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Theoretical evaluation of trans-critical CO₂ systems in supermarket refrigeration. Part II: System modifications and comparisons of different solutions

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ABSTRACT

The performance of CO₂ refrigeration systems strongly depends on the operating conditions. The specific characteristics of low critical temperature and high operating pressure limit its applications and imply the implementation of different control strategies. This study compares the performance of different CO₂ system solutions for supermarket refrigeration with R404A system. Some possible modifications and improvements on the CO₂ system have been investigated. The COP of the investigated CO₂ system solution can be improved by about 3–7% along the ambient temperature range of 10–40 °C. The annual energy consumption calculations in three different climates; cold, moderate and hot, show that the centralized trans-critical CO₂ system is good solution for cold climates whereas the NH₃–CO₂ cascade system has the lowest energy consumption in hot climates. Both systems proved to be good alternatives to R404A DX system for supermarket refrigeration.

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Evaluation théorique des systèmes frigorifiques de supermarché au CO₂ transcritique. Partie II : modifications des systèmes et comparaison de plusieurs systèmes

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COP	coefficient of performance
dT	temperature difference, °C
DX	direct expansion
IHE	internal heat exchanger
h	enthalpy
P	pressure, bar
s	entropy
T	temperature, °C
T ₂	evaporating temperature, °C
ε	heat exchanger effectiveness

air	for air
amb	ambient
app	approach temperature
opt	optimum
product	for product
product, air	difference between product and air

The main characteristics that make CO₂ unique compared to other refrigerants are the high operating pressure (64 bar at 25 °C) and low critical temperature of 31 °C; this implies that at relatively low ambient temperatures the cycle will have to operate in the trans-critical region with high pressures. Despite operating the system at the optimum discharge pressure trans-critical operation will result in loss in COP compared to other refrigerants; therefore, means to improve system's COP and to avoid operating the system close or higher than the critical temperature must be implemented.

Different system solutions can be used with CO₂ in supermarket refrigeration; in part I of this study the centralized and parallel system solutions have been analyzed. In the centralized system solution the low pressure (freezing), high pressure, and the medium temperature circuits merge in the accumulator/tank, schematic of this system solution is shown in Fig. 1. The parallel solution consists of two separate direct expansion (DX) circuits; one serves the medium temperature level cabinets and the other serves the freezers.

In part I of this study the centralized system was found to be better solution than the parallel system. In order to obtain the best performance from this system, proper evaluation of the influence of some components on its performance should be implemented, thus system optimization can be achieved. Some of the parameters that should be evaluated are the influence of using internal heat exchangers (IHE), single and two-stage

Fig. 1 – CO₂ centralized system solution with two- or single-stage compression at the high stage.

In order to assess the performance of trans-critical CO₂ system concept it should be evaluated against different applicable alternatives. Two parallel DX circuits for medium and low temperature levels with R404A as the refrigerant is referred to as the conventional system solution. An alternative solution that uses natural refrigerants is a cascade system with CO₂ at the low and medium temperature levels with NH₃ or propane at the high stage. The CO₂ trans-critical parallel system solution, presented in part I of this study, should also be included in the comparison. It is not the best system solution for CO₂ from energy consumption point of view but it may have some technical and economical advantages over the centralized solution, in some cases these could be stronger factors in the choice of the system to be installed.

Some of the improvements that can be applied to the trans-critical CO₂ system could be also implemented on conventional system or other alternative solutions. However, the key advantages of using CO₂ which are mainly related to the global environment and occupants' safety make it viable to invest in improving this technology. Higher installation cost is usually attached to the fact that the technology is being in the development phase and new system solutions must be implemented where unconventional components may be used. In order to help offsetting the investment cost; energy consumption calculations should be performed to demonstrate that savings

in the running cost should be expected which would shorten the pay off period. Experimental results that verify advantages highlighted in the simulations should encourage implementing the trans-critical CO₂ systems. However, if there would be a deviation between the experimental results and the simulation model then the reasons for differences should be investigated and improvements on the implementation of the theory should be applied.

