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Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part I: Modeling, simulation and optimization of two system solutions

*Samer Sawalha**

Department of Energy Technology, The Royal Institute of Technology (KTH), Brinellvägen 68, SE-10044 Stockholm, Sweden

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ABSTRACT

Using CO_2 trans-critical system solutions in supermarket refrigeration is gaining interest with several installations already running in different European countries. Using a computer simulation model, this study investigates the performance of two main system solutions: centralized with accumulation tank at the medium temperature level and parallel with two separate circuits for low and medium temperature levels. Both system solutions are presented and the simulation model is described in details. Calculations have been performed to design the systems and optimize their performances where basic layout and size of each solution have been defined. For ambient temperature range of 10–40 °C, the reference centralized system solution shows higher COP of about 4–21% than the reference parallel solution. Using two-stage compression in the centralized system solution instead of single stage will result in total COP which is about 5–22% higher than that of the reference centralized system and 13–17% higher than that of the improved two-stage parallel system. The two-stage centralized system solution gives the highest COP for the selected ambient temperature range.

Evaluation théorique des systèmes frigorifiques de supermarché au CO_2 transcritique. Partie I : modélisation, simulation et optimisation de deux solutions

Mots clés : Système frigorifique ; Supermarché ; Système à compression ; Dioxyde de carbone ; Cycle transcritique ; Système biétagé ; Réservoir ; Modélisation ; Simulation ; Optimisation

* *Tel.*: +46 (0)8 790 7889; fax: +46 (0)8 204 161.
E-mail address: samer@energy.kth.se

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Nomenclature

C	heat capacity rate ($W K^{-1}$)
COP	coefficient of performance
CR	circulation ratio
dT	temperature difference (°C)
DX	direct expansion
IHE	internal heat exchanger
P_1	condensing pressure (bar)
P_2	evaporating pressure (bar)
T	temperature (°C)
T_1	condensing temperature/gas cooler exit temperature (°C)
T_2	evaporating temperature (°C)
x	vapor quality
ε	heat exchanger effectiveness
η	efficiency

Subscript

air	for air
amb	ambient
app	approach temperature
c	cold fluid
Equal,PR	equal pressure ratios
gc	gas cooler
h	hot fluid
i	in
inter	intercooler
is	isentropic
L	low temperature/freezer
min	minimum
o	out
opt	optimum
product	for product
product,air	temperature difference between product and air

1. Introduction

The three main solutions where CO_2 is applied in supermarket refrigeration are the indirect, cascade and the trans-critical systems. The indirect system solution with CO_2 as secondary fluid is used for low temperature applications where pressure levels are reasonably low and conventional components can be used. In recent years, components for CO_2 became more available which paved the way to install cascade and trans-critical systems in supermarkets. The cascade arrangement implies that a temperature difference will exist in the cascade condenser, which decreases the evaporating temperature on the high stage and reduces its COP. An efficient trans-critical CO_2 system will by pass the need for the cascade condenser, which may improve the COP. In order to evaluate CO_2 trans-critical system solution against other alternatives, such an efficient CO_2 system should be defined.

It is not practically hard to measure the performance of a field installation but it is difficult to compare two real field installations since operating parameters, system requirements and ambient conditions are not usually identical. Therefore, a computer simulation model seems to be a convenient tool as a first step in the direction of evaluating the trans-critical solution.

The two main possible system solutions where CO_2 can be used in supermarket applications are the parallel and centralized arrangements. As can be seen in Fig. 1, the parallel solution consists of two separate circuits: one serves the medium temperature level cabinets and the other serves the freezers. Direct expansion (DX) is applied on both temperature levels and two-stage compression is used for the low temperature circuit. This will decrease the discharge temperature, minimize losses in the compression work, and reduce the enthalpy difference across the compressors. Since the temperature lift is presumed to be small on the medium temperature circuit, single stage compression is used. System with similar solution has been presented by Girotto et al. (2003).

In the centralized system solution, Fig. 2, the three circuits in the system merge in the accumulator/tank. Thereby, in this solution the medium temperature cabinets' evaporators are flooded with CO_2 which is circulated by a pump while DX is used in the freezers. Compression on the high stage is done in one stage. Similar system has been discussed and investigated by Schiesaro and Kruse (2002).

The parallel system seems to be more applicable mainly due to its similarity to conventional DX systems. The fact that there are two separate circuits makes it more convenient

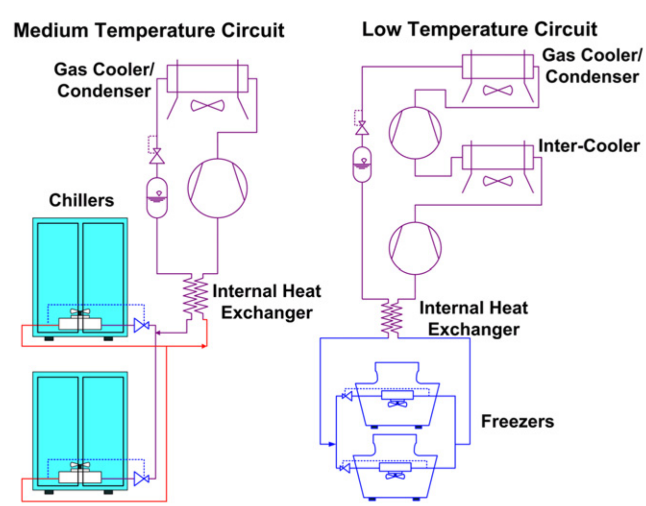

Fig. 1 – Parallel system solution with single stage compression on the medium temperature level with internal heat exchanger (IHE) on both circuits. Similar solution can be found in Girotto et al. (2003).

