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Investigation of Heat Recovery in CO₂ Trans-critical Solution for Supermarket Refrigeration

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Abstract

Using computer simulation modeling this study investigates the performance of a CO₂ trans-critical system with heat recovery from the de-superheater. The influence of sub-cooling (or further cooling) in the condenser/gas cooler on system performance is investigated. Following the suggested control strategy in this study, the extra operating energy demand required to recover the needed heating energy from the analyzed CO₂ system is smaller than what a typical heat pump would require for the same load. This is the case for almost all ambient temperatures over a full season. When taking the simultaneous heating and cooling loads into account, the CO₂ trans-critical system has lower annual energy usage in an average size supermarket in Sweden when compared to a conventional R404A refrigeration system with separate heat pump for heating needs. CO₂ trans-critical systems are efficient solutions for simultaneous cooling and heating needs in supermarkets in relatively cold climates.

Keywords

Carbon dioxide, CO₂, Supermarket, Refrigeration, Modeling, Heat recovery

Nomenclature

COP	coefficient of performance, -
COP1	heating coefficient of performance, -
COP2	cooling coefficient of performance, -
$\dot{E}$	power consumption, kW
FC	floating condensing
HP	heat pump
HR	heat recovery
HRR	heat recovery ratio, -
HVAC	heating ventilation and air conditioning
$\dot{Q}$	capacity, kW
$\dot{Q}_1$	heating capacity/demand, kW
$\dot{Q}_{HR}$	heating demand that can be covered by the de-superheater, kW
P	pressure, bar
T	temperature, °C
Subscript	
amb	ambient
cab	cabinet
cond	condenser/condensing

De-sup	de-superheater
disch	discharge
el	electric
ev	evaporation
f	freezer/low temperature level
FC	floating condensing
GC	Gas cooler
HP	heat pump
HR	heat recovery
m	medium temperature level
opt	Optimum
SC	sub-cooling
tot	total

1. Introduction

Supermarkets are intensive energy consumers with constantly increasing number of installations. Simultaneous cooling and heating is often needed in a typical supermarket in northern Europe. Sometimes the amount of heating energy rejected from the refrigeration system is even higher than the supermarket's needs. The rejected heat can be utilized with proper system design and control.

Heat recovery system solutions in supermarkets are used mainly to heat up the space air. The practical experience indicated that though seemingly high quantity of heating energy is rejected by supermarket refrigeration systems, only 40-70% of the necessary heating energy can be recovered (Arias, 2005). Arias (2005) explains that in a typical Swedish supermarket HVAC and refrigeration systems are installed and operated by separate companies. Therefore, HVAC and refrigeration systems are coupled by heat exchanger which in a way separates the entities, i.e. responsibilities, so the control of both systems is done independently. This has been cited as one of the reasons for the low heat recovery percentage.

An important parameter in the analysis of the heat recovery potential for refrigeration system is the match between the heating and cooling demands. At low outdoor temperatures the relative humidity is low and so is the cooling demand on the system, at the same time the heating demand in the supermarket increases. Therefore, the relative size/capacity of the refrigeration system to the heating needs in the supermarket and the seasonal profile must be considered to design an efficient refrigeration system for simultaneous cooling and heating needs.

Heat recovery from refrigeration systems in supermarkets has been typically done by elevating the condensing pressure to a level where the coolant fluid of the condensers has the required supply temperature to the heating system (Cecchinato et al., 2010), this method is known as fixing the head pressure. This is being mostly used with systems running with

HFC's where the condensers reject the heat to a coolant loop with a dry cooler, the indirect heat rejection is mainly applied to minimize the refrigerant charge.

The environmentally friendly natural refrigerants are seen to be a long term solution in refrigeration applications especially in supermarkets. Supermarket refrigeration systems working with natural refrigerants provide new possibilities for heat recovery. Cecchinato et al. (2012) analyzed several combinations of refrigeration and heat recovery solutions and compared their energy use to a conventional system. Some of the natural based solutions showed higher energy efficiency compared to the conventional one.

Being natural, CO₂ does not present unforeseen threats to the environment and it is relatively safe. Certain system solutions based on CO₂ as the refrigerant, cascade and trans-critical, are seen as efficient solutions for supermarket refrigeration (Sawalha, 2008).

The common heat recovery method of fixing the head pressure used in conventional refrigeration systems is not suitable for systems running with only CO₂ as the working fluid in the system, i.e. trans-critical systems. This is mainly because CO₂ trans-critical systems for supermarket applications have relatively low COP compared to conventional at high heat sink temperatures; this is why it have been installed mainly in cold climates with direct heat rejection to the ambient air where the operation is mostly sub-critical (Sienel T. and Finck O., 2007).

