JETIR.ORG ISSN: 2349-5162 | ESTD Year : 2014 | Monthly Issue
JOURNAL OF EMERGING TECHNOLOGIES AND INNOVATIVE RESEARCH (JETIR)
An International Scholarly Open Access, Peer-reviewed, Refereed Journal

THERMODYNAMIC ANALYSIS AND PERFORMANCE COMPARISON OF THE DIFFERENT STATE-OF-ART CO_2 SUPERMARKET REFRIGERATION SYSTEMS FOR HIGH AMBIENT

UGA RAM PRAJAPAT
ME (Thermal) Student, Dept. of Mechanical Engineering, MBM Engineering College, Jodhpur, Rajasthan, India- 342001

Abstract: In this paper, the CO_2 booster supermarket's refrigeration systems are analyzed. The analysis has included a standard booster (CB1), a booster with parallel compression (CB2), four configurations of the booster with dedicated mechanical subcooling (CB3) and an R404A multiplex direct expansion refrigeration systems. The thermodynamic analysis of the different configurations of the CO_2 refrigeration systems are described in this paper. The CB3 system has four different configurations in this investigation that uses refrigerants like as R134a, R290, R1234ze and R1234yf. From the investigation, the following parameters are compared for all investigated systems: COP, total power input, optimal gas cooler pressure and flash gas mass flow rate. The performance and power input of the all investigated systems are compared with an R404A multiplex DX refrigeration system for supermarkets in warm climates. Finally, the results from this investigation are concluded in brief.

Keywords: **Supermarket, Refrigeration, Carbon dioxide, Booster, Performance, High ambient**

Nomenclatures:
COP – Coefficient of Performance
DMS – Dedicated Mechanical Sub-cooling
DX – Direct Expansion
EES – Energy Equation Solver
GWP – Global Warming Potential
HS – High Stage
LCCP – Life Cycle Climate Performance
LS – Low Stage
LT – Low Temperature
MT – Medium Temperature
ODP – Ozone Depletion Potential
PC – Parallel Compression
TEWI – Total Equivalent Warming Impact
CB1 – Standard trans-critical CO_2 booster system
CB2 – CO_2 booster system with PC
CB3 – CO_2 booster system with DMS cycle
CO_2 – Carbon di-oxide (chemical name of R744)
h – Enthalpy (kJ/kg)
$\dot{m}$ – Mass flow rate (kg/sec)
$\dot{Q}$ – Heat transfer rate or Amount of heat (kW)
T – Temperature (°C)
$\dot{W}$ – Power input (kW)

Subscripts
1, 2, 3, …… and a, b, c, d – cycle states
Cond – condenser
DMS – dedicated mechanical sub-cooling
Evap – evaporator
FG – flash gas
GC – gas cooler
HS – high stage
in – inlet
LS – low stage
LT – low temperature
MT – medium temperature
PC - parallel compression
rec – receiver
Ref - refrigerant
Tot – total
amb - ambient

1. Introduction

In this paper, the CO_2 booster supermarket's refrigeration systems are analyzed. The analysis has included a standard booster (CB1), a booster with parallel compression (CB2), four configurations of the booster with dedicated mechanical subcooling (CB3) and an R404A multiplex direct expansion refrigeration systems. The thermodynamic analysis of the different configurations of the CO_2 refrigeration systems are described in this paper. The CB3 system has four different configurations in this investigation that uses refrigerants like as R134a, R290, R1234ze and R1234yf. These systems are therefore mathematically modeled and simulation is conducted in MATLAB 2014. The properties of the refrigerant are invoked using REFPROP V9.0. From the investigation, the following parameters are compared for all investigated systems: COP, total power input, optimal gas cooler pressure and flash gas mass flow rate. The performance and power input of the all investigated systems are compared with an R404A multiplex DX refrigeration system for supermarkets in warm climates. Finally, the results from this investigation are concluded in brief.

2. Literature review

Finckh and Sienel [2010] investigated the market introduction of commercially viable CO_2 supermarket refrigeration system. The booster system has been proven as a viable solution for many markets in Europe. Thermodynamic analysis of transcritical CO_2 booster refrigeration systems in the supermarket was performed by Ge and Tassou [2011]. They examined the effects of various parameters. In addition, optimal intermediate pressure was also proposed to improve the system performance. Ommen and Elmegaard [2012] successfully validated a numerical model based on the characteristic curves method to a CO_2 booster supermarket refrigeration system. The results indicated that the cost of the LT cooling product is about twice as high as that of the MT one. Beshr et al. [2015] performed a Life Cycle Climate Performance (LCCP) analysis considering several refrigerants and systems combinations and locations. The performance of the R744 booster refrigeration system was experimentally measured by Sharma et al. [2015], ranged from 3.3 to 1.4 at the ambient temperatures from 10°C to 35°C. The performance of the transcritical CO_2 booster refrigeration system was evaluated by Fricke et al. [2016], over an outdoor ambient temperature range of 15.6 to 32.2°C. The resulting coefficient of performance (COP) of the system was found to vary from 2.2 (at an ambient temperature of 32.2°C) to 4.1 (at an ambient temperature of 15.6°C). Tsamos et al. [2017a] presented an experimental investigation of finned-tube gas coolers/condensers with different designs in a CO_2 booster system. The heat exchangers were mounted in a specially designed test facility that allowed the control of different test conditions and parameters, including air on temperatures and flow rates of the refrigerant, approach temperatures and operating pressures of the cycle.

