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Research Paper

Field measurements of supermarket refrigeration systems. Part II: Analysis of HFC refrigeration systems and comparison to CO₂ trans-critical

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HIGHLIGHTS

- Three Swedish supermarkets using typical HFC refrigeration solutions are analyzed.
- Field measurements of the HFC systems are compared to CO₂ trans-critical systems.
- Detailed analysis of field measurements combined with modelling is performed.
- CO₂ systems have higher COP than HFC systems for outdoor temperatures below 24 °C.
- CO₂ systems use about 20% less energy than a typical HFC system.

ABSTRACT

This part of the study investigates the performance of HFC refrigeration systems for supermarkets and compares the performance with alternative CO₂ trans-critical solutions. The investigated HFC system solutions are typical in supermarkets in Sweden. The analysis in this study is based on field measurements which were carried out in three supermarkets in Sweden. The results are compared to the findings from Part I of this study where five CO₂ trans-critical systems were analyzed.

Using the field measurements, low and medium temperature level cooling demands and COP's are calculated for five-minute intervals, filtered and averaged to monthly values. The different refrigeration systems are made comparable by looking at the different COP's versus condensing temperatures. The field measurement analysis is combined with theoretical modelling where the annual energy use of the HFC and CO₂ trans-critical refrigeration systems is calculated.

Comparing the field measurement and modelling results of COP's for HFC and CO₂ systems, the new CO₂ systems have higher total COP than HFC systems for outdoor temperatures lower than about 24 °C. The modelling is used to calculate the annual energy use of HFC and new CO₂ system in an average size supermarket in Stockholm, new CO₂ systems use about 20% less energy than a typical HFC system.

The detailed analysis done in this study (Part I and Part II) proves that new CO₂ trans-critical refrigeration systems are more energy efficient solutions for supermarkets than typical HFC systems in Sweden.

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1. Introduction

Supermarkets are known as one of the largest emitters of high GWP refrigerants in Europe according to the Intergovernmental Panel for Climate Change special report [1] and European Partnership for Energy and the Environment report [2]. As a consequence, they are one of the main refrigeration sectors which will be affected significantly by the recent EU F-gas regulation [3]. This regulation contains a ban to use any refrigerant with GWP higher than 150 for supermarkets centralized refrigeration systems larger than 40 kW from year 2022, with exception for primary cycle in cascade configurations to use refrigerants with GWP up to 1500.

The conventional refrigeration system in European supermarkets is direct expansion HFC-based systems in medium and low temperature levels. The amount of refrigerant charge in medium- and large-size supermarkets is in the range of hundreds to few thousands of kilograms and due to the long pipe runs and numerous piping connections, the leakage rate is reported to be 3–22% by different researches [1]. In some countries, such as in Sweden, indirect system solutions have been applied to confine the HFC use in...
The breakthrough of HFCs use in European supermarkets dated back to mid-90s. Adoption to the Kyoto protocol led to the phase out of CFCs, and use of HCFCs in new systems has been banned by an EU regulation in 2000 [4]. This shift from CFCs and HCFCs to HFCs has been accelerated and fortified by the Swedish government which didn’t allow new systems to be installed with R22 since 1998 and refilling of R22 from year 2002 [5].

The impact of implementing these international and national regulations on the Swedish refrigerants market has been studied and reported for a sample of 300–400 Swedish supermarkets by Engsten [6]. The study reported that the five largest refrigerants applied in Swedish supermarkets in mid-90s were R22 (40%), R134a (18%), R404A (17%), R502 (16%) and R12 (10%). This changed rapidly in a short period of time and R404A (70%) and R134a (24%) became the dominant refrigerants used by early 2000s.

R404A and R134a have remained the dominant refrigerants in Swedish supermarkets at the present time. R404A has a high GWP value of about 3900, and as mentioned earlier, will not be allowed to be used in European supermarkets according to the recent F-gas regulation. R134a has GWP value of 1300 and, moreover, it is not suitable for low temperature applications as the corresponding saturation pressures for evaporation temperatures lower than −25 °C is lower than atmospheric pressure.

To summarize, most of the conventional refrigerants in service today are not considered applicable choices for the near future. Alternative solutions using low GWP refrigerants and suitable for supermarket applications are in demand in the Swedish and European refrigeration sector.

CO2 has been suggested as an environmentally friendly refrigerant and has been applied in the supermarket refrigeration sector in the past two decades. The present dominant CO2 refrigeration technology uses solely CO2 where the system can operate in the trans-critical region; such systems are referred to as CO2 trans-critical. About 6500 supermarkets in the world use CO2 trans-critical solutions, according to the market development research company Shecco [7].

In addition to the direct environmental benefits of implementing CO2 trans-critical systems, their energy efficiency performance needs to be investigated. Many researchers ([8–14]) compared the performance of CO2 trans-critical to conventional HFC systems by computer modelling and/or laboratory testing. In general, it was found that CO2 trans-critical systems use either equivalent or lower energy compared to conventional HFC systems.

The findings from the theoretical and experimental comparisons was supported by research work on field measurements ([15–20]). More details of the research work can be found in the introduction of Part I of this study [21].