With CO₂ systems it is possible to recover heat by cooling, i.e. de-superheating, the compressor's discharge gas due to its relatively high temperature. About 55°C for isentropic compression between -10 and 20°C with 10K superheat, compared to about 35°C in case of R404A which is mostly used in conventional systems. When the CO₂ system operates around the critical point the potential to recover heat from discharge gas is higher and the system has to be properly controlled to obtain good system efficiency.

Reinholdt and Madsen (2010) investigated two operation strategies for maximization of refrigeration COP and for maximization of amount of recovered heat at different hot water

temperature requirements. The study concluded that the heat recovered rates from the system can be increased considerably. Colombo et al. (2010) studied the energy savings when applying heat recovery in CO₂ trans-critical refrigeration system in an existing supermarket. Considerably energy savings have been observed and the CO₂ system showed lower energy consumption when compared to an existing R404A system.

Tambovtsev et al. (2010) discussed aspects related to recovering heat from CO₂ trans-critical refrigeration system and emphasized the importance of applying the gas cooler bypass and an optimally tuned control algorithm. Some of the control strategies have been tested by Tambovtsev et al. (2011) and proved to be in line with the predicted system performance.

The research work done on heat recovery in CO₂ refrigeration system in supermarkets lacks the detailed analysis of the control strategy and how it affects the energy consumption of the system also how it can be compared to conventional solutions.

This study theoretically investigates the performance of a CO₂ trans-critical booster refrigeration system when operated to cover the simultaneous cooling and heating needs in an average size supermarket in Sweden.

2. Heat recovery solutions in supermarkets

A typical supermarket in Sweden uses between 35-50% of its total electricity use for refrigeration equipment (Lundqvist, 2000), similar is the case in the United States (Richard Royal, 2010; Arthur D. Little Inc, 1996). Consequently, the refrigeration system rejects considerable amount of heating energy which can be used to reduce or cover some of the common needs such as; floor heating, heating for HVAC system, service water heating and some special applications such as defrosting of evaporator coils.

When the refrigeration system is used for heat recovery it is usually coupled to a heating system that requires relatively low supply temperature. In an HVAC system about 33°C of

approach temperature for the heating system is required (Arias and Lundqvist, 2006; Minea, 2010), in this study 35°C has been used. Assuming 5K temperature difference across the heating system, 30°C return temperature from the heating system is used in this study.

The amount of heat removed from a refrigeration system at a certain temperature level depends on the refrigerant used in the system and the system solution applied; cascade, parallel trans-critical, or booster trans-critical, or indirect. It also depends on the heat recovery solution used to recover the heating energy from the refrigeration system.

A refrigeration system without control to recover heating energy should normally operate in floating condensing where the condensing temperature follows the ambient temperature to a minimum condensing level. This minimum level is a guideline adapted in the refrigeration industry mainly to ensure proper function of the expansion valve and to avoid cold condenser, which may cause problems for oil return and frost formation in cold weathers (Wulfinghoff, 1999). Different recommendations are given for the minimum condensing temperature that should be maintained, in some sources it should be around 20°C (Wulfinghoff, 1999; Richard Royal, 2010; Pearson, 2011), while as low as 5°C was suggested by Arias (2005). A minimum of 10°C condensing suggested by Arthur D. Little Inc (1996) has been adapted in the calculations in this study because it agrees with field measurements and experience from Swedish supermarkets.

In the floating condensing case, the heating needs in the supermarket are covered mainly by district heating or a separate heat pump system. Different system solutions for heat recovery are presented in Figure 1.

Figure 1: Simple schematics of selected heat recovery solutions

Figure 1a shows the layout of fixed head pressure heat recovery. In this heat recovery solution the system operates in floating condensing and when heating is required the condensing pressure is elevated in order to provide the proper temperature for the heating system. As can be seen in the figure, the coolant extracts heat from the condenser of the refrigeration system and rejects the required heat to the HVAC system before entering the dry cooler which rejects the remaining heat to the ambient. This solution is mostly applied in conventional HFC solutions but not suitable for CO₂ systems.

Figure 1b is a simple schematic of the heat pump cascade solution for heat recovery. In this solution, the heat pump extracts heat from the condenser coolant at low temperature and transfers it to the HVAC system at higher temperature levels. This system solution enables the use of rejected heat from the condenser while the refrigeration system operating at relatively low condensing temperature.

Figure 1c is a simple schematic of a system running with heat recovery from the de-superheater which is installed before the air cooled condenser. This heat recovery solution is viable in the systems operating with refrigerants that have relatively high discharge temperatures such as CO₂ and NH₃.

Similar arrangement to the cascade heat pump solution is a refrigeration system with heat pump at sub-cooling as in Figure 1d. In this case the heat pump is connected after the condenser in the refrigeration system so it operates at low condensing pressure when the ambient temperature is low and heating is needed. The heat pump at this mode recovers the heat from the refrigeration system and provides further sub-cooling improving its efficiency.