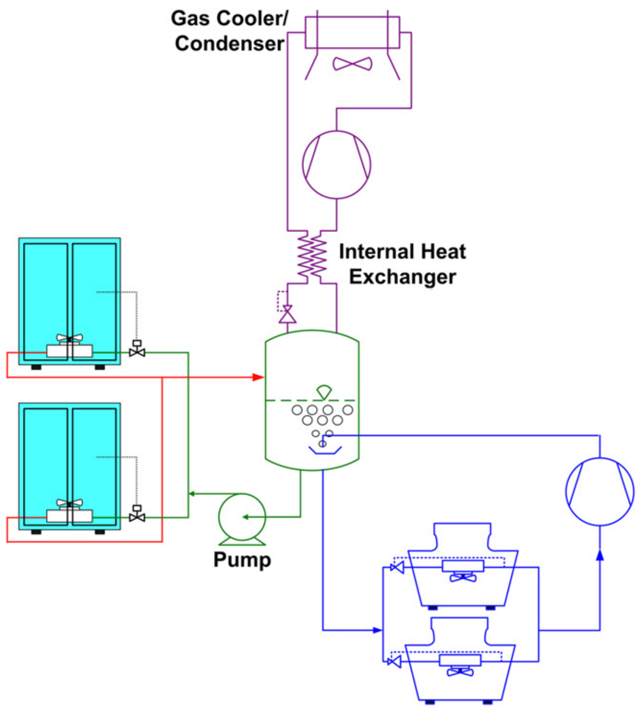

Fig. 2 – Centralized system solution with single stage compression at the high stage. Similar solution can be found in Schiesaro and Kruse (2002).

to shut down one circuit if failure occurs, while keeping the other circuit unaffected. An important practical difference between the two systems is the oil return mechanism; in the parallel system a proper sizing of suction lines and the evaporator pipes in addition to an efficient oil separator should secure sufficient oil return to the compressors. In the centralized system, solution oil escaping from the oil separator will "dissolve" in the tank liquid or float on the surface, or partly dissolve and float, depending on the oil type and its miscibility characteristics in CO_2. Therefore, an additional oil separation loop should be added to the system to remove oil from the tank and return it to the high stage compressors.

2. Simulation model

2.1. Description

The model is written in Engineering Equations Solver (EES) and uses the required products' temperatures and the ambient conditions as the boundaries of the systems. At the medium temperature level the cooling capacity is 150 kW and 50 kW for the freezing load; these capacities are typical for a medium size supermarket in Sweden. Product temperature at the medium level is +3 °C and −18 °C for the frozen food. The design condensing temperature is 30 °C. Evaporating temperatures are calculated to provide the required product temperature. Some of the parameters inserted in the model have been selected from an experimental work on an NH_3–CO_2 cascade system, detailed description of this system can be found in Sawalha et al. (2005).

In order to calculate performances, losses and capacities within the system, the main system components are modeled as follows.

2.1.1. Display cabinets

Approach temperatures and the temperature change of air across the heat exchangers were used to calculate the evaporation temperature of CO_2.

Heat transfer between the refrigerant side and the air was not modeled. Instead, approach temperatures and air temperature difference were obtained from experimental data for the medium temperature and freezing cabinets of an existing installation described by Perales Cabrejas (2006). The temperature profiles in medium temperature cabinet and DX freezer are shown in Fig. 3a and b, respectively.

The product temperatures were used as the target values; a first guess of the product temperature is inserted to obtain the temperature profile in the cabinet; the resulting evaporating temperature value is used to calculate pressure and temperature drops on the refrigerant side. When the temperature drop is included in the temperature profile a new product temperature will be calculated based on the evaporating temperature. Thereafter, according to the deviation in the product temperature, the evaporation temperature is then changed, so the product temperature is as close as possible to the target value.

2.1.1.1. Medium temperature cabinets.

In NH_3–CO_2 cascade system experimental rig presented by Perales Cabrejas (2006), temperatures of products at different positions in the cabinets have been measured in order to locate the position with the highest product temperature. The cabinets are of 5 kW cooling capacity each, with two parallel 72 m length circuits with 5/8″ (15.9 mm) diameter.

Product–air temperature difference ($dT_{product,air}$) is the difference between the warmest product temperature and the

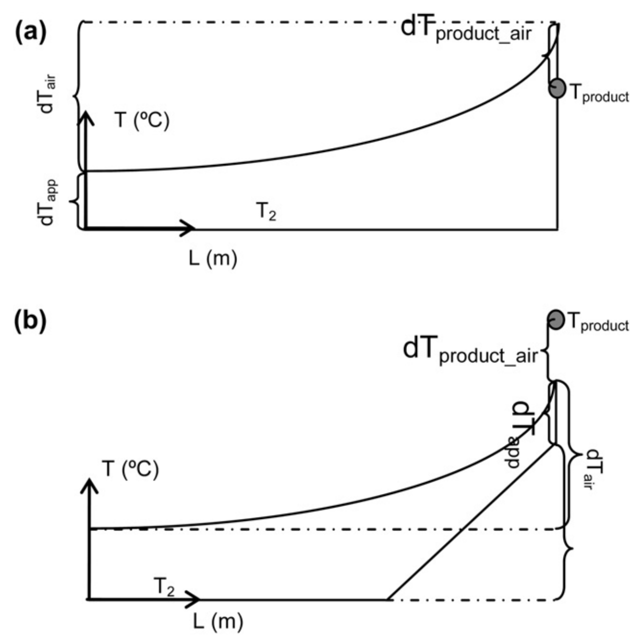

Fig. 3 – Temperature profile in (a) medium temperature cabinet and (b) DX freezers.