The studies on field measurements found in the literature usually had some limitations that can be crucial for the energy performance analysis and comparisons among different systems, such as: limited set of studied parameters [15], analysis based on annual electricity use comparison rather than detailed thermodynamic performance [17], and missing detailed description of the system design and layout [18,20].

In Part I of this study [21], field measurements of five supermarkets using three different CO2 trans-critical solutions were analyzed for periods of 4–18 months. The study included clear description of the system solutions and the analysis method was explained in details. The different COP’s used in the analysis were clearly defined and the systems were made comparable despite the different operating conditions and supermarket locations. The study showed the effect certain improvements have on increasing the efficiency of CO2 systems, old and new (improved) system solutions were compared in details; up to 40% increase in total COP was observed in the new systems compared to the old ones.
It is essential to compare the CO₂ trans-critical systems to conventional HFC solutions in order to conclude if CO₂ systems are energy efficient replacement. Therefore, the study in Part I is extended to include typical HFC systems in Sweden, which is presented in this paper (Part II).

This paper includes detailed description and analysis of three typical HFC systems in Sweden. The analysis method, assumptions, and COP definitions are clearly explained. The systems are compared to the CO₂ trans-critical systems in Part I of this study using field measurements and calculation models.

The work presented in both papers (Parts I and II) is a comprehensive study with high level of details that facilitates drawing concrete conclusions on the performance of CO₂ trans-critical versus conventional HFC systems in field installations. The work is combined with computer modelling which allows for annual energy use calculations that can be applied in different climates.

2. Systems description

The three HFC systems analyzed in this paper are variations of the same technical solution which is presented in a simple schematic in Fig. 1. The system consists of two parallel refrigeration units; one serves the medium temperature level (ML) cabinets and the other serves the low temperature level cabinets; i.e. freezers (FR). Cooling is provided to the ML cabinets by a heat transfer fluid (referred to as brine in this paper) in indirect loop arrangement, while direct expansion (DX) is applied on the freezers. The liquid after the condenser is sub-cooled on both ML and FR units with the use of a separate heat exchanger; denoted as sub-cooler. The two refrigeration units are not completely isolated, since the FR unit is sub-cooled by the brine at the ML level.

ML and FR units are equipped with internal (or suction line) heat exchangers (IHE) to provide further sub-cooling in the liquid line by superheating the relatively cold vapor at the compressor’s suction line. Electronic expansion valves are used in the ML cabinets while thermostatic expansion valves are used in the freezers.

The three supermarket refrigeration systems in this study are typical solutions in Swedish supermarkets; therefore, they are referred to as Reference Systems 1, 2 and 3, denoted as RS1, RS2 and RS3. The systems are located at different sites in Sweden. RS1 is in the small town of Arvidsjaur which is in the north of Sweden. RS1 is in operation since October 2008 with design cooling capacities of 87 and 18 kW for ML and FR units respectively; both ML and FR units use R404A as refrigerant. Two frequency controlled compressors operate in tandem in both ML and FR units.

The supermarket RS2 is placed in Tumba in Stockholm and in operation since October 2008. The design cooling capacity is 175 kW at ML and 36 kW at FR. RS2 refrigeration system consists of four units. Two units for ML with R407C as refrigerant, each ML unit has two frequency controlled compressors working in tandem. The other two units work with R404A and serve the freezers; each unit has a single frequency controlled compressor.

RS3 has been running since March 2008 in Birsta-Sundsvall, in the center-east part of Sweden. RS3 is the largest of the three HFC systems in this study with cooling capacities of 410 and 81 kW for ML and FR respectively. RS3 has two ML units using different type of refrigerant; R404A and R407C. Each unit has two frequency controlled compressors working in tandem. The freezers are served by two R404A units with a single frequency controlled compressor.

RS3 is the only system in this study that has the heat recovery function; where a heat pump is connected to the indirect loop at the condensers. However, the pressure of the high stage in the refrigeration units in RS3 is controlled in floating condensation mode; the heat recovery does not contribute to an extra power consumption in the refrigeration cycle.

The most important features of the three systems are summarized in Table 1.

3. Measurements and evaluation method

Cooling demands in the systems and COP's are the main parameters used to evaluate their performance. The COP's are used to compare the HFC systems with the CO₂ refrigeration system solutions presented in the first part of this study [21]. The COP of the system is the ratio of its cooling demand to its electric power consumption. The cooling demand can be determined by measuring/

![Fig. 1. A simple schematic diagram of the reference refrigeration system (RS), valid for the three cases studies: RS1, RS2, and RS3.](image-url)
estimating the mass flow rate of refrigerant and the enthalpies at the inlet and exit points of the evaporators. The mass flow rate of refrigerant is estimated using the compressors’ manufacturer data, where the enthalpies are determined in the system by measuring the pressures and temperatures at the needed points.

The following sub-sections explain in details the main performed measurements and the evaluation methods.

3.1. Pressure and temperature measurements

A single pressure measurement is installed at each pressure level in each unit in the system. Temperature measurements have been installed before and after each of the main components in the systems (except the evaporators). The pressure and temperature at key cycle points have been used to calculate the required thermophysical properties such as: enthalpy, density, and entropy.

Despite the missing temperature measurements before and after the evaporator the enthalpy difference across the evaporator can be determined by using the following two state points (see Fig. 2): liquid inlet to IHE (\(T_{IHE,liq,in}\)) and vapor exit of IHE (i.e. compressor inlet, \(T_{comp,in}\)).