In general in the heat recovery operation mode, the condensing pressure of the refrigeration systems is controlled according to the temperature requirement of the heating system and the heating needs.

Solutions in Figure 1b, c, and d are viable for CO₂ due to the possibility to operate the system at relatively low condensing temperatures.

3. System description

In general, two temperature levels are required in supermarkets for chilled and frozen products; temperatures of around +3°C and -18°C are commonly maintained respectively. In such application, with a large difference between evaporating and condensing temperatures, the cascade or other two-stage solutions become favorable and are adaptable for the two-temperature level requirements of the supermarket.

The use of CO₂ in supermarket refrigeration started with applying CO₂ as the working fluid in indirect systems for freezing applications. Then it has been applied in cascade solutions mainly with R404A in the high temperature stage. After gaining experience and the availability of components CO₂ has been used in trans-critical system solutions, mainly in northern Europe. The two main trans-critical system solutions applied are the parallel and the

booster, the later is the latest in CO₂ systems development series and has been applied in most of the installations in Sweden.

The system analyzed in this study is a CO₂ booster system solution with heat recovery from the de-superheater, simple schematic presented in Figure 2. The discharge pressure of the system is controlled according to the required heat in the supermarket, when there is no heating need in the supermarket the discharge pressure is kept as low as possible following the ambient temperature.

The efficiency of the low temperature stage in the booster system in Figure 2 can be improved by introducing a receiver where the refrigerant expands after the condenser/gas cooler to a pressure slightly higher than the pressure of the medium temperature level; i.e. two stage throttling. The flash gas in the receiver is expanded before the high stage compressor. However, the system in Figure 2 was analyzed in this study due the high number of installations with similar system solution applied in Swedish supermarkets.

Figure 2: Schematic diagram of a booster system with heat recovery in the de-superheater

The performance of the CO₂ systems is sensitive to the condensing, or gas cooler exit, temperature when compared to direct expansion R404A systems; it has similar or higher COP at condensing temperatures lower than 25°C (Sawalha, 2008; Sienel T. and Finck O., 2007). This is why it is essential to keep the condensing, or gas cooler exit, temperatures of the CO₂ systems as low as possible and its application has been mostly in the relatively cold climates in Europe, over 400 installations (Kauffeld, 2010). In the system solution in Figure 2 heat is rejected directly to the ambient via the condenser/gas cooler so the system is running at the lowest condensing temperature possible.

The calculation models used in this study are written using Engineering Equations Solver (EES) software, its basic function is to provide the numerical solution to a set of algebraic equations. It has many built-in mathematical and thermo-physical property functions for refrigerants (Klein, 2006). The assumptions used in the calculations are based on the field measurements analysis of several supermarkets in Sweden (Johansson, 2009; Freléchox, 2009; Nidup, 2009; Tamilarasan, 2009).

Evaporation temperatures at the medium and low temperature levels are -10 and -35°C respectively. 10K of internal superheat is assumed in all evaporators, while external superheat values of 10 and 15K are assumed in medium and low temperature levels respectively. 5K of approach temperature difference is assumed in the gas cooler and de-superheater. Data sheets of commercially available compressors have been used to generate its total efficiency as a function of the pressure ratio, Dorin compressors TCS373-D and SCS362-SC have been used for the high and low stages respectively (DORIN Innovation, 2011). When the system operates in the trans-critical mode the discharge pressure is calculated according to the following equation (Sawalha, 2008; Liao et al., 2000):

$$P_{opt,disch} = 2.7 \cdot t_{GC,exit} - 6 [bars] \quad (1)$$

The cooling demand of the refrigeration system at the medium temperature level is dependent on the ambient conditions and assumed to change linearly between full capacity of 200 kW at 35°C and 50% of the full capacity at 10°C ambient, below which the demand remains constant. Similar cooling demand profile is suggested by Skovrup (2010); however, the change of profile with ambient temperature and the temperature below which it becomes constant could be different from one installation or climate zone to another.

The main reason for the change in cooling demand with ambient temperature is the change in relative humidity; which follows the ambient temperature. Air dehumidification is not usually applied in supermarkets in Sweden; therefore, the indoor relative humidity follows the outdoors. This results in higher frost formation rate on the evaporator coils which deteriorates its refrigerating capacity.

The cooling demand at the low temperature level is assumed to be constant at 35 kW independently of the ambient temperature. These assumptions used for the cooling demand are based on analysis of field measurements data (Freléchox, 2009; Tamilarasan, 2009) and representative of an average size supermarket in Sweden.

The COP of the system is calculated based on the above assumptions. The medium temperature (high-stage) and the low-stage COP's are plotted in Figure 3 against the ambient temperature. "COP_{2,low-stage}" is for the cycle between -35 and -10°C. The low temperature COP₂ (COP_{2,f}) is for the cycle between -35°C and the temperature in the condenser/gas cooler. COP_{2,f} is introduced in this plot to be able to compare the CO₂ booster system with a conventional system with R404A, presented later in this paper.