air inlet temperature to the display cabinet's heat exchanger. Experiments showed that the product temperature value falls between the air inlet and outlet temperatures of the display cabinet's heat exchanger; it is about 3 °C lower than the air inlet temperature. This is due to the fact that in the vertical display cabinets used in the experimental rig, an air curtain was implemented which limits infiltration of warm air from the surrounding into the cabinet's cold envelope. The air entering the evaporator is a mixture of cold air from the products envelope and warm air from the curtain. Moreover, cold air leaving the cabinet's evaporator is partly supplied from the vertical rear of the cabinet through small holes, which make the products closer to the cold air supply. The air temperature change (dT_{air}) is 7 °C and the temperature difference between the air and the refrigerant (dT_{app}) is 2 °C.

2.1.1.2. DX freezing cabinet. The freezing load is divided over 2.5 kW cabinets which have a single coil of 72 m length and diameter of 1/2″ (12.7 mm). Cabinets with such configurations were used in the laboratory cascade test rig presented by Perales Cabrejas (2006).

Experiments showed that the warmest product temperature is 2.5 °C higher than the air inlet temperature to the freezer's evaporator. A reason for the product temperature being higher than the air inlet temperature could be that freezers were exposed to large radiation surface and ceiling lightning. Air temperature change across the heat exchanger is 7.5 °C.

The lowest approach temperature between air and the refrigerant is about 5 °C; in the simulation model this could happen at the air inlet or exit ends of the heat exchanger and depends on the superheat value. Looking at the temperature profile in Fig. 3b, it can be observed that with high superheat, approach will be at the air inlet side while for low superheat value this will take place at the air exit end; similar to the case of the flooded medium temperature cabinets. As Sawalha et al. (2006) reported, the lowest superheat value that could be reached in the cascade experiment with stable operation of the freezers is 9 °C, this value has been used as an input to the simulation model.

2.1.2. *Internal heat exchanger (IHE)*

The IHE is modeled using the effectiveness as the input variable which is calculated as:

$$\varepsilon = \frac{C_c(T_{c,o} - T_{c,i})}{C_{min}(T_{h,i} - T_{c,i})} \tag{1}$$

where in this case $C_c = C_{min}$. The inlet conditions of the heat exchanger are used as the input parameters where the exit temperature from the cold side is calculated. Using temperature and pressure at the exit of the cold side of the IHE the enthalpy is calculated and the enthalpy difference at the cold side is assumed to be the same as at the hot side; enthalpy and pressure are then used to calculate the temperature at the exit of the hot side.

2.1.3. *Other heat exchangers*

Condenser/gas cooler and intercooler are simulated by assuming 5 °C approach temperature at the inlet of the heat sink side. No sub-cooling is assumed in the condensers.

2.1.4. *Pump*

The efficiency of the pump is assumed to be 50% with 400 W heat losses.

2.1.5. *Compressors*

Isentropic efficiency values of compressors were obtained from Eq. (2) which is a curve fit by Brown et al. (2002) for CO_2 compressor in mobile air conditioning system. This correlation has been compared by Chen and Gu (2005) to two other compressor curve fits, Robinson and Groll's (1998) and Liao et al.'s (2000), and was adapted as a good approximation for the real performance of an open type CO_2 compressor.

$$\eta_{is} = 0.9343 - 0.04478\left(\frac{P_1}{P_2}\right) \tag{2}$$

P_1 and P_2 are the discharge and suction pressures.

It must be pointed out that Brown et al. (2002) suggested this formula for pressure ratios higher than 2. However, applying this correlation for lower pressure ratio values gives results which are close to the ones generated by the correlations of Robinson and Groll (1998) and Liao et al. (2000). The correlation presented by Liao et al. (2000) is a curve fit of experimental data of a CO_2 compressor and has been used in calculations for pressure ratios close to 1.5. Other examples of tested compressors with good isentropic efficiency at pressure ratios lower than 2 can be found in Cutler et al. (2000) and Giannavola et al. (2000).

2.1.6. *Pressure and temperature drop calculations*

Two phase flow pressure drop calculations use Friedel's correlation which is based on the two phase multiplier. The Clapeyron equation is used to convert the pressure drop to an equivalent change in saturation temperature. Single phase pressure drop is calculated by using the Gnielinsky correlation for the friction factor in turbulent flow. More details about pressure and temperature drop calculations used in this model can be found in Sawalha and Palm (2003).

No fittings' pressure drop is included in the calculations. Pipe length could be increased by 50% to account for the pressure drop in fittings, as suggested by Dossat (1991). However, this was not applied in the calculations because the model used to calculate the two phase pressure drop over-estimates the actual pressure drop by about 50% in average, according to Shahzad (2006).

The flow in the medium temperature display cabinet starts as single phase flow and evaporates along the heat exchanger pipe and exits with certain quality that is the inverse value of the circulation ratio ($x = (1/CR)$). The pressure drop calculations were done by simplified algorithm, which was shown to give results close to those of Friedel's correlation presented in Hewitt (1998). The same method is used for calculating the pressure drop in the DX freezers with two phase flow conditions at the inlet and single phase flow conditions at the exit. The simplified assumption is used due to the expected small pressure and temperature drops in the display cabinets, less than 0.5 °C, which will have insignificant influence on the system's COP.