Assuming no heat loss in the IHE, the enthalpy change on the warm and cold sides of the IHE (\(\Delta h_{IHE}\)) can be considered equal. Therefore, the enthalpy difference between “IHE,liq,in” and “comp,in” can be assumed equal to that across the evaporator, which is explained graphically in the plots in Fig. 2b. Fig. 2a is a simple schematic showing the location of the temperature measurement points around the evaporator and IHE.

As can be observed in the P-h diagram in Fig. 2b, the following relation can be established:

\[
\Delta h_{\text{evap}} = \Delta h_{\text{measur}} = h_{\text{comp.in}} - h_{\text{IHE.liq.in}}
\]  

(1)

The external superheat in ML unit can be assumed negligible due to the short suction line; therefore, the temperature at the compressor inlet can be assumed equal to the temperature of the vapor leaving the IHE (\(T_{IHE,vap.out}\)). However, in the case of FR unit the IHE is placed in the freezing cabinets where the suction line is quite long extending between the shopping area and the machine room. This will lead to large external superheat values; 12–31 K were measured in the CO₂ systems in part 1 of this study [21].

Using Eq. (1) to estimate the enthalpy difference across the evaporator for FR will result in including the external superheat as useful cooling load in the system which will result in higher COP than the actual value. Each degree Celsius of external superheat is expected to reduce the COP of ML and FR units with R404A by about 0.6%. The internal superheat, however, will have negligible positive effect on ML and FR COP’s, around 0.15% per °C. Superheat inside the IHE will have similar effect as the internal superheat in the evaporators. Full tables can be found in Granryd et al. [5].

In order to estimate the external superheat value, assumptions had to be made on the FR refrigeration unit’s. The only measured temperature around the IHE in the FR is the liquid temperature at the inlet (\(T_{IHE,liq,in}\)). The measured, assumed and calculated temperatures around the evaporator and IHE are indicated in the schematic in Fig. 3. Due to lack of measurements on this part of the circuit, the internal superheat in the freezer cabinets and the effectiveness of the IHE had to be assumed. 10 K of internal superheat was assumed, which agrees in average to what have been measured in the systems analyzed in part 1 of this study [21]. The effectiveness of the IHE is assumed 50%.

The vapor exit temperature from the IHE, \(T_{IHE,vap.out}\), is calculated using the following equation for the effectiveness of the heat exchanger:

\[
E = \frac{(m \cdot c_p)_{\text{cold}} \times (T_{\text{cold.out}} - T_{\text{cold.in}})}{(m \cdot c_p)_{\text{hot}} \times (T_{\text{hot.in}} - T_{\text{hot.in}})}
\]  

(2)

where in this case \((m \cdot c_p)_{\text{cold}} = (m \cdot c_p)_{\text{hot}}\). Therefore, the equation becomes:

\[
E = \frac{(T_{\text{cold.out}} - T_{\text{cold.in}})}{(T_{\text{hot.in}} - T_{\text{cold.in}})} = \frac{(T_{IHE,vap.out} - T_{\text{comp.in}})}{(T_{IHE,liq.in} - T_{\text{comp.in}})}
\]  

(3)

By assuming internal superheat (i.e. \(T_{\text{comp.in}}\) is known), \(T_{IHE,vap.out}\) can be calculated from Eq. (3). The external superheat is then calculated as the difference between \(T_{\text{comp.in}}\) and \(T_{IHE,vap.out}\). Using this

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**Fig. 2.** (a) Schematic of temperature measurements around the evaporator and IHE in a ML unit. (b) P-h diagram with state points around the evaporator and IHE.
refrigeration system separate energy measurements are needed. An indirect method of estimating the energy use of each compressor, and unit, in the system is to use the compressor manufacturer data to curve fit the volumetric and the overall efficiencies (\( \eta_{\text{v}} \) & \( \eta_{\text{in}} \)) as function of the pressure ratio (PR). Compressor types and the curve fit of efficiencies curves for each compressor model in the system can be found in [23].

Using the overall efficiency of the compressor, its electricity use (kW) can be calculated using Eq. (4).

\[
\dot{E}_{\text{el,compr}} = \frac{m_v \cdot \Delta h_{\text{compr}}}{\eta_{\text{in}}} \tag{4}
\]

where \( \Delta h_{\text{compr}} \) is the isentropic enthalpy (kJ/kg) difference across the compressor and \( m_v \) is the mass flow of refrigerant (kg/s), which is calculated using Eq. (5):

\[
m_v = \eta_{\text{v}} \cdot \rho_{\text{compr,in}} \frac{f_{\text{nom}}}{f_{\text{nom}}} V_s
\tag{5}
\]

where \( \rho_{\text{compr,in}} \) (kg/m\(^3\)) is the refrigerant density at the compressor inlet, \( f_{\text{nom}} \) is the nominal frequency (Hz) and \( f_{\text{nom}} \) is the measured compressor frequency (Hz) (or the sum of the frequencies of the two compressors working in tandem), \( V_s \) (m\(^3\)/s) is the swept volume flow rate at the nominal frequency.