2. Optimization of the centralized trans-critical CO₂ system

2.1. Reference CO₂ system solution

According to the analysis in part I of this study; the centralized CO₂ system with two-stage compression on the high stage produces higher COP than the single-stage centralized and the parallel system solutions. Fig. 1 is a schematic diagram of two- and single-stage centralized system solutions.

The two-stage centralized solution is referred to as the reference solution to which modifications are applied and performance is related. The effectiveness of the high stage IHE is 50% and the ambient temperature is 30 °C with 9 °C superheat in the DX freezers. Cooling capacity at the medium temperature level is 150 kW at –4 °C and in the freezers it is 50 kW at –34.5 °C. These evaporating temperatures will insure product temperatures of +3 °C and –18 °C, respectively. Gas cooler pressure is 88 bar, the pressure in the intercooler is 56 bar and the pressure at the medium temperature level/tank is 31 bar.

2.1.1. Parametric evaluation

Potential for improvements in the reference system can be investigated using the simulation model (described in detail in part I), in which the influence of components and parameters on the performance can be evaluated. Different systems with different refrigerants will respond differently to certain system modifications; this will depend on the thermophysical properties of the refrigerant which determines how the T-s and P-h diagrams look like. Investigated components and parameters are as follows.

2.1.1.1. High stage IHE. The value for IHE effectiveness that is used in the reference system is 50%, at which the optimum intermediate pressure correlation was generated. As can be seen in Fig. 2 the IHE effectiveness influence on the optimum discharge pressure is small; at 30 °C ambient temperature with 100% effectiveness the value for the optimum pressure is 2% less than the 10% IHE, the high stage COP is about 3% higher. The improvement is higher at high ambient temperature; this can be seen in Fig. 2 where at 40 °C ambient 5% reduction of the optimum pressure value is observed and this results in 9% higher high stage COP. One more advantage that can be observed in the plot in Fig. 2 is that the optimum COP dome becomes flatter with IHE, this will make the system's performance less sensitive to the control of the optimum discharge pressure, especially at high ambient temperatures.

The IHE effectiveness will also affect the optimum intermediate pressure (pressure between high stage compressor stages) and the resulting COP. With an efficient IHE the volumetric

Fig. 2 – High stage COP at different discharge pressures and ambient temperatures for CO₂ trans-critical cycle with different IHE effectiveness values.

refrigerating cooling effect will be increased and the start of the compression process will move to the area of greater superheat where the isentropic lines become flatter. Thus, not only the volumetric refrigerating effect but also the resulting compression work will be higher for the same pressure ratio. As can be seen in Fig. 3, with higher superheat, i.e. higher IHE efficiency, the optimum intermediate pressure value will be reduced. This is due to the fact that the isentropic lines are steeper at lower superheat values which means that it will be more efficient to reject the heat at lower intermediate pressure. In this case with higher IHE efficiency, higher pressure ratio compression will take place in the second stage of the compressor, thereby moving a larger part of the compression to the left in the P-h diagram where the isentropic lines are steeper.

With the intercooler exit temperature assumed constant, the inlet temperature to the intercooler at the intermediate pressure level will be higher with highly effective IHE; this will result in more heat rejected at the intermediate pressure level which reduces the power demand on the high stage compressor. Consequent to the reduced compressor work with

Fig. 3 – High stage COP at different intermediate pressures (pressure between high stage compressor stages), IHE effectiveness values, and ambient temperatures. Evaporating temperature is –9 °C.

high IHE effectiveness the COP of the system will be improved and this is what can be observed in Fig. 3. At 20 °C ambient temperature a 100% effective IHE will improve the COP of the high stage by about 4% over the 50% IHE. At 30 °C ambient the improvement will be about 6%.

2.1.1.2. Low stage IHE. Using the IHE at the low stage with 100% effectiveness and a low stage compressor with isentropic efficiency of about 80% will result in discharge gas temperature of about 75 °C, which is acceptable for good lubricant performance. The rate in increase of compressor power due to the flattening isentropic lines with greater superheat values will be slightly higher than the increase of cooling capacity; this will result in lowering the COP of the low stage. Using IHE with 100% effectiveness will degrade the low stage COP by 3% compared to a system without IHE.