Pressure drop is neglected in the rest of the heat exchangers in the system and in the high stage circuit pipes.

3. Optimization of the system

3.1. *Pipe lines' sizing*

Pipe length between the machine room and the display cabinets is assumed to be 60 m, which is an estimate for an average size supermarket.

The size of each pipe was chosen in a way to have reasonable pressure drop in liquid supply lines. Dossat (1991) recommended that 70 kPa should not be exceeded in direct expansion systems. The suction and return lines were sized according to the guideline that 0.5–2 °C of temperature drop in the suction line in a direct expansion system is acceptable, according to Stoecker (1998). The same guidelines were followed in the sizing of the medium temperature distribution lines. If high pressure drop will be allowed in the medium temperature level, the pump power will be high and the resulting head will increase the temperature in the cabinets. The pipes were assumed to be well insulated and no heat transfer with the surrounding was accounted for.

Tube size selection took into consideration the possibility of running the system within a reasonable margin of load fluctuations and boundary changes. Also some parameters, e.g. circulation ratio, can be changed while the system still gives reasonable pressure drops with the selected sizes.

Fig. 4 shows the temperature and pressure drops in the supply and the return lines in the medium temperature circuit. The calculations are made for different tube diameters and different circulation ratios at saturation temperature of −8 °C. Value of 2 is used as the design circulation ratio at the rated capacity.

As can be seen from the figure, 1 3/8″ (34.9 mm) supply line will result in temperature and pressure drops of about 0.35 °C and 30 kPa, respectively. On the return line, the temperature drop in 1 5/8″ (41.3 mm) is 0.9 °C which corresponds to a pressure drop of about 75 kPa. In the same figure, another plot is made where the circulation rate is increased to have a circulation ratio of 3 and the results on the temperature drop side show that the values are still within the acceptable range.

On the low temperature side, the plot in Fig. 5 shows the calculation for two solutions where direct expansion or flooded

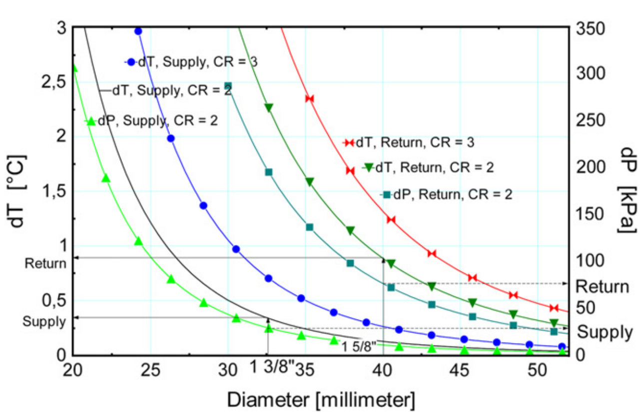

Fig. 4 – Temperature and pressure drops in the medium temperature distribution lines.

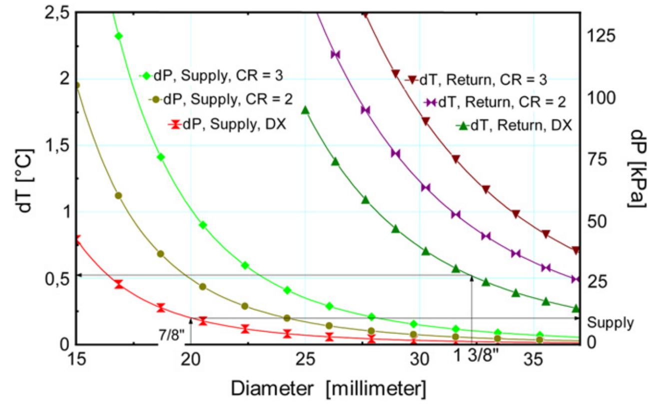

Fig. 5 – Temperature and pressure drops in the low temperature distribution lines.

evaporators can be applied. As can be seen in the figure, for DX solution or flooded evaporator with circulation ratios of 2 and 3, the temperature and pressure drops are within the acceptable level for 7/8″ (22.2 mm) on the liquid line and 1 3/8″ (34.9 mm) for the return line.

3.2. *Optimum high pressure for trans-critical mode*

At high ambient temperatures and when the exit temperature of CO_2 in the gas cooler gets higher than the critical temperature of CO_2 (31 °C), then the operating pressure becomes independent of the gas cooler exit temperature. As can be seen in Fig. 6, different pressures can be selected and it is clear from the plot that there is an optimum pressure to achieve the highest COP. For different temperatures, the isotherm shape will change and this suggests that the optimum operating pressure will depend on the ambient temperature.