In this case the system is operated in floating condensing without control for heat recovery. As can be seen in the plot, the condensing (or gas cooler exit) temperature follows the ambient temperature with the minimum condensing temperature of 10°C.

Figure 3: Medium temperature (High-stage), low temperature and low-stage COP2 of the CO2 booster system

As can be seen in the plot, “COP2,low-stage” decreases with increasing ambient temperature. It is because in this specific booster solution without receiver (i.e. two stage throttling); the inlet temperature of the low stage expansion valve is equal to the exit temperature of the condenser/gas cooler, assuming no heat losses in the system.

For ambient temperatures lower than 5°C the condensing temperature is controlled to a fixed value of 10°C and it is possible to provide sub-cooling using the cold ambient air. However, the condenser is kept at a temperature higher than freezing to avoid any problems related to frost formation, so a minimum condenser exit temperature of 5°C is assumed, i.e. maximum of 5K sub-cooling is achieved. The COP2 of the system with this sub-cooling is plotted in dotted lines in Figure 3.

4. System analysis

The return temperature from the heating system is assumed to be 30°C, which is what can be expected as a return from HVAC system. When the refrigeration system is operated in the heat recovery mode, usually it is controlled so that when more heating energy is needed from the refrigeration system the discharge pressure is increased so more heating energy is available in the de-superheater. Figure 4 is a plot of the high stage cycle on a P-h diagram for different discharge pressure values, no sub-cooling is assumed.

Figure 4: CO2 P-h diagram with the high stage cycle of the booster system at different discharge pressures

As can be observed in the plot, the available heat that can be recovered from the de-superheater increases by raising the discharge pressure. At a discharge pressure higher than the critical point all the heat rejected by the system can be recovered in the de-superheater, the difference can be observed between steps (3) and (4) where 89 kW of heat can be recovered at discharge pressure of about 67 bars compared to 230 kW at about 88 bars. The discharge pressure values in Figure 4 are selected to cover the possible operating range of the system.

In order to evaluate the performance of the system in the heating mode the heating demand is calculated at different ambient temperatures for an average size supermarket in Sweden using the program CyberMart. The program CyberMart regards the supermarket system as several subsystems with all the different zones integrated, all the calculation details of the program can be found in the Doctoral Thesis of Arias (2005). The heating demand values have been plotted against ambient temperature and the linear plot in Figure 5 has been generated. As can be seen in the plot, heating is needed when ambient temperature is lower than 10°C; this assumption is based on analysis of the monthly average temperatures at the start and the end of the heating season for some cities in Sweden (Nidup, 2009). Similar analysis has been done for a case in the United States where it was observed that no space heating was required for ambient temperatures higher than about 9°C (Richard Royal, 2010).

Figure 5: Medium and low temperature level COP2 and heating demand for different ambient temperatures.

No sub-cooling is assumed and heating system return temperature is 30°C.

Figure 5 also shows a plot of the system's medium and low temperature (freezing) COP2 at different ambient temperatures. As can be seen in the plot, the system is controlled for heat

recovery so when the ambient temperature drops the COP increases due to lower condensing pressure. Above 10°C ambient, the COP2 in relation to ambient temperature follows the trends in Figure 3. When the system is controlled to cover the heating demand, below 10°C ambient, the discharge pressure is raised to match the heating demand (explained in Figure 4) and the corresponding COP2 of the refrigeration system drops.

Based on the assumed cooling and heating demands in the supermarket and the calculated systems COP2, in Figure 5, the booster system will have the power consumption presented in Figure 6.

Figure 6: Power consumption of the refrigeration system at different ambient temperatures in floating condensing and heat recovery modes. No sub-cooling is assumed.

As can be observed in the plot, with floating condensing the power consumption of the compressors decreases with decreasing ambient temperatures. The minimum condensing temperature is 10°C, so for low ambient temperatures the power consumption of the system is flat because the cooling demand is also assumed constant. In the case when the system is

controlled for heat recovery, for ambient temperatures lower than 10°C the power consumption increases in order to provide enough heating energy to the supermarket.

At lower ambient temperature than 0°C (i.e. heating demand higher than 90kW from Figure 5) the system operates at a discharge pressure higher than the critical. From Figure 4 it can be observed that for discharge pressures higher than the critical, more heat can be recovered from the system per unit increase in pressure. This explains the decreased steepness of the power consumption line in Figure 6 for ambient temperatures lower than 0°C.