Load ratio is defined as the relation between the cooling capacity at the medium temperature level and the capacity at the low temperature level. Due to the assumed load ratio of 3, this will have a very small influence, about 0.5%, on the total COP of the reference system at 30 °C ambient temperature. This can be seen in the “no de-superheat” line in Fig. 4.

Usually, the IHE is preferred to be used in supermarket application to provide sub-cooling to the liquid supply line to the cabinets in order to prevent evaporation of the refrigerant as a result of the pressure drop. For CO₂, the pressure drop is much smaller than for other refrigerants, using the same tube diameter, which indicates that only a small IHE would be needed to provide a few degrees of sub-cooling.

2.1.1.3. Low stage de-superheat by ambient. The discharge temperature of the low stage compressor will be about 50 °C when the low stage IHE is not used and the superheat in the evaporator is 9 °C. If the effectiveness of the IHE is 50%, then the discharge temperature is about 60 °C. As ambient temperatures are normally much lower there is a potential for removing heat from the low stage directly to the environment so the cooling capacity on the high stage will be reduced and consequently the compressor power will be lowered. This heat removal to the environment at the medium pressure level is referred

Fig. 4 – Influence of the IHE and the free de-superheat on the total COP. Ambient temperature is 30 °C.

to in this study as “free de-superheating” and the heat exchanger is denoted “de-superheater”.

Fig. 4 is a plot of the influence of the effectiveness of the IHE on the total COP of the reference system with and without de-superheater. The improved COP with higher IHE effectiveness is due to the fact that the low stage discharge temperature gets higher and more heat can be removed from the hot gas. For low stage IHE with 50% effectiveness and de-superheating 2% improvement on the total COP can be achieved; this can be seen in Fig. 5 which represents the percentages of improvements on total COP with de-superheater at different IHE effectiveness and ambient temperatures.

It is obvious that the amount of heat that can be rejected from the de-superheater is dependent on the heat sink temperature; at low ambient temperatures more heat can be removed to the ambient due to lower de-superheater exit temperature. As can be seen in Fig. 5, the improvement on the total COP at 10 °C ambient temperature will be about 2.7% compared to 2% at 30 °C.

For cases with lower load ratios than the value of 3, which is used in the reference system, the influence on the total COP will be greater. This is due to the higher heat load that can be rejected to the ambient at the medium pressure level; the plot in Fig. 5 clearly indicates the improvement due to de-superheating for one case with load ratio of 2 at ambient temperature of 30 °C, total load is kept the same.

2.1.1.4. DX evaporator superheat. The superheat value that is used in the reference system was 9 °C which is based on experimental data for NH₃–CO₂ cascade system, reported by Sawalha et al. (2006). The freezers that are used in the experimental cascade system are not specially designed for CO₂; they have long single coil with large diameter. This has been chosen in the experimental rig in order to evaluate the performance of CO₂ in conventional display cabinets. If the cabinet would be specially designed for CO₂ it should be with smaller and shorter pipes then it will be possible to operate the system with lower superheat value with good stability, the low

Fig. 5 – Influence of the free de-superheating on improving the total COP for different values of effectiveness for the low stage IHE and ambient temperatures. One case for load ratio of 2 is calculated.

pressure drop of CO₂ will permit the usage of smaller pipes in the heat exchanger compared to cabinets for other refrigerants. If the superheat value is reduced to 7 °C then the low stage COP will increase by 8% which accounts for about 1.3% improvement in the total COP. The reason for improving the COP is the increase in evaporating temperature due to shifting the approach temperature to the air exit side of the evaporator.

As can be seen in Fig. 6, the influence of the superheat value will be higher at lower ambient temperatures; this is due to the fact that the work of the low stage compressor at low ambient temperatures will account for larger share of the total power consumption.