In order to obtain the optimum discharge pressure for each ambient temperature, the plot in Fig. 7 is generated where the discharge pressure is varied and the COP is plotted

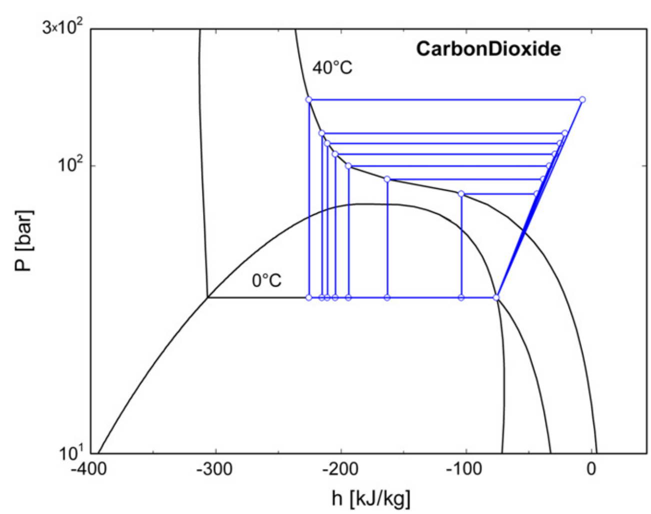

Fig. 6 – CO_2 trans-critical cycle with gas cooler exit temperature of 40 °C and different discharge pressures.

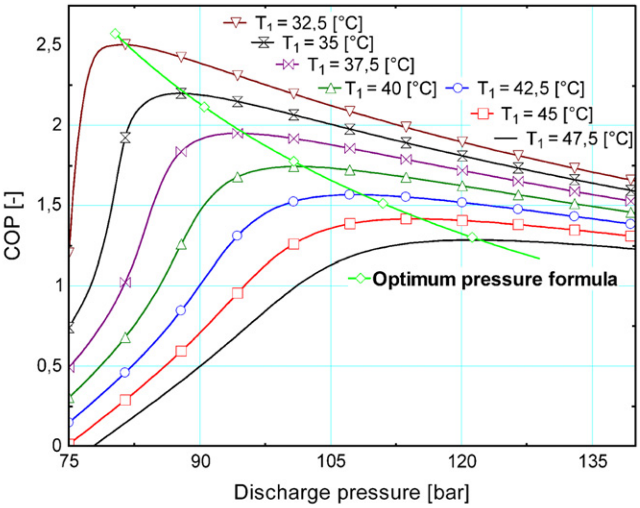

Fig. 7 – COP of CO_2 trans-critical cycle vs. discharge pressure at different gas cooler exit temperatures (denoted T_1).

against the pressure for different ambient temperatures (related to gas cooler exit temperature). The optimum values for the high pressure were curve fitted as a function of the ambient temperature.

The approach temperature in the gas cooler was assumed to be 5 °C and constant for all the range of ambient conditions in sub- and trans-critical operations. The approach was assumed to take place at the air inlet side. Chen and Gu (2005) used Brown et al.'s (2002) curve fit of the gas cooler exit temperature for different ambient conditions; as can be seen in the plot in Fig. 8, the value of the approach temperature ranges around 5 °C which means that it is a reasonable approximation. Kauf (1998) used a fixed value of 2.9 °C while Liao et al. (2000) used the gas cooler exit temperature, instead of the ambient temperature, as the input value for the optimum discharge pressure correlation.

The evaporating temperature used in the calculations was −8 °C; according to Kauf (1998) and Chen and Gu (2005), the evaporating temperature has insignificant influence on the optimum discharge pressure. However, Liao et al. (2000) included the influence of the evaporating temperature in the optimum discharge pressure correlation.

An important parameter which influences the optimum discharge pressure is the isentropic efficiency of the compressor. Chen and Gu (2005) used Brown et al.'s (2002) correlation while Kauf (1998) and Liao et al. (2000) used their own curve fit of real compressors.

Curve fitting the optimum pressure values developed in Fig. 7 in relation to the ambient temperature yields the following correlation, which has been used in the simulation model:

$$P_{opt} = 2.7(T_{amb} + T_{gc,app}) - 6.1 \tag{3}$$

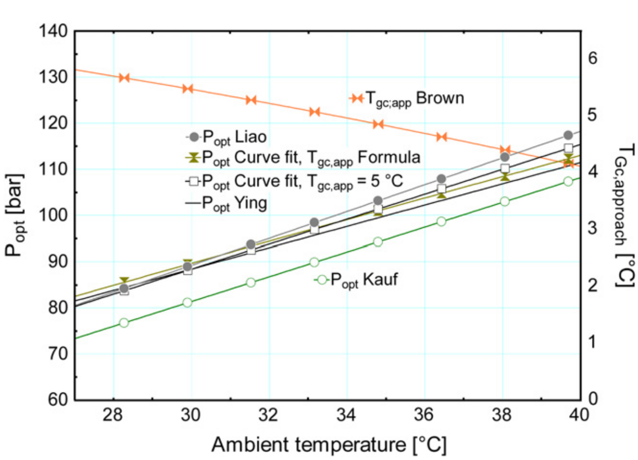

Fig. 8 – Optimum discharge pressure and the gas cooler approach temperature for CO_2 trans-critical cycle at different ambient temperatures.

Fig. 8 is a comparison between different correlations to calculate the optimum discharge pressure. The line labeled "P_{opt} Curve fit, $T_{gc,app}$ Formula" is the curve fit developed from a similar plot in Fig. 7 but using Brown et al.'s (2002) correlation for the approach temperature estimate instead of the constant 5 °C. In Liao et al.'s (2000) correlation the gas cooler exit temperature was related to the ambient temperature by using 5 °C fixed approach temperature. As can be seen in the plot, except for Kauf's (1998) which uses lower approach temperature, the optimum pressure correlations are in good agreement and the lines slightly diverge at high ambient temperatures, more than 36 °C.