The circulation pumps in the indirect loops connected to the condensers (\( E_{\text{cond,pump}} \)) and ML evaporators (\( E_{\text{brine,pump}} \)) run constantly at full capacity. They are assumed to use energy equal to the installed capacity; which are 1.5 kW each for \( E_{\text{cond,pump}} \) and \( E_{\text{brine,pump}} \) in RS1, 3.0 kW each in RS2, and 6.0 kW each in RS3.

The indirect loop at the condenser is connected to ML and FR units, as can be seen in Fig. 1. For separate COP calculation for ML and FR the energy use of the pump in the condenser’s indirect loop needs to be divided between ML and FR units. The division of energy use of this pump was assumed to follow the cooling demand ratio; i.e. Load Ratio (LR), according to the expression in Eq. (6):

\[
\frac{E_{\text{cond,pump,ML}}}{E_{\text{cond,pump,FR}}} = \frac{Q_{\text{ML,cab}}}{Q_{\text{FR,cab}}} = LR
\tag{6}
\]

where \( Q_{\text{ML,cab}} \) (kW) is the cooling load in ML cabinets and \( Q_{\text{FR,cab}} \) (kW) is the cooling load in FR cabinets (freezers).

The energy use estimation of the compressors and pumps used in this study have been compared to the total energy used measured in the system; which included all compressors, pumps and dry cooler fans. The total estimated energy used is about 10% lower in average than the measured energy use for all the measured months; the difference can be attributed to the energy use of the dry cooler fans [23].

### 3.3. Data acquisition, synchronization and filtering

Temperatures, pressures and compressors’ electric motor frequencies are logged every 5 min, no further data synchronization is needed. Calculations for the main parameters for the systems analysis such as, cooling demands and COP, are calculated for each time step. These two system parameters are analyzed step-by-step for filtering the data. Negative cooling demand values or higher than the expected system’s capacity with the given boundaries are removed. Also points with negative or unrealistically high COP’s, for instance higher than 7, are not considered valid.

An additional filtering was applied for the speed of change in COP, an increase or decrease of COP by 50% in the step of 5 min is considered unrealistic and taken away.

Transient condition in the system (for instance at compressors’ stop-start, defrost, etc.) can cause instability in the data resulting in
the unrealistic values that have been filtered out. At transient conditions some measurements may respond slower than others, for example temperature versus pressure measurements. In a laboratory environment, test measurements are usually taken when the system is at stable conditions, similar approach has been followed in this study; however, the system is considered to run at stable conditions at all times except for certain points that have been eliminated with the explained criteria in this section. The monthly averages are based on the filtered data.

Between 7 and 15% of the data are filtered out for the medium temperature level cabinets, only 1–5% for freezers.

3.4. Cooling demand and COP calculations

The cooling demands in ML and FR units (kW) are calculated using Eq. (7)

$$Q_2 = \dot{m}_r \cdot \Delta h_{\text{evap}}$$ (7)

where $\dot{m}_r$ (kg/s) is obtained from Eq. (5) and $\Delta h_{\text{evap}}$ (kJ/kg) is obtained from Eq. (1) for the ML evaporator cooling load calculations. However, for the FR level the external superheat values presented in Table 2 were assumed and $h_{\text{comp,in}}$ is replaced with $h_{\text{comp,out}}$ in Eq. (1).

Since FR units are DX, then the cooling demand calculated using Eq. (7) is equal to the cooling load at the FR cabinets ($Q_{\text{FR,cool}}$). However, the evaporator of the ML unit provides cooling to the brine loop ($Q_{\text{ML,cool}}$) which cools the ML cabinets ($Q_{\text{ML,cool}}$) and sub-cools the FR unit ($Q_{\text{FR,sc}}$), this can be observed in Fig. 1. $Q_{\text{ML,cool}}$ is calculated, by the low temperature stage, assuming saturation condition at the exit of the condenser. The energy balance around the ML brine loop can be expressed in Eq. (8).

$$Q_{\text{ML,tot}} = Q_{\text{ML,cool}} + Q_{\text{FR,sc}}$$ (8)

Using Eq. (8) the cooling load (kW) at the ML cabinets ($Q_{\text{ML,cool}}$) can be separated.

The COP of the refrigeration system (or of the individual units) is the ratio of the cooling demand to the electric power consumed to provide the cooling demand. For the ML units COP is defined in the following equation:

$$\text{COP}_{\text{ML}} = \frac{Q_{\text{ML,tot}}}{E_{\text{compr,ML}} + E_{\text{brine pumps}} + E_{\text{cond pumps}, \text{ML}}}$$ (9)

where $Q_{\text{ML,tot}}$ and the total electricity consumption of the ML compressor/s $E_{\text{compr,ML}}$ in kW are calculated using Eqs. (7) and (4) respectively. $E_{\text{cond pumps,ML}}$ is then calculated using Eq. (6).

The COPML in Eq. (9) can be used to calculate the share of energy used by the ML unit to provide each of the cooling loads in Eq. (8). Therefore, the energy used by ML unit to provide the sub-cooling of FR unit ($E_{\text{ML,for FR}}$) can be calculated by the following relation:

$$\text{COP}_{\text{ML}} = \frac{Q_{\text{FR,sc}}}{E_{\text{ML,for FR}}}$$ (10)

where $E_{\text{ML,for FR}}$ is used in the definition of the COP of the FR units in the following expression:

$$\text{COP}_{\text{FR}} = \frac{Q_{\text{FR,cool}}}{E_{\text{compr,FR}} + E_{\text{ML,for FR}} + E_{\text{cond pumps,FR}}}$$ (11)

where $E_{\text{cond pumps,FR}}$ is calculated using Eq. (6).