2.1.1.5. Flooded freezing cabinets. A solution that may improve the COP of the system is to use a low pressure receiver and pump the refrigerant into the freezers in the same way used in the medium temperature cabinets. In the case of the flooded evaporator freezing cabinet, the superheat value is zero and the approach temperature is assumed to be similar to the case of medium temperature cabinets and equals 2 °C; this assumption is due to the improved heat transfer conditions on the refrigerant side. Lowest temperature approach will be at the air exit side of the heat exchanger; therefore, the temperature profile in the heat exchanger can be presented by the sketch in Fig. 7. The warmest product temperature is 2.5 °C higher than the air inlet temperature and the air temperature difference across the heat exchanger is 7.5 °C; these assumptions are similar to the values used for the DX freezer calculations.

In this case with no superheat in the evaporator and smaller approach temperature it is clear, by looking at the temperature profile in Fig. 7 that the evaporating temperature will be higher than in DX case to provide the required product temperature. The evaporating temperature increase due to pump pressure is also taken into account in calculating the product and evaporating temperatures. The pump power needed with a circulation ratio of 2 is included in the total power consumption calculations. Using flooded freezers improved the total system COP which can be seen in Fig. 6, especially at low ambient temperatures on the scale.

Fig. 6 – Total COP at different ambient temperature with 7 and 9 °C DX freezer superheat and flooded evaporator.

Fig. 7 – Temperature profile in the freezer's flooded evaporator.

2.2. Modified centralized system solution

Based on the results of the parametric analysis presented above, it will be possible to evaluate the performance with different system variations; thus the combination of variations which will give the best centralized system performance can be concluded. The COP is compared to that of the reference system solution and presented as a percentage of improvement. The different modifications that can be applied on the system can be found in the schematic diagram in Fig. 8.

The IHE used for low and high stages have effectiveness of 50%; the design of IHE is a trade off between heat transfer and pressure drop and this value is seen to be practically acceptable.

Influence on COP by the superheat in the DX freezers' heat exchangers, flooding the freezers, free de-superheat with and without IHE are calculated and plotted in Fig. 9. As can be seen in the plot; the best COP can be achieved by flooding the evaporator with low stage IHE and free de-superheat, up to 7% improvement at 10 °C can be achieved. Accordingly, this system is defined as the modified centralized solution due to its best COP.

The flooded evaporator solution implies that low pressure receiver and a pump should be used which will add to the installation cost compared to the DX solution. In this case the pump will provide head before the distribution lines which should be sufficient enough to avoid flashing gas in the liquid line, and since solenoid valves will be used to control the flow in the evaporators, instead of an expansion valve, it will be of little importance to strive to avoid flashing gas in the liquid line. Consequently, the reason to use the IHE will be solely to provide higher compressor discharge temperature so more heat can be rejected to the environment. Flooded system with de-superheater and without IHE will produce improved COP over a flooded evaporator without de-superheating and IHE at ambient temperatures lower than 25 °C, above which ambient temperature will be very close or higher than the compressor discharge temperature and no heat will be removed.

DX system with IHE and free de-superheating shows better COP at all temperature ranges than the flooded evaporator without de-superheating. The improvements vary from 2 to about 6% over the whole temperature ranges compared to the reference system. The presence of the IHE is essential with relatively long distribution lines and DX freezers; therefore, the additional component in this solution is the de-superheater which will make this concept more favorable from installation point of view.

Fig. 8 – Modified centralized CO₂ system with DX or flooded evaporator options.

3. Conventional and alternative system solutions

A key objective from investigating the trans-critical centralized CO₂ system is to find out whether or not it will be

a good replacement to conventional technologies and how efficient it is compared to other possible alternatives.

3.1. Conventional system with R404A

Systems with R404A are taken as a reference for conventional technologies where it is used as replacement for CFCs. The R404A system consists of two separate DX circuits; one is single-stage for the medium temperature level and the other is two-stage compression circuit. Due to the steepness of the isentropic lines for R404A the two-stage compression with intercooling has very little influence on improving the COP on the medium temperature level. The two-stage compression at the low temperature level may not be a conventional solution with R404A; however, it has been used in the calculations to see what COP a good R404A system can produce. IHE with 50% effectiveness is used in both low and medium temperature levels' circuits.

