions, by changing the assumption of R404A DX system minimum condensing temperature from 10°C to 20°C.

5. CONCLUSION

This paper investigated the key operating parameters and energy efficiency of an integrated CO_2 trans-critical system installed in a Swedish supermarket. Heating and air conditioning are integrated into the CO_2 trans-critical booster refrigeration system in this supermarket. The Supermarket has applied several features to improve energy efficiency including heat recovery for space and tap water heating, providing air conditioning, and the use of parallel compression. Using field measurements data, the key operating parameters in the CO_2 system are studied in selected warm (July 2014) and cold (January 2014) months. The daily- and hourly-averaged cooling and heating loads and COPs are analysed to explore the system performance in day-night and winter-summer running conditions. Furthermore, the integrated energy system concept is modelled and compared with separated stand-alone energy systems in three different European climate conditions; Stockholm for cold, Paris for mild and Barcelona for warm climates.

The cooling and heating loads delivered by the CO_2 integrated system are studied daily and hourly to quantify the performance of the system in warm and cold periods. The study shows that the system is able to provide the entire air conditioning demands and a great share of the available heat is recovered for tap water and space heating. Studying the loads fluctuation, it is shown that some loads are rather constant throughout the cold and warm periods; this includes LT freezers refrigeration and tap water heating loads. On the other hand, space heating and air conditioning loads are, as expected, strongly dependent on the climate conditions. These loads are also influenced by the presence of the customers in the supermarket. MT cabinets refrigeration load is found to be relatively stable in warm and cold months. However, it is about 5-10 kW higher in the warm month due to higher humidity content of the air.

Summer period performance analysis shows that the system runs in sub-critical floating condensing mode when the outdoor temperature is lower than 25°C. For higher outdoor temperatures the system runs in trans-critical mode where the gas cooler pressure follows an optimization algorithm to maximize the COP of the system. The pressure in the receiver is regulated to provide the AC system with the required 7-8°C forward secondary fluid. Parallel compression is used as the regulator of the receiver pressure for ambient temperatures higher than 25°C, while for ambient temperatures lower than 15°C flash gas by-pass valve is used. Both of these receiver pressure control strategies are used in the 15-25°C ambient temperatures region. The importance of parallel compression usage during AC delivery has been studied; it is found that COP_{AC} and COP_{tot} of the system are 65% and 7% higher using "AC with PC" comparing with "AC without PC", respectively. Air conditioning (AC) study shows that AC is delivered with high COP values in the case of using parallel compression.

Winter period performance analysis shows that the high pressure side of the system is controlled for heat recovery. The pressure is fixed to 80-85 [bar] and the gas cooler is by-passed to recover the entire rejected heat when the ambient temperature is very low.
The heat recovery function of the system is analysed for both tap water and space heating circuits. The measurements analysis shows that the system is able to provide the required tap water temperature of 55-60°C throughout the year. The system is also able to provide the space heating forward temperature of up to 45°C. Furthermore, heat recovery study proves that CO_2 system provides the heating demands with high COP_{HR} values, usually within the 4-6 range. This efficiency is comparable to the majority of commercial heat pumps in the market.

Using computer simulation models, the CO_2 integrated system is compared with other alternatives of energy systems in supermarkets, it is shown that the system consumes about 11% less electricity than stand-alone conventional HFC solutions for refrigeration (i.e. indirect HFC), heating and AC in North of Europe. In this region, the integrated heating and AC functions of the CO_2 integrated system consume about 5% and 19% less electricity, respectively, comparing with stand-alone HFC heat pump and AC systems. The electricity use for AC in the CO_2 integrated system is found to be always lower than stand-alone HFC units, 17% less in Paris and 12% less in Barcelona. However, total electricity use for refrigeration, heating, and AC is higher for CO_2 integrated system compared to conventional HFC in central and southern Europe; by 9-15% in Paris and by 12-20% higher in Barcelona. The conventional refrigeration system in central and southern Europe is assumed to use direct expansion HFC concept.

The energy efficiency study of the integrated CO_2 system functions confirms that it can be used as an environmentally friendly all-in-one supermarket energy solution suitable for cold climates. For the integrated CO_2 system to be considered as an efficient solution in mild and warm climates key modifications may need to be applied; such as flooded evaporation, by the use of ejector(s) or pump circulation, mechanical sub-cooling, and gas cooler evaporative cooling.

6. LIMITATION AND FUTURE WORK

The main focus of this study has been on the energy efficiency evaluation of an integrated CO_2 system. However, this study could be a broader reflection of the energy flows in the supermarket if there would be access to the measurements of building total heating demand/load or the auxiliary load provided by the district heating. Access to valid and sufficient measurements and processing this data is a time-consuming and challenging work in the studies of this kind.
What could be improved in the measurement system is the electric power measurements of the MT compressors, and direct heating and AC loads measurements on the water or secondary fluid side. This will help evaluate the accuracy of calculations and save a significant amount of time.

Study on the modifications needed to apply to increase the energy efficiency of the CO_2 system is ongoing in this research project. Based on the results, the definition and characteristics of *state-of-the-art* supermarket refrigeration system will be presented. The authors would like to build this system with the partner companies, install the proper measurements and collect and analyze the field measurements.

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