It is worth noting that most of the studies concerning the optimum CO_2 discharge pressure are performed in the mobile air conditioning and heat pumps applications. This implies that the capacities and components' sizes, to which the correlations have been generated, are much smaller than those for commercial application. However, the assumptions that have been used to develop the correlation in Eq. (3) are relevant for supermarket applications and the compressor efficiencies do not seem unrealistic; it ranges slightly around 80% within the operation range.

3.3. *Optimum intermediate pressure*

When CO_2 operates in the trans-critical region it looses in the COP compared to other refrigerants at the same temperature levels. A way to improve the COP, especially when high temperature lift is needed, is to introduce two-stage compression with intercooler. In general, the optimum intermediate pressure depends on the shape of the isotherm at which the heat is rejected, the slope of the isentropic compression lines and the dependence of the isentropic efficiency on the pressure ratio. In case of CO_2, the high operating pressure results in relatively low-pressure ratios, hence, the variations in the isentropic efficiency are small. Using Brown et al.'s (2002) correlation, the isentropic efficiency is 84% at pressure ratio of 2 and 76% at pressure ratio of 4.

3.3.1. *Medium temperature level*

For evaporating temperature of −8 °C and different ambient temperatures, the COP at different intermediate pressures (pressures in-between the two compressor stages) are calculated and plotted in Fig. 9. IHE with 50% effectiveness was used.

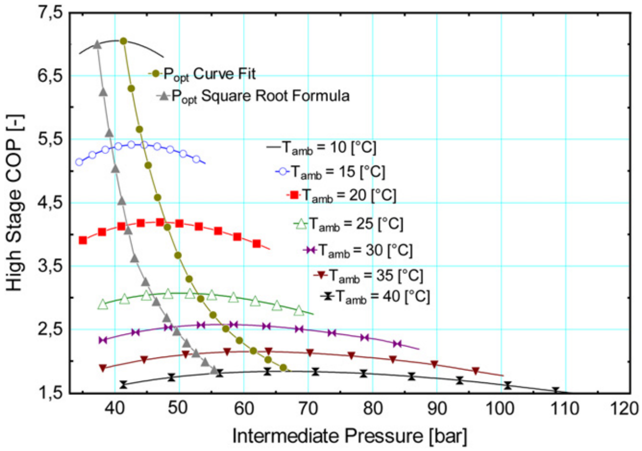

Fig. 9 – COP of high stage vs. intermediate pressure (pressure between high stage compressor stages) at different ambient temperatures. IHE effectiveness is 50% and evaporating temperature is −8 °C.

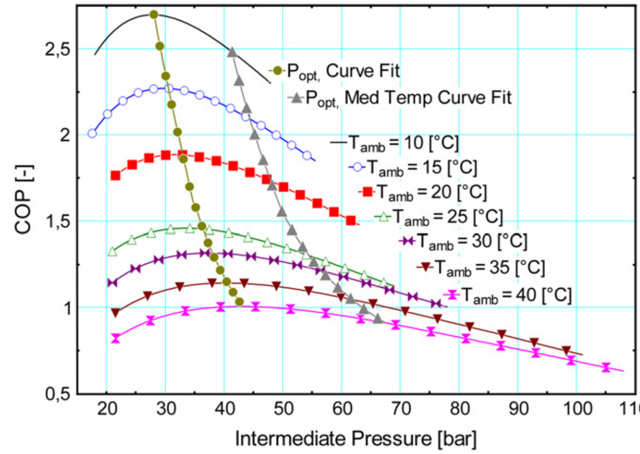

Fig. 10 – COP vs. intermediate pressure (pressure between compressor stages) at different ambient temperatures. IHE effectiveness is 50% and evaporating temperature is −37 °C.

The values of the optimum intermediate pressure for different ambient temperatures were curve fitted and the following correlation was obtained:

$$P_{inter,opt} = 0.01(T_{amb} + T_{inter,app})^2 + 0.23(T_{amb} + T_{inter,app}) + 35.53 \quad (4)$$

The intercooler's approach temperature was assumed to be 5 °C for all ambient temperature levels. Eq. (4) is plotted in Fig. 9 and labeled "P_{opt} Curve Fit" where it can be seen that it passes very near to the optimum pressure at the selected ambient temperatures. The plot "P_{opt} Square Root Formula" uses the following expression to determine the intermediate pressure:

$$P_{EqualPR} = (P_2 \times P_1)^{0.5} \quad (5)$$

This formula results in equal pressure ratios on both compressors and is suggested in Granryd (1999) as a good approximate for the optimum pressure. As can be seen in Fig. 9, the plot of Eq. (5) falls below the optimum pressure.

3.3.2. *Low temperature level*

The same method has been used to obtain a correlation for the optimum intermediate pressure for an evaporating temperature of −37 °C; this applies to the low temperature circuit in Fig. 1. Fig. 10 is a plot of the COP for different ambient temperatures and intermediate pressure values. The denoted "P_{opt} Med Temp Curve Fit" is the plot using the correlation in Eq. (4). As can be seen in the plot, the correlation is not valid for freezing temperature and new correlation should be used. The correlation in Eq. (6) is used where Fig. 10 shows that the curve fits with the optimum pressure values:

$$P_{inter,opt,L} = 0.0008(T_{amb} + T_{inter,app})^2 + 0.4552(T_{amb} + T_{inter,app}) + 21.1 \quad (6)$$

3.4. *Circulation ratio*

According to the experimental study on the optimum circulation ratio for CO_2 in flooded evaporator presented by Sawalha et al. (2006), increasing the circulation ratio resulted in an increase in pressure drop across the heat exchanger with no improvement on the heat transfer. This indicates that the circulation ratio should be chosen as low as possible to ensure complete evaporation at the highest load expected. Therefore, circulation ratio value of 2 was chosen which should ensure that at load fluctuations and at start up, dry evaporation will not occur.