The temperature drop in the cabinets and distribution lines are taken to be equal to the values used in sizing the CO₂ system. In practice this will require larger components with R404A or higher temperature/pressure drops for components of the same size. The equal temperature drop assumption is used as a reasonable approximation so detailed sizing and calculations of the pressure and temperature drops for R404A system will not be needed in the model.

The same assumptions used in part I of this study to simulate the DX cabinets were applied in the R404A case. Same

Fig. 9 – Total COP relative improvement for different centralized system variations compared to the reference centralized system solution at different ambient temperatures.

approach temperature was used in the R404A condenser and the isentropic efficiency of the R404A compressor was assumed to be 80%.

3.2. $\text{NH}_3\text{-CO}_2$ cascade

The amount of refrigerant that is used in supermarket refrigeration installation is usually large; therefore natural refrigerants are preferred to be used from environmental point of view. A natural solution which is a strong alternative to conventional systems is the $\text{NH}_3\text{-CO}_2$ cascade concept. Fig. 10 is a schematic of such system solution which has been applied in a laboratory environment, as reported by Sawalha et al. (2005). In this system concept CO_2 is used in the lower stage where pressure levels are acceptable; at condensing temperature of -4°C the pressure is about 31 bar. The temperature difference between CO_2 and NH_3 across the cascade condenser is 2.5°C , which is based on the experimental results presented by Perales Cabrejas (2006). CO_2 has good safety characteristics so it can be used in the shopping area; however, gas detectors should be placed because CO_2 does not have smell or color so it is not self-alarming when it leaks. In this solution, NH_3 is confined into the machine room and leakage will not come in contact with the occupants. NH_3 will have favorable operating conditions in this solution since the pressure level in the system is higher than atmosphere and air leakage into the system is not possible.

3.3. Parallel trans-critical CO_2

This system uses only CO_2 and has been described in part I of this study. It might be preferred over the centralized

trans-critical solution from practical and installation cost points of view. Though the centralized system solution shows better COP values over the parallel system solution this system is being investigated as a possible alternative which should be evaluated against conventional and other alternatives. The parallel system solution has two-stage compression at both temperature levels' circuits.

4. Performance calculations

4.1. Simulation model and total COP calculations

Detailed explanation of the computational simulation model used in evaluating different systems was presented in part I of this study. COP has been calculated for the reference centralized system solution, modified centralized, and the three alternative system solutions. Fig. 11 is a plot of the total COP of each system at different ambient temperatures. As can be seen in the plot, different systems are better at different temperature ranges; except for the parallel system solution which has the lowest COP at all calculated temperatures.

All simulated trans-critical CO_2 systems show lower COP than the other compared systems when the ambient temperature forces the operation into the trans-critical region or close to the critical temperature. The cascade system shows higher COP than the R404A system at the calculated range of ambient temperatures. For ambient temperatures lower than 23°C the centralized system solutions show higher COP than the R404A and for temperatures lower than 16°C the modified centralized solution shows the best COP among all analyzed systems.

If R404A system would operate at a fixed condensing temperature of 40°C , regardless of the change in ambient temperature, then all the solutions with floating condensing produce better COP at ambient temperatures lower than 27°C . With fixed head pressure the cascade system is the only one that has better COP than R404A system.

Fig. 10 – $\text{NH}_3\text{-CO}_2$ cascade system.

Fig. 11 – Total COP of different system solutions for ambient temperatures between 10 and 40°C .

4.2. Ambient temperatures

Since performance differences between the systems under investigation vary over the ambient temperature range, the differences in annual energy consumption will depend on in which climate the system will operate.

Three different climate examples were selected. The first is a typical European climate represented by the climate of Frankfurt/Germany. This city was also selected by Wertenbach and Caesar (1998) for mobile air conditioning applications. A hot climate in the USA is represented by Phoenix – Arizona/USA. Stockholm/Sweden is the third example, selected as a representative of a cold climate.

The ambient temperatures and their variation over each hour of the year were generated using Meteonorm (Remund et al., 2001) for every hour along the year. As can be seen in Fig. 12 the temperature in Stockholm and Frankfurt is lower than 10 °C for more than 50% of the time. In Phoenix the temperature levels tend to be higher; the temperature is above 30 °C for the highest number of hours compared to the rest of the temperature ranges.