In order to evaluate the system performance in heat recovery mode the heating COP ($COP_{1_{HR}}$) of the refrigeration system is used, it is expressed in the following equation:

$$COP_{1_{HR}} = \frac{\dot{Q}_1}{\dot{E}_{HR} - \dot{E}_{FC}} [-] \quad (2)$$

It is defined as the ratio between the heating demand, plotted in Figure 5, to the power consumed to provide the heating demand, which is the difference between the power consumption of the refrigeration system in heat recovery mode and floating condensing mode, i.e. the difference between the plots in Figure 6. The refrigeration system in this study is assumed to recover all the heating demand in the supermarket; therefore, $\dot{Q}_1 = \dot{Q}_{HR}$.

$COP_{1_{HR}}$ of the booster system with 30°C return temperature from the heating system is plotted in Figure 7. The case of 20°C return temperature from the heating system is also plotted. The figure clearly shows the advantage of having low return temperature from the heating system. It is practically difficult to achieve low return temperature down to 20°C from the HVAC system; however, this plot could be an indication of the gain in efficiency when operating lower than 30°C return temperature from the heating system.

Figure 7: COP_{1_HR} of CO₂ booster system (heating system return temperature of 20 and 30°C).

The refrigeration system in these calculations is not controlled for sub/further-cooling in the condenser/gas cooler when the system is running in the heat recovery mode. When the ambient temperature is low the heating demand is high and the discharge pressure is raised to recover heat, in this case the gas cooler can be operated to further cool the refrigerant before passing the expansion valve. This further cooling is referred to as sub-cooling in this study.

5. Influence of sub-cooling on the system's COP

Running the gas cooler to cool the refrigerant down in the heat recovery mode has two main effects on the system performance, it increased the system's COP₂ which is a positive influence but it also reduces the available heating energy for recovery from the de-superheater at certain discharge pressure; this is due to the smaller refrigerant mass flow rate running in the system with sub-cooling. The sub-cooling influence can be observed in the plot on the P-h diagram in Figure 8. The low stage cycle is not plotted in the figure for clarity. The influence of varying mass flow rate of refrigerant on heat transfer in the heat exchangers is not taken into consideration, constant approach temperature differences has been used in the heat exchangers.

Figure 8: The sub-cooling influence on system performance on P-h diagram

In order to recover the same amount of heating energy from the system with sub-cooling, it has to operate at higher discharge pressure than without sub-cooling. For the case of

recovering 55kW from the de-superheater, the system with sub-cooling (5°C condenser/gas cooler exit temperature) must operate at higher discharge pressure than without sub-cooling, but still it has higher COP₂ due to the positive influence of sub-cooling. This can be observed in Figure 9.

Figure 9: The sub-cooling influence on system performance on P-h diagram, operating the system to provide 55kW heat from the de-superheater for the cases of with and without sub-cooling.

It can be concluded from the analysis of Figures 8 and 9 that for the system to provide a certain heating demand it has to operate at a certain discharge pressure depending on the amount of sub-cooling provided by the condenser/gas cooler. In order to analyze the influence of the condenser/gas cooler operation on the system performance the system is analyzed for the following two cases:

5.1 Case 1

The condenser/gas cooler is operated to provide no sub-cooling in the sub-critical region. In the trans-critical operation the system rejects all the heat in the de-superheater. In Figure 10,

the heating demand is presented on the x-axis as the heat recovery ratio (HRR), which is defined as the ratio of the heating demand ($\dot{Q}_1$) (totally covered by the de-superheater; $\dot{Q}_1 = \dot{Q}_{1HR}$) to the total cooling demand at the medium temperature level; including the heat rejected from the low temperature cycle. HRR is expressed in the following equation:

$$HRR = \frac{\dot{Q}_1}{\dot{Q}_{m,tot}} [\%] \quad (3)$$

Where

$$\dot{Q}_{m,tot} = \dot{Q}_{m,cab} + \dot{Q}_{f,cab} + \dot{E}_{f,shaft} [kW] \quad (4)$$

7% of the electric power consumption of the booster compressor is assumed to dissipate as heat loss to the surroundings.

$\dot{Q}_{m,cab}$ and $\dot{Q}_{f,cab}$ are assumed constant and equal to 100 and 35kW respectively for temperatures lower than 10°C ambient. $\dot{E}_{f,shaft}$ is relatively small and ranges between 11 and 18kW; calculated using the COP_{2,low_stage} in Figure 3 and $\dot{Q}_{f,cab}$ of 35kW. Using equation (4) the resulting $\dot{Q}_{m,tot}$ ranges between 146 and 153kW; therefore it can be treated as constant at 150kW in the heat recovery mode conditions.

For a certain heating and cooling demands, i.e. HRR, the system will have to operate at specific discharge pressure, read on the right y-axis in Figure 10, and the corresponding $COP_{2,m}$ is read on the left y-axis. As an example, for a heating demand of 90kW (at 0°C ambient in Figure 5) the HRR equals 60%; using equation (3) $HRR=90/150$. In order to meet the heating demand of 90kW, i.e. $HRR=60\%$, from the de-superheater, the system will have to run at a discharge pressure of about 70bar and the $COP_{2,m}$ will be around 2.4, according to the plots in Figure 10.