The total COP for the whole refrigeration systems, COPtot, is defined as the ratio of the total cooling demand on the refrigeration cabinets at ML and FR, to the electric power consumption of all its compressors and pumps. Expressed as:

$$\text{COP}_{\text{tot}} = \frac{Q_{\text{ML,cool}} + Q_{\text{FR,cool}}}{E_{\text{compr,ML}} + E_{\text{compr,FR}} + E_{\text{brine pumps}} + E_{\text{cond pumps}}}$$ (12)

This expression of the total COP is sensitive to the load ratio (LR) which is expressed in Eq. (6). Therefore, the total COP can be calculated at certain load ratio according to expression in the following equation:

$$\text{COP}_{\text{tot,LR}} = \frac{1 + \frac{\text{LR}}{100}}{\text{COP}_{\text{ML}} + \text{LR} \cdot \text{COP}_{\text{FR}}}$$ (13)

Details of the derivation of this expression can be found in part 1 of this study [21]. This normalization of COP can help comparing systems with different load ratios. Typical load ratio value in an average size supermarket system in Sweden is approximately 3.

4. System analysis

4.1. RS1

Data for RS1 has been collected for the period of October 2008 to June 2009. The monthly average values of the total cooling demands in ML and FR cabinets are plotted in Fig. 4. Outdoor, condensing and evaporation temperatures are also plotted in Fig. 4. Condensing and evaporation temperatures presented in the plot are the average values of the corresponding temperatures of all units.

As can be observed in the plot, the evaporation temperatures at the medium and low levels are almost constant; about −8 and −28 °C. Cooling demands at ML and FR are around 42 and 11 kW respectively. The cooling demand at ML did not vary much during the studied period; mainly due to low outdoor temperature, even during June, since the system is located in the north of Sweden. Data for the warmest months of July and August were not available at the time of analysis.

It can be observed in the plot in Fig. 4 that the condensing temperature is at almost fixed level (about 13 °C) during the relatively low outdoor temperature period; from November 2008 to March 2009. The system during this period is controlled for the lowest allowed condensing temperature. In Fig. 5 the condensing temperatures are differentiated between ML and FR units instead of plotting the average. They follow a similar trend but condensing temperature of FR is averagely 6 °C lower than ML; differences in condenser size can lead to such discrepancy.

![Fig. 4. Monthly average values of total cooling demands in ML and FR cabinets, outdoor, condensing and evaporation temperatures. The period is October 2008 to June 2009 for RS1.](image-url)
The COP in the plot is for a load ratio of 3 using the expression in Eq. (13). As can be observed in the plot the COP increases with decreasing condensing temperatures and it stabilizes for the period of constant condensing temperature. It can be observed in the plot that the sub-cooling value is quite high for FR unit, up to 22 K, which is due to the use of a dedicated sub-cooler in the system, refer to the schematic in Fig. 1 for system layout. Such sub-cooler is able to keep the temperature of sub-cooled liquid relatively constant all over the year; therefore, high sub-cooling is achieved in warm months. ML sub-cooling is almost constant at about 10 K.

4.2. RS2

Data for RS2 has been collected for the period of November 2008 to June 2009. Figs. 6 and 7 represent RS2 key parameters as was done for RS1 in the previous section in Figs. 4 and 5 respectively. It can be observed in Fig. 6 that the monthly average cooling demands at ML is about 77 kW in the relatively cold period (November to March) and increases to just below 100 kW in June; however, the FR cooling demand is almost constant for the whole study period at about 21 kW.

It can also be observed in Fig. 6 that the condensing temperature follows the outdoor temperature; however, with larger temperature difference between condensing and outdoor temperatures when the outdoor temperature is relatively low: about 15 K in February compared to about 10 K in May. This may indicate that the system is running at the lowest allowed condensing temperature for this system when the outdoor temperature is relatively low.

The evaporation temperature in the ML cabinets is about –9°C which is close to the value in RS1 (–8°C), however, in FR cabinets the evaporation temperature is about –32°C which is lower than that in RS1; about –28°C.

COP_{ML}, COP_{FR} and COP_{tot} at load ratio of 3 for RS2 are plotted in Fig. 7. Average condensing and sub-cooling temperatures have also been plotted. These parameters show similar trends to those in RS1. However, sub-cooling at ML is slightly lower than in RS1.

4.3. RS3

Data for RS3 has been collected for the period of June to December 2009. The system's key performance parameters have similar trends to RS1 and RS2 but much higher cooling demands at ML and FR, as can be observed in Fig. 8. Evaporation temperatures at ML and FR cabinets are –8 and –30°C respectively.
It can be observed that the monthly cooling demands at ML varies between 105 and 140 kW; it corresponds to 25–34% of the design cooling capacity indicated in Table 1, this is the lowest ratios compared to RS1 (48%) and RS2 (44–57%).