A CO₂ trans-critical system in Stockholm or Frankfurt will operate sub-critically for most of the time while in Phoenix the system will operate in the trans-critical region for about 40% of the time.

4.3. Annual energy consumption

Using ambient temperature from Meteonorm (Remund et al., 2001) for the selected cities as an input variable to the simulation model the annual energy consumption for each system in Fig. 11 is calculated. An additional system is added to the comparison which is a single-stage centralized system solution. It was added to the group of analyzed systems because it may practically be more applicable than the two-stage centralized system solutions.

When the ambient temperature is lower than 10 °C, the model uses 10 °C as the input variable which is chosen as a limit for the floating condensation. It is important to point out that

the heat sink temperature supplied to the condenser/gas cooler and the intercooler should be the same otherwise if the intercooler would have lower heat sink temperature then the refrigerant may condense at the inlet of the high stage compressor. When there will be need to control the heat sink temperature it will be difficult to achieve this with air cooler; therefore, water/brine loop may be needed to reject the heat which will make the temperature difference between the refrigerant and ambient higher than the assumed 5 °C in this model.

Fig. 13 represents the annual energy consumption for the different systems under investigation for the three selected cities. As can be seen in the figure, the two-stage parallel system solution has the highest energy consumption among all examined solutions in the three selected climates.

The values presented in Fig. 13 can be related to each other by percentages, this is detailed in Table 1 where systems in the rows are related to the ones in the columns as a percentage. In the table, values of the coldest and hottest cases are chosen for the comparison. In case of Swedish weather conditions the modified centralized system solution, highlighted in yellow, has the lowest energy consumption over the year, 12% less than R404A and 4% less than cascade, followed by the reference centralized system solution and NH₃–CO₂ cascade. All the investigated CO₂ systems, except the parallel solution, consume less energy over the year compared to R404A. This is also correct in the case of Frankfurt and can be observed in Fig. 13. The modified centralized system also has the lowest energy consumption and consumes 2% less energy compared to the cascade.

In the case of hot climate, the NH₃–CO₂ cascade system, highlighted in yellow, has the lowest annual energy consumption; 18% lower than R404A. The modified centralized system has energy consumption which is almost the same as that of R404A. Looking at the three centralized system solutions, it is clear that the modifications to the system are more important at high ambient temperatures where the percentage of improvement is higher and reduction in annual energy consumption is also high.

As has been pointed out in part I of this study, the two-stage compressor is assumed to have good isentropic

Fig. 12 – Number of hours per year for different ambient temperature levels in Stockholm/Sweden, Frankfurt/Germany, and Phoenix – Arizona/USA.

Fig. 13 – Annual energy consumption for different supermarket systems in Stockholm/Sweden, Frankfurt/Germany, and Phoenix – Arizona/USA.

Table 1 – Percentages of annual energy consumption differences between the different solutions

	Annual energy consumption change in % (Row to Column)	R404A Float	2-St. Parallel	NH ₃ /CO ₂ Cascade	1-St. DX-Centralized (Ref)	2-St DX-Centralized	Modified Centralized
Stockholm	R404A Float	0	-6	9	4	10	14
	2-St. Parallel	7	0	16	11	17	21
	NH ₃ /CO ₂ Cascade	-8	-14	0	-5	0	4
	1-St.DX-Centralized (Ref)	-4	-10	5	0	6	10
	2-St DX-Centralized	-9	-15	0	-6	0	4
	Modified Centralized	-12	-17	-4	-9	-3	0
Phoenix	R404A Float	0	-17	22	-13	-8	-1
	2-St. Parallel	20	0	47	5	11	19
	NH ₃ /CO ₂ Cascade	-18	-32	0	-29	-24	-19
	1-St.DX-Centralized (Ref)	15	-5	40	0	6	14
	2-St DX-Centralized	8	-10	32	-6	0	7
	Modified Centralized	1	-16	23	-12	-7	0

Systems in rows related to the ones in columns. The highlighted cell in yellow identifies the sytem that shows the lowest energy consumption. (For interpretation of the references to colour in this table, the reader is referred to the web version of this article).

efficiency at low pressure ratios where such compressor may not be available in the markets nowadays; therefore, the single-stage centralized system seems to be more applicable. As can be noticed in Fig. 13, the single-stage centralized system solution has slightly lower energy consumption than the R404A system in cold and moderate climates but it has higher energy consumption in all climates compared to NH₃–CO₂ cascade.