Using the HRR term in this study instead of the heating demand directly makes the results in Figure 10, and later figures, applicable to similar system solutions with wide range of different heating and cooling demands.

Figure 10: COP_{2,m} and the corresponding discharge pressure of the booster system as a function of HRR. The case is for the system without sub-cooling in the condenser/gas cooler.

Three regions in the plot can be observed; “Region 1” is where the system is in sub-critical operation, the decrease in COP_{2,m} is sharp with increasing the demand for heat recovery. “Region 2” is where the system is about to switch to trans-critical operation; this region can be identified to start at few bars below the critical point, about 69 bars. In this region much larger amount of heating energy can be recovered with a slight increase in discharge pressure. “Region 3” is where relatively large increase in discharge pressure results in slight increase in the amount of heating energy to be recovered, consequently sharper drop of COP_{2,m} is observed. “Region 3” is where the isotherm of the CO₂ temperature at the exit of the de-superheater starts to become steep, the 35°C isotherm in Figure 4.

5.2 Case 2

The condenser/gas cooler is operating to provide sub-cooling in the sub-critical region and further cooling after the de-superheater in the trans-critical operation. The degree of sub-cooling that can be achieved in the system depends on the ambient temperature and the capacity at which the condenser/gas cooler operates.

Calculations are made for different condenser/gas cooler exit temperatures, $COP_{2,m}$ and the corresponding discharge pressures are presented in Figures 11 and 12 respectively. $COP_{2,f}$ is not plotted in Figure 11 for clarity purpose but it has similar trend to $COP_{2,m}$. The positive influence of sub-cooling explained in Figure 9 can be observed in Figure 11 as well; for example, when the system operates to provide heating that corresponds to HRR of 60%, the $COP_{2,m}$ increases from about 2.4 in the case of without sub-cooling to about 3.0 in the case of 5°C exit temperature of the condenser/gas cooler, this is despite the need to operate the system at higher discharge pressure; 68 compared to about 78 bars respectively, see Figure 12.

Figure 11: COP_{2,m} of the booster system as a function of HRR. The cases are for different condenser/gas cooler exit temperatures.

Figure 12: Discharge pressure as a function of HRR for different condenser/gas cooler exit temperatures.

It can be observed in “Region 1” in the plots of Figure 11 that the system will have the highest COP when controlled to achieve the lowest gas cooler exit temperature. Therefore, the condenser/gas cooler should operate at full capacity and the discharge pressure should be regulated to match the required heating demand from the refrigeration system.

This is correct up to a point, start of “Region 2” in Figure 11, where more heating energy will be needed and if the condenser/gas cooler is still running at full capacity the drop in COP will be steep, as can be seen in all plots in Figure 11, this is where the isotherm line starts to become steep, notice the 35°C isotherm in Figure 4. The discharge pressure at which the

COP starts to drop in a steeper trend is about 88 bars, as can be observed in the start of “Region 2” in Figure 12.

In order to operate the system at the highest COP possible in “Region 2” in Figure 11, the system must be operated at the maximum discharge pressure to achieve the highest COP, about 88 bars in this case, and the condenser/gas cooler capacity should be reduced so more heating energy will be available in the de-superheater to be supplied to the heating system. The operation will follow the arrow crossing “Region 2” in Figure 12. The maximum operating pressure for highest COP in heat recovery mode is dependent on refrigerant exit temperature from the de-superheater; it follows the same correlation for the optimum discharge pressure for maximum COP in a CO₂ refrigeration system, expressed in equation (1).

With increasing the heating demand the condenser/gas cooler should eventually be switched off so all the system’s heating energy can be rejected in the de-superheater, this is where “Region 3” in Figures 11 and 12 starts. Beyond this point the only way to recover more heating energy from the system is by increasing the discharge pressure. It can be observed in Figures 11 and 12 that higher than a discharge pressure of about 88 bars the system will have a relatively steeper drop in the COP and sharp increase in discharge pressure. If we assume that due to the sharp increase in discharge pressure, the start of “Region 3” is the limit of the refrigeration system to provide heating at reasonable efficiency then the refrigeration system can provide HRR of about 150%, i.e. 225 kW of heat in this case study.

Calculating for the system to run with optimum control to achieve the highest COP possible, the approach temperature difference in the condenser/gas cooler is assumed to be 5K when the condenser/gas cooler is running at full capacity in “Region 1” in Figure 11. The resulting medium and low temperature COP’s are plotted in Figure 13. The operating discharge pressure and condenser/gas cooler exit temperature are plotted in Figure 14.

Figure 13: COP_{2,m} and COP_{2,f} as a function of HRR. The cases are for operation with no sub-cooling in the condenser/gas cooler and for system controlled for highest COP possible.