The cooling demand at ML increased with outdoor temperature, which is also observed in RS2. This is a result of lack of doors in most of the medium temperature cabinets and the higher humidity in the indoor air in summer. RS1 is installed in the north of Sweden and was not analyzed for the summer period so the relation between outdoor temperature and cooling demand at ML was not possible to observe clearly.

The cooling demand for FR cabinets is constantly around 40 kW, resulting in RS3 having the lowest value of cooling capacity when normalized by the design cooling capacity (approximately 50%, against 61% and 55% for RS1 and RS2 respectively). RS3 seems to have a much larger installed capacity, related to the measured cooling needs, compared to RS1 and RS2.

The cooling demands at the low temperature level did not vary much during the analyzed period for the three reference systems. This is mostly due to the use of glass lids that separates the cold air in the freezers from the supermarket environment.

A key feature that distinguishes RS3 from RS1 and RS2 is the heat recovery function where part of its heat is rejected to a heat pump connected to the secondary loop at the condenser side. Heat recovery is needed when the outdoor temperature is low and that usually coincides with the refrigeration system running at the minimum allowed condensing temperature, this can be observed for the months of October to December in Fig. 8 where the outdoor temperature drops below 10°C. Therefore, the heat recovery connection/control does not have strong influence on the system’s performance, since the system will be running at the minimum condensing temperature allowed at low outdoor temperatures anyway.

5. System comparison

The different COPs of RS1, 2 and 3 are plotted against the condensing temperatures, in the following figures, where the average evaporation temperature for each system is indicated at the corre-

**Fig. 9.** \( \text{COP}_{\text{ML}}, \text{COP}_{\text{FR}}, \text{COP}_{\text{tot}LR=3} \) at load ratio of 3 for RS3 are plotted in Fig. 9. Average condensing and sub-cooling temperatures have also been plotted. These parameters show similar trends to those in RS1 and RS2.

**Fig. 10.** \( \text{COP}_{\text{ML}} \) at different condensing temperatures for RS1, 2 and 3. Evaporation temperature for each system is indicated at the plot’s legends.

**Fig. 11.** \( \text{COP}_{\text{FR}} \) at different condensing temperatures for RS1, 2 and 3. Evaporation temperature for each system is indicated at the plot’s legends.

responding 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. This will bring focus on the performance of the refrigeration system.

\( \text{COP}_{\text{ML}} \) as defined in Eq. (9) for the three reference systems are plotted in Fig. 10 against the condensing temperatures.

It can be observed in Fig. 10 that RS1 and RS2 have similar \( \text{COP}_{\text{ML}} \) values while RS3 has lower by about 14–20%, which can be attributed to the high electric power consumption of the circulation pumps at condenser and evaporator loops consumed in RS3; 12 kW in total compared to 3 kW for RS1 and 6 kW for RS2. This corresponds to 20, 18 and 24% of overall compressors power for RS1, RS2 and RS3 respectively at condensing temperature of about 25°C.

RS3 also has lower sub-cooling than RS1 and RS2 at the same condensing temperatures, which can be observed in the ML and FR sub-cooling plots (\( \Delta T_{\text{sc,ML}} \) and \( \Delta T_{\text{sc,FR}} \)) in Figs. 5, 7 and 9. The lower sub-cooling in RS3 contributes to its relatively lower \( \text{COP}_{\text{ML}} \).
5.1. Comparison with CO₂ trans-critical refrigeration system

Five supermarket refrigeration systems working with CO₂ as refrigerant have been studied in details in the first part of this study [21]. The five CO₂ systems are trans-critical, where the system solution is seen as an alternative to the conventional refrigeration systems in Sweden, which are presented in this paper. The CO₂ trans-critical systems discussed in part I of this study are referred to as TR1, TR2 (NO SC), TR3, TR4, and TR5. The TR systems that will be included in the comparison in this paper are TR1 and TR2 (NO SC) since they represent the early installations of such systems in Sweden. The legend NO SC (No Sub-cooling) in TR2 indicates that the effect of the ground source sub-cooling on the performance of TR2 has been disabled.

TR4 and TR5 are also included in this study because they represent newer CO₂ systems. TR3 has been excluded from the comparison because it had strong effect of sub-cooling in heat recovery mode, especially at high discharge pressures, which makes a direct comparison with HFC systems inconclusive.

The analysis that has been performed in the first part of this study for the TR systems followed the same approach in this paper. COP_ML and COP_FR for all the RS and TR systems are comparable since the definition of the COP is the same; it is the ratio of the cooling demand (at ML or FR) to the electric power consumed to provide the cooling demand. The necessary energy balance calculations have been made for the different system groups to fulfill this definition. COP_ML is calculated for a load ratio of 3 \((\text{COP}_{\text{ML,LR=3}})\) following the definition in Eq. (13).

\[
\text{COP}_\text{ML} = \frac{Q_{\text{ML}}}{W_{\text{elec}}} \quad \text{(13)}
\]

\[
\text{COP}_\text{FR} = \frac{Q_{\text{FR}}}{W_{\text{elec}}} \quad \text{(11)}
\]

\[
\text{COP}_{\text{tot}} = \frac{Q_{\text{tot}}}{W_{\text{elec}}} \quad \text{(12)}
\]

\[
\text{COP}_{\text{tot,LR=3}} = \frac{Q_{\text{tot,LR=3}}}{W_{\text{elec}}} \quad \text{(13)}
\]

COP₂₅, calculated using Eq. (11), is plotted versus the condensing temperature for the three systems in Fig. 11.