4. Results and discussions

4.1. *Parallel system*

The performance of the parallel system is evaluated by calculating the COP of the low and the medium temperature circuits. The reference case is defined to have two-stage compression at the low temperature level and single stage

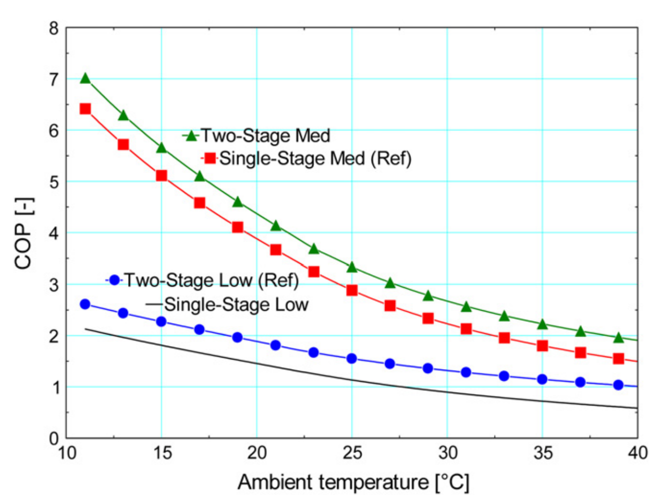

Fig. 11 – Two- and single stage compression arrangements' COPs for medium- and low temperature circuits in the parallel CO_2 system.

compression for the medium temperature circuit, the system in Fig. 1. The COP is calculated for different ambient conditions and plotted in Fig. 11. The improvement of COP when two-stage compression is used in both stages compared to the single stage compression can be seen in the plot; along the calculated temperature range the improvement on medium stage COP varies between 9% and 27%. The improvement at this circuit is important since the load is usually higher at the medium temperature level; therefore, energy consumption savings will be more prominent.

The two-stage compression at the low temperature level will result in significant improvements on the COP; 22–72% higher COP values than for the single stage compression over the selected ambient temperature range is calculated. In both cases the COP improvement is higher at high ambient conditions.

The total COP of the system is the ratio between the total cooling capacity and the total energy consumption. Total COP values are plotted in Fig. 12 for different solutions of the parallel system arrangement. When two-stage compression is used in both circuits, the total COP becomes 4–16% higher than the reference case and 15–45% higher than the case of single stage compression on both circuits.

4.2. Centralized system

The COP calculation for the centralized system solution is done in the same way as for the parallel arrangement. The pump power on the medium temperature circuit is included in the system's energy consumption, which is usually very small compared to the total energy consumption; in this case 740 W were consumed to run the pump for circulation ratio of 2. The reference centralized system solution is defined to have single stage compression at the high stage, system in Fig. 2.

In Fig. 13, the total COP is plotted for the reference centralized system solution, which shows 4–21% better COP compared to the reference parallel system. The improved parallel system solution with two-stage compression in both circuits gets close to the reference centralized system solution and at ambient temperatures higher than 30 °C it gives up to 6% better COP. Modifying the centralized reference system to two-stage compression on the high stage results in 13–17% increase in COP over the improved parallel system.

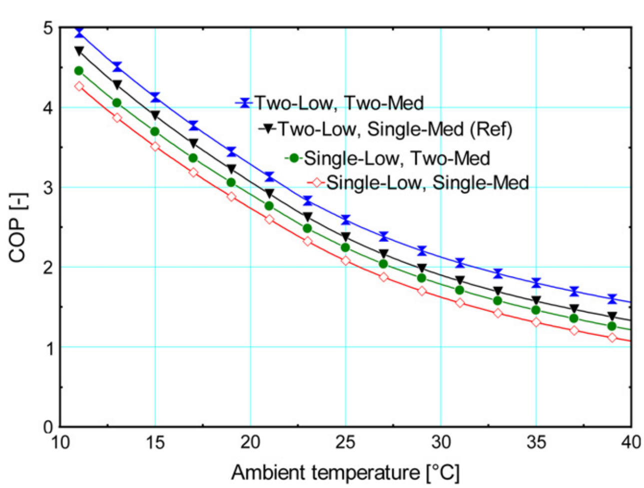

Fig. 12 – Total COPs for the parallel CO_2 system with the possible combinations of single- and two-stage compression arrangements at the two temperature levels.

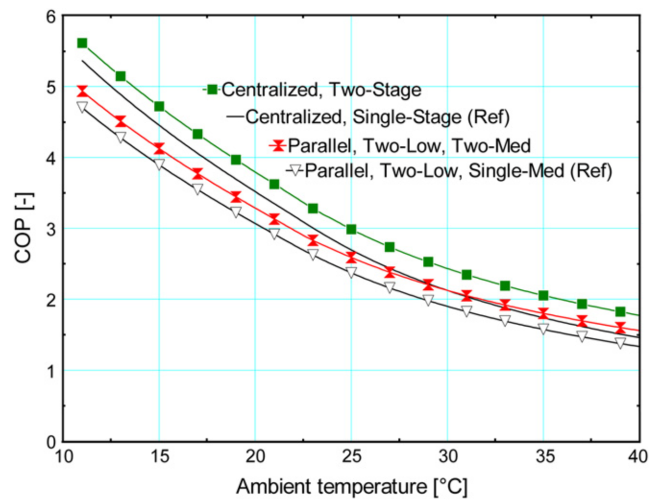

Fig. 13 – Total COPs for the centralized and parallel CO_2 systems for reference arrangements and for two-stage compression cases.