Figure 14: Discharge pressure and condenser/gas cooler exit temperature as a function of HRR. The cases are for operation with no sub-cooling in the condenser/gas cooler and for system controlled for highest COP.

As can be seen in Figure 13, the COP is higher for the system using the condenser/gas cooler for sub-cooling. The three regions identified in earlier plots can also be identified in this plot. At the curves' edges with low HRR there is almost no difference in the COP between both cases, this is due to the relatively high ambient temperature so negligible or little sub-cooling can be achieved in the system.

In Figure 14, it can be observed that in "Region 1" the condenser/gas cooler exit temperature decreases following ambient temperature (increasing HRR) because the condenser/gas cooler is running at full capacity. The discharge pressure increases compared to the "no sub-cooling" case so the same amount of heating energy can be recovered. "Region 2" starts when the system reaches the maximum discharge pressure for highest COP; about 88 bars in this case. The pressure is fixed, as can be seen in "Region 2", and the condenser/gas cooler fans are controlled for reduced capacity to recover more heat in the de-superheater, this results in higher condenser/gas cooler exit temperature.

The resulting total power consumption of the refrigeration system when controlled for the highest COP is plotted in Figure 15. The power consumption for the cases of heat recovery without sub-cooling and for floating condensing are also plotted. The corresponding COP_{1HR} are plotted in Figure 16.

Figure 15: Power consumption of the refrigeration system at different ambient temperatures. The cases are for heat recovery mode with no sub-cooling, system controlled for highest COP possible and for floating condensing without heat recovery.

Figure 16: $COP1_{HR}$ for the refrigeration system with heat recovery from the de-superheater. For the cases of heating system return temperatures of 30°C with no sub-cooling and for controlled for highest COP possible.

It can be observed in the plot that the booster system's COP_{1HR} increases significantly, for most of the ambient temperature range, due to the use of sub-cooling.

The power consumption of the condenser/gas cooler fans is not included in the calculations of this study. However, it is estimated to be about 4.5 kW at full capacity for the condenser/gas cooler of the analyzed system. This is based on a commercial heat exchanger data; AlfaBlue Double Row 8XL BXDS803CY-H by Alfa Laval (Alfa Laval, 2012). Including the fan power consumption at full capacity in the heat recovery mode may reduce the COP_{1HR} of the refrigeration system by about 10%. But, the fan power at partial load in the floating condensing was not included which makes the reduction in COP_{1HR} lower.

It can be observed in Figure 14 that the gas cooler exit temperature can be lower than 0°C which may cause problems of oil return and frost formation (Wulfinghoff, 1999). If the system is controlled to have minimum gas cooler exit temperature of 5°C in the heat recovery and floating condensing modes then the resulting COP_{1HR} will be 2-20% lower along the calculated range than that of the system controlled for highest COP_2 possible (the top curve in Figure 16).

The details for system simulation in this study have been related to a case of an average size supermarket in Sweden. The control strategy suggested to achieve the highest COP can be generalized for similar refrigeration systems with different sizes and in different climate zones. However, the values for the different COP's presented in this study will vary depending on the relations between the cooling and heating needs and the ambient temperatures; i.e. how much sub-cooling can be achieved at different HRR's.

6. Annual energy usage calculations

Having developed cooling and heating demand profiles for an average size supermarket and calculated CO₂ refrigeration system efficiencies at different ambient temperatures, the power

consumption of the refrigeration system to cover the cooling and heating demands at a given ambient temperature can be calculated.

A common solution to provide the space heating needs in supermarkets is to use a separate independent heat pump. For a heat pump providing water at 35°C to the heating system with heat source temperature of 5°C and assuming compressor's overall efficiency of 65% the heating COP (COP_{HP}) is 3.8. The refrigerant used for the calculation is R407C and the corresponding condensing and evaporating temperatures are 40 and -5°C respectively. Comparing this value to COP_{HR} of the refrigeration system plotted in Figure 16 it can be observed that CO₂ refrigeration system has higher heating COP for most of the operation range. Keeping in mind that COP_{HP} is an average value calculated for a rather high heat source temperature of 5°C. There are different types of heat pumps and the simplified assumptions used for the independent heat pump in this study aims at giving general guidelines of the energy usage for the heating needs in the supermarket.

The annual energy usage for an average size supermarket in Sweden in the climate of Stockholm area is calculated for different system solutions. For the CO₂ trans-critical refrigeration system running in floating condensing without heat recovery the annual energy consumption is 445MWh. If a separate conventional heat pump with an average COP_{HP} of 3.8 is used to cover the heating needs the total annual energy usage of the refrigeration system and the separate heat pump is about 565MWh.

However, for a CO₂ booster refrigeration system with heat recovery from the de-superheater the annual energy usage covering the heating and cooling demands is about 540MWh, which is about 4% lower than using a separate conventional heat pump system.