It can be observed in Fig. 11 that the RS1 has the highest COP₂₅ which is mainly due to the higher average evaporation temperature compared to RS2 and RS3. It can be also observed that despite the higher evaporation temperature in RS3 compared to RS2, still COP₂₅ for both systems are quite similar; the reason for this is related to the higher pumping power in RS3.

Combining the COP получищих and COP₂₅ in a total COP using the expression in Eq. (13) for a system with load ratio of 3 results in the plots presented in Fig. 12.

The reference systems have similar design and the boundary conditions are comparable, this is why the systems have comparable COP₂₅ values as can be observed in Fig. 12. The main reason for the lower COP₂₅ value for RS3 is the relatively lower sub-cooling in the ML units and the higher pumping power as result of an installed capacity much higher that the measured cooling load.

Fig. 12. COP₂₅ at different condensing temperatures for RS1, 2, and 3. Evaporation temperatures at ML and FR for each system are indicated at the plot’s legends.

5.2. Modelling

In order to calculate the annual energy use of CO₂ systems and RS computer models have been developed using Engineering Equation Solver (EES) [24]. The computer models calculate the COP of the systems at different outdoor temperatures. The three systems that have been modelled are: RS, CO₂ system in old installations, and CO₂ system in new installations. The modelled RS represents the three reference systems presented in this paper, the older CO₂ system represents the solution in TR1 and TR2 (NO SC) systems, and the newer CO₂ systems represent the solution in TR4 and TR5 systems.

Table 3 includes the input parameters to the calculation models of the three system solutions. The input parameters are based on the field measurement data, where average values for each system group (i.e. RS, CO₂, and newer CO₂) are usually used.

RS2 and RS3 use R407C in medium temperature stage; in the modelling; however, only R404A has been used to represent the three reference systems. The influence of using R404A instead of R407C does not have strong influence in the resulting system COPs.

The calculated COP₂₅ for the three systems at different condensing temperatures are plotted in Fig. 16. COP₂₅ from the field measurements for RS123 are also plotted.

As can be observed in the plots in Fig. 16 good agreement is evident between the modelled and field measurement's COP₂₅ for RS123. It can also be observed that the older CO₂ systems have lower efficiency than RS at all modelled condensing temperatures. However, the newer CO₂ system solutions have higher COP₂₅ than...
RS at condensing temperatures lower than about 24 °C. The main reasons for differences in efficiency between old and new CO₂ systems have been discussed in details in Part I of this study [21].

Based on the field measurement data, 5 K can be assumed as temperature difference between condensing and outdoor temperatures for CO₂ systems. However, for RS, 10 K is assumed due to the use of indirect loop for heat rejection; i.e. brine loop connecting the condenser and the dry cooler, as can be observed in the schematic in Fig. 1. COPₜₐ₉ at load ratio of 3 for newer CO₂ systems and RS versus outdoor temperatures are plotted in Fig. 17. COPₘ₉ and COPₚ₉ versus outdoor temperatures for the same systems are plotted in Fig. 18. Minimum condensing temperatures for both systems in this calculation is assumed to be 10 °C.

The plots of COPₜₐ₉ at LR of 3 in Fig. 17 show that newer CO₂ systems have higher efficiency than RS for outdoor temperatures lower than about 24 °C. The main contribution to the high COPₜₐ₉ in newer CO₂ systems comes from COPₘ₉, which is much higher in newer CO₂ systems compared to RS, as can be observed in Fig. 18. However, COPₚ₉ for both systems are comparable.

In order to calculate the annual energy use of the systems, the cooling demand profile as a function of the outdoor temperature is assumed to change linearly between maximum capacity of 200 kW at 35 °C and 50% of the maximum capacity at 10 °C ambient, below which the capacity remains constant. The cooling demand at the low temperature level is assumed to be constant at 33 kW, independent of the ambient temperature. These assumptions used for the cooling capacities are based on analysis of field measurements data ([25,23]); and representative of an average size supermarket in Sweden.

Using the COP values in Fig. 18 and the assumed load profiles for an average size supermarket in Stockholm/Sweden, the annual energy use for new CO₂ systems and RS can be calculated. The average hourly outdoor temperatures for Stockholm are obtained from Meteonorm software [26], the resulting bin-hour temperature profile is reported in Fig. 18. Hourly average temperature is lower than 24 °C for more than 95% time in a year; CO₂ systems have nearly always better performance in a cold climate region as Stockholm area.
Fig. 15. COPtot at different condensing temperatures for CO₂ systems compared with the three RS’s (RS123). Evaporation temperatures at medium and low levels for each system are indicated at the plot’s legend.