The generated COP values at low ambient temperatures in floating condensing and for the two-stage compression cases are based on the assumption that the compressor operates with isentropic efficiency according to Brown et al.'s (2002) correlation for pressure ratios down to 1.2. The pressure ratio in the simulations ranged from 1.7 to 3.7 in the case of single stage centralized and from 1.2 to 2.2 in the case of two-stage centralized.

5. Conclusions

A computer model has been designed which simulates the performance of a supermarket refrigeration system with CO_2 trans-critical cycle. Two possible solutions, parallel and centralized, for the trans-critical system have been defined. The simulation model is used to size the system components and optimize its performance. Correlations for the optimum gas cooler pressure and intermediate pressure between compressor stages in low- and high-pressure stages have been developed. The performance of the optimized system solutions has been performed and analyzed.

Over the ambient temperature range of 10–40 °C, the reference centralized system solution shows 4–21% better COP than the reference parallel system. The improved parallel system with two-stage compression at the medium temperature results in lower COP than the centralized reference one at ambient temperatures lower than 30 °C, and slightly better COP at higher temperatures. Two-stage compression with intercooler must be applied in the circuit for freezing products in the parallel system in order to improve the COP; 22–72% improvement can be achieved.

Using two-stage compression at the high stage of the centralized system solution improves the COP by 5–22% over the reference centralized system and 13–17% over the improved two-stage parallel system. It is essential for the application of this solution with floating condensing to have a compressor with good isentropic efficiencies at the low pressure ratios required in this case. Hence, the two-stage compression on the high stage of centralized system will result in the best COP among the discussed solutions.

REFERENCES

Brown, J.S., Yana-Motta, S.F., Domanski, P.A., 2002. Comparative analysis of an automotive air conditioning systems operating with CO_2 and R134a. Int. J. Refrigeration 25 (1), 19–32.

Chen, Y., Gu, J., 2005. The optimum high pressure for CO_2 transcritical refrigeration systems with internal heat exchangers. Int. J. Refrigeration 28 (8), 1238–1249.

Cutler, B., et al., 2000. Development of a trans-critical carbon dioxide environmental control unit. In: 4th IIR Gustav Lorentzen Conference on Natural Working Fluids, Indiana, USA.

Dossat, R.J., 1991. Principles of Refrigeration, third ed. Prentice Hall, cop., Englewood Cliffs, NJ.

Giannavola, M.S., et al., 2000. Experimental investigations of an automotive heat pump prototype for military, SUV and compact cars. In: 4th IIR Gustav Lorentzen Conference on Natural Working Fluids, Indiana, USA.

Girotto, S., Minetto, S., Nåksa, P., 2003. Commercial refrigeration system with CO_2 as refrigerant experimental results. In: 21st IIR International Congress of Refrigeration, Washington, D.C., USA.

Granryd, E., et al., 1999. Refrigeration Engineering. Department of Energy Engineering, Division of Applied Thermodynamics and Refrigeration, Royal Institute of Technology (KTH), Stockholm, Sweden.

Hewitt, G.F. (Ed.), 1998, Heat Exchanger Design Handbook, vol. 2. Begell House, Inc, NY.

Kauf, F., 1998. Determination of the optimum high pressure for transcritical CO_2-refrigeration cycles. Int. J. Therm. Sci. 38 (4), 325–330.

Liao, S.M., Zhao, T.S., Jakobsen, A., 2000. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles. Appl. Therm. Eng. 20 (9), 831–841.

Perales Cabrejas, C., 2006. Parametric Evaluation of NH_3/CO_2 Cascade System for Supermarket Refrigeration in Laboratory Environment. Department of Energy Technology, The Royal Institute of Technology (KTH), Stockholm.

Robinson, D.M., Groll, E.A., 1998. Efficiencies of transcritical CO_2 cycle with and without an expansion turbine. Int. J. Refrigeration 21 (7), 577–589.

Sawalha, S. Palm, B., 2003. Energy consumption evaluation of indirect systems with CO_2 as secondary refrigerant in supermarket refrigeration. In: 21st IIR International Congress of Refrigeration, Washington, D.C., USA.

Sawalha, S., Rogstam, J., Nilsson, P.-O., 2005. Laboratory tests of NH_3/CO_2 cascade system for supermarket refrigeration. In: IIR International Conference on Commercial Refrigeration, Vicenza, Italy.

Sawalha, S., Soleimani, K.A., Rogstam, J., 2006. Experimental and theoretical evaluation of NH_3/CO_2 cascade system for supermarket refrigeration in laboratory environment. In: 7th IIR Gustav Lorentzen Conference on Natural Working Fluids, Trondheim, Norway.

Schiesaro, P., Kruse, H., 2002. Development of a two stage CO_2 supermarket system. In: IIF-IIR, New Technologies in Commercial Refrigeration, Urbana, Illinois, USA.

Shahzad, K., 2006. An Ice Rink Refrigeration System Based on CO_2 as Secondary Fluid in Copper Tubes. Energy Technology Department, Royal Institute of Technology (KTH), Stockholm, p. 80.

Stoecker, W.F., 1998. Industrial Refrigeration Handbook. McGraw-Hill, p. 782.