CO₂ trans-critical solutions for refrigeration systems in supermarkets are rather new alternative for conventional systems running with R404A. Figure 17 is a schematic of the most common conventional R404A system solution in supermarkets in Sweden. Brine is used in the secondary loop in indirect system solution to serve the medium temperature level

and to further sub-cool the refrigerant in the low temperature unit, which runs with direct expansion. In order to minimize the refrigerant charge in the system an indirect solution usually is used for heat removal from the condenser.

Figure 17: Schematic diagram of a conventional R404A system

Similar assumptions to what is used for the CO₂ systems have been used for the calculations of the conventional. The modeled compressors are Bitzer 4J-22.2 for the medium and 4VCS-6.2 for the low temperature levels respectively (Bitzer, 2011). Some of the main assumptions are using 50% effectiveness for the internal heat exchanger. The pumping power is assumed as 6% of the medium temperature cabinets cooling demand. 5K of approach temperature difference is assumed in all heat exchangers except for the sub-cooling heat exchanger in the low temperature unit where 2K was used. 6K logarithmic mean temperature difference is assumed in the medium temperature unit evaporator, 7K of internal superheat is used. This results in evaporation temperature of about -12°C providing brine at -

8°C to the cabinets. The resulting medium and low temperature COP2 is presented in Figure 18.

Figure 18: Medium and low temperature levels COP2 of the R404A conventional refrigeration system

When comparing the values in this figure to the values of the CO₂ system in Figure 3 it must be kept in mind that the CO₂ system rejects heat directly to the ambient via the condenser/gas cooler (Figure 2) while the conventional R404A system has a coolant loop connecting the condensers and the dry cooler (Figure 17).

The annual energy usage of such a conventional system without control for heat recovery and using a conventional heat pump to provide the necessary heat is about 575MWh which is about 2% higher than a booster CO₂ system with direct heat rejection to ambient air and a separate conventional heat pump (565MWh). When compared to CO₂ booster system with heat recovery from the de-superheater (540MWh) the conventional system with separate heat pump has 6% higher annual energy use. These are the two systems that will mostly be compared in supermarket installations in Sweden; as conventional and alternative solutions.

If the conventional system solution is assumed to reject heat directly to the ambient air then it will operate at lower condensing temperature and the total energy usage for refrigeration and heating using a separate conventional heat pump will be about 535MWh. This is slightly lower, about 1%, than the CO₂ booster system with heat recovery from the de-superheater (540MWh).

The environmental gains of using CO₂ in the refrigeration systems instead of synthetic refrigerants and the possible savings in installation cost due to the absence of the separate heat pump in the CO₂ system with de-superheater are additional factors that can contribute to favouring CO₂ systems in supermarkets refrigeration.

7. Conclusions

CO₂ trans-critical refrigeration system with heat recovery from the de-superheater was modeled. Cooling and heating COP's of the system were calculated at different ambient temperatures for floating condensing and heat recovery modes. The heating COP of the trans-critical refrigeration system ($COP_{1_{HR}}$) was defined and compared to that of a typical heat pump ($COP_{1_{HP}}$). The influence on the system's COP by sub-cooling in the condenser/gas cooler during heat recovery mode was studied.

Sub-cooling in the condenser/gas cooler in the heat recovery mode increases the system's COP; therefore, the condenser/gas cooler should be operated at full capacity in the heat recovery mode as long as the pressure is lower than the maximum value to achieve the highest COP. When the heating needs reach a high value where the maximum discharge pressure for the highest COP is reached then the pressure should not be increased and the condenser/gas cooler fans speed should be reduced to increase the recovered heat from the system. The maximum heating capacity of the refrigeration system is reached when the discharge pressure is at the maximum value for highest COP and the condenser/gas cooler is switched off, or by-passed. For the case analyzed in this study the system can provide heating energy about 1.5 times the total demand at the medium temperature level (i.e. $\dot{Q}_{m,tot}$).

The maximum discharge pressure for highest COP in heat recovery operation is dependent on the CO₂ exit temperature from the de-superheater; in this case 88 bars for 35°C exit from the de-superheater. It follows the same correlation for the optimum discharge pressure for maximum COP in a CO₂ refrigeration system.

Following the suggested control strategy in this study, the analyzed CO₂ system produces heating $COP_{1_{HR}}$ higher than a typical heat pump for almost all the calculated temperature range. The control strategy

The studied CO₂ system with heat recovery from the de-superheater has slightly lower energy usage in an average size supermarket when compared to conventional R404A refrigeration system with a separate heat pump for heating needs.

CO₂ trans-critical systems for supermarkets refrigeration are efficient solutions in cold climates. Recovering heating energy from the de-superheater following the proper control strategy makes the solutions also efficient for heat recovery, which is especially important in cold climates and makes the CO₂ systems more favorable.

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