Table 3
Input parameters for the simulation models.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Older CO₂ system (TR1-TR2, NO SC)</th>
<th>Newer CO₂ system (TR4-TR5)</th>
<th>Reference system (RS123)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flash gas by-pass</td>
<td>No</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>ML compressors total efficiency</td>
<td>65%</td>
<td>70%</td>
<td>60%</td>
</tr>
<tr>
<td>FR compressors total efficiency</td>
<td>45%</td>
<td>55%</td>
<td>65%</td>
</tr>
<tr>
<td>ML evaporation temperature [°C]</td>
<td>-9.5</td>
<td>-7.0</td>
<td>-8.0</td>
</tr>
<tr>
<td>FR evaporation temperature [°C]</td>
<td>-34.0</td>
<td>-31.5</td>
<td>-30.0</td>
</tr>
<tr>
<td>ML internal superheat [K]</td>
<td>11.0</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>FR internal superheat [K]</td>
<td>10.0</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>ML external superheat [K]</td>
<td>16.0</td>
<td>8.0</td>
<td>0</td>
</tr>
<tr>
<td>FR external superheat [K]</td>
<td>18.0</td>
<td>13.0</td>
<td>10.0</td>
</tr>
<tr>
<td>ML sub-cooling [K]</td>
<td>0</td>
<td>0</td>
<td>7</td>
</tr>
<tr>
<td>FR sub-cooling [K]</td>
<td>NA</td>
<td>NA</td>
<td>18</td>
</tr>
<tr>
<td>Temperature difference outdoor/condensation [K]</td>
<td>5.0</td>
<td>5.0</td>
<td>10.0</td>
</tr>
<tr>
<td>Pumps consumption [% per compressors power consumption]</td>
<td>NA</td>
<td>NA</td>
<td>20%</td>
</tr>
<tr>
<td>IHE (effectiveness)</td>
<td>No</td>
<td>No</td>
<td>Yes (50%)</td>
</tr>
</tbody>
</table>

Fig. 16. Plots of calculated COPtot, LR=3 of older CO₂ systems, newer CO₂ systems, and RS. Field measurement values for RS123 are also plotted.
The energy consumption calculations show that RS has an annual energy use of about 405 MW h while new CO$_2$ system uses 20% less energy; i.e. about 322 MW h. For lower LR the newer CO$_2$ system will still have lower annual energy use because the COP$_{FR}$ of both systems are comparable, as observed in Fig. 18. The savings in annual energy use in newer CO$_2$ systems will still be around 20% if the load at the low temperature level is increased to 50 instead of 33 kW in this calculation.

The analysis in this section was done to Stockholm city in Sweden; however, the COP plots and the calculation method, which was explained in details, can be used to analyze the performance of the systems in other climates.

6. Conclusions

Field measurements of three supermarkets in Sweden using typical HFC refrigeration system solution were analyzed for periods of 7–9 months. The refrigeration systems, the analysis method, and the required assumptions are explained in details. The reference systems were compared to alternative CO$_2$ trans-critical systems discussed in details in Part I of this study. The different refrigeration systems are made comparable by looking at the different coefficients of performance (COP’s) versus condensing temperatures. The field measurement analysis is combined with theoretical modelling where the annual energy use of the HFC and CO$_2$ trans-critical refrigeration systems was calculated.

Monthly averages of cooling demands ($Q_a$), low temperature level COP ($\text{COP}_{FR}$), medium temperature COP ($\text{COP}_{ML}$) and total COP ($\text{COP}_{\text{tot}}$) were plotted for the three reference HFC supermarkets. The cooling demands at the low temperature level did not vary much during the analyzed period for the three reference systems. However, at the medium temperature level the cooling demand increased with outdoor temperature increase, observed in RS2 and RS3.

$\text{COP}_{\text{ML}}$, $\text{COP}_{FR}$ and $\text{COP}_{\text{tot},LR=3}$ for the three RS’s have been plotted against the condensing temperature. RS3 had lower $\text{COP}_{\text{tot}}$ mainly due to higher energy use for the circulation pumps and lower sub-cooling. RS1 had higher $\text{COP}_{FR}$ due to higher evaporation...
temperature in the freezers. COP_{Tot,LR}=x for RS was relatively lower than RS1 and RS2 due to the combined effect of lower COP_{ML} and COP_{FR}. COP_{Tot,LR}=y for the three RS varied between about 4 and 2.5 for condensing temperatures between 12 and 28 °C respectively.

Comparing the field measurement results of COPs for RS and CO₂ systems shows that the new CO₂ systems have higher COP_{ML} than RS and old CO₂ systems. COP_{FR}, however, is higher for RS123 compared to new and old CO₂ systems, this is mainly due to the positive effect of the large sub-cooling (10–30 K) of FR by ML units.

Calculation models for RS and CO₂ systems have been developed in order to calculate annual energy use of the systems. The computer models were used to calculate the COP of the systems at different outdoor temperatures. The calculation results show that COP_{ML} for the new CO₂ system are higher than RS while COP_{FR} are at the same level. COP_{Tot,LR}=z for the new CO₂ system is higher than RS for outdoor temperature lower than 24 °C.

Based on the modelling results, the annual energy use of the new CO₂ system in an average size supermarket in Stockholm is about 20% lower than RS system.

The detailed analysis done in this study (Part I and Part II) proves that new CO₂ trans-critical refrigeration systems are more energy efficient solutions for supermarkets than typical HFC systems in Sweden. The analysis method and results presented in this study can be used to expand the analysis for different case studies in other climate conditions which will help verifying the potential of CO₂ trans-critical solution in other countries.

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