5. Discussions and conclusions

Performance calculations for possible CO₂ system solutions in supermarket refrigeration and a conventional R404A system have been performed. Influences of different components on the performance of the centralized trans-critical CO₂ system solution have been investigated and the system has been modified and optimized to obtain higher COP. COPs for all systems under investigation have been generated for different ambient temperatures. Annual energy consumptions for the systems have been calculated for three different climates; cold, moderate and hot.

Using IHE in the high stage of the centralized system solution will slightly reduce the optimum discharge pressure value and improve the COP. At 20 °C ambient temperature a 100% effective IHE will improve the COP of the high stage by about 4% over the 50% IHE, at 30 °C ambient the improvement will be about 6%. Additionally, the IHE on the high stage will produce flatter optimum COP curve which implies that the system

will be less sensitive to the optimum discharge pressure control. Effective IHE will also improve the COP with intercooler in which more capacity can be rejected to ambient. It will also reduce the optimum intermediate pressure value.

High values for the IHE effectiveness have been used in the model calculations, in practice IHEs with effectiveness close to 100% do not exist. The point of this analytical approach is to make the influence of the parameter more prominent and to show the ultimate benefit from striving to seek improvement on the IHE performance. The design of the IHE will be a trade off between heat transfer and pressure drop, where a very effective IHE will have high pressure drop which will increase the compressor work. Such an analysis was not included in this study and eventually a moderate effectiveness value of 50% was chosen.

The IHE on the low stage will have very little influence on reducing the COP but it is essential to provide sub-cooling in liquid distribution lines not to have flashing gas before the expansion valves. Improvements on the COP can be achieved when the IHE is combined with de-superheating at the medium pressure level. In this case the better the IHE the more heat that can be rejected in the de-superheater, especially when the ambient temperature is low. About 2.7% improvement on total COP can be achieved at 10 °C ambient temperature with 50% effective IHE, 2% at 30 °C ambient. The lower the load ratio the higher the total COP improvement de-superheating will provide.

Additional improvement on the system can be achieved by reducing the superheat value in the DX freezers or by using

low pressure receiver with flooded evaporator. The best COP resulting from the centralized system solution was with the arrangement of flooded freezers, low stage IHE, high stage IHE, and de-superheater. At low ambient temperature of 10 °C the improvement on the total COP is about 7% compared to the reference centralized system solution. Using DX freezers instead of flooded one with the same arrangement will give comparable COP results; this is mainly due to the greater superheat with DX system and IHE which will increase the inlet temperature to the de-superheater. DX system will have the advantage of lower installation cost and simpler oil handling mechanism.

The differences between the COPs of the analyzed systems vary depending on the ambient temperatures. The modified centralized system solution shows the highest COP among the rest of the systems for ambient temperatures lower than 16 °C, above which the NH₃–CO₂ cascade system has the highest COP. The parallel CO₂ system solution shows the lowest COP along the selected temperature range. It must be pointed out that these results are dependent on the assumptions in the simulation model.

Examining the systems in three different climate conditions; cold, moderate, and hot, showed that the CO₂ systems, except parallel, are better for cold climates, such as the case of Stockholm. Whereas NH₃–CO₂ cascade system is better for hot climates, such as the case of Phoenix – USA; in this climate case the CO₂ modified centralized system has only 1% higher annual energy consumption than the conventional R404A and the modifications on the centralized system solution proved to be more important for high ambient temperature

operating conditions. This analysis shows that from energy consumption point of view the CO₂ centralizes solution is good for low ambient temperatures and the NH₃–CO₂ cascade is good for high ambient temperatures. Both systems proved to be good alternatives to R404A DX system for supermarket refrigeration.

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