Mylona et al. [2017] noticed that the CO_2 booster system was found to be the more energy efficient system as well as CO_2 emissions. They concluded that a 17.4% reduction in the total annual energy takes place and the TEWI was dropped by 44% over the baseline system. A similar theoretical study was carried out by Gullo et al. [2017]. They assessed energy saving a maximum of 17% in cold and mild climates over an R404A conventional system. Tsamos et al. [2017b] estimated an energy saving by about 2% on the part of a CO_2 booster system over CO_2 cascade system in both moderate and warm climates. Santosa et al. [2018] simulated the trans-critical CO_2 system with booster hot gas bypass in tropical climates using the EES program. Result showed that the COP decreased gradually with the ambient temperature variation increasing. The advanced exergy analysis based R744 booster refrigeration system's performance was investigated by Gullo et al. [2019]. They were found that the multi-ejector assisted CO_2 system can reduce the total exergy destruction rate by about 39% in comparison with standard CO_2 booster system.

The importance of optimizing the intermediate pressure in a CO_2 refrigeration cycle assisted with parallel compression was demonstrated by Minetto et al. [2005]. They also experimentally proved the feasibility and the reliability of such technology. Chiarello et al. [2010] carried out an experimental study on a CO_2 supermarket system with a parallel compressor for hot climates. They recommended paying close attention to the design temperature of the auxiliary compressor and thus the efficiency of the whole system. Minetto et al. [2014] carried out a review regarding the technological aspects, attained knowledge and some experimental results on the subject of auxiliary compressors combined with ejectors. A transcritical CO_2 booster system with parallel compression was analyzed by Purohit et al. [2017b] using control strategies to optimize the COP. By adopting the PC, the load on the high stage compressor was reduced and extra vapour produced at the receiver was compressed back to the gas cooler.

Improvements in the overall system performance can be achieved by promoting the interchangeability of HS and parallel compressors were suggested by Hafner [2017]. He suggested that the compressors can be linked to either the HS or the parallel suction group with the aid of on/off valves located upstream of them. This would also lead to a "gap-free" control of the refrigeration load, besides lowering the installation cost. The theoretical assessment of R744 booster system with parallel compression was proposed by Pardinas et al. [2017]. They concluded that the adoption of PC is leading to energy savings from 7 to 19% over a conventional booster configurations at an ambient temperature above 10°C. The control strategies for the refrigeration systems were derived by Tsamos et al. [2017b] from experimental tests in the laboratory on a conventional booster refrigeration system. The results showed that the CO_2 booster system with gas bypass compressor can provide the best performance with 5.0% energy savings for the warm climate (Athens, Greece) and 3.65% for the moderate climate (London, UK).

The possibilities of enhancing the performance of CO_2 trans-critical refrigeration systems using a dedicated mechanical sub-cooling cycle were theoretically analysed and discussed by Llopis et al. [2015]. They observed that the cycle combination will allow increasing the COP with 20% and the cooling capacity with 28.8%. The results indicated that this cycle is more convenient to above 25°C ambient. Hafner and Hemmingsen [2015] performed a theoretical study of DMS systems for CO_2 cycles. They compared the performance of this system with an R-404A direct expansion plant, with the same system without DMS and with a system working with an ejector and parallel compressor. Comparative energy and environmental analysis among seven different commercial R744 refrigeration solutions over CO_2 cascade system were performed by Gullo et al. [2016a]. They computed the maximum drop in TEWI about 10.6% in Athens and 25.3% in Valencia for the configurations with both mechanical subcooling and parallel compression that of the cascade system. Gullo et al. [2016b] presented the CO_2 system with dedicated mechanical subcooling that it represents an interesting solution in order to drop the energy consumption in warm climates.

Llopis et al. [2016] experimentally evaluated the performance of CO_2 refrigeration cycle with and without dedicated mechanical sub-cooler. They reported maximum improvement in the cooling capacity and COP by 55.7% and 30.3% at the evaporating temperature of -10°C can be achieved, respectively. Beshr et al. [2016] carried out three laboratory tests at steady state conditions in order to implement an experimentally validated simulation model of a CO_2 booster refrigerating plant with an R134a dedicated mechanical subcooling. Purohit et al. [2017a] performed the comparative assessment of low-GWP based refrigerating plans operating in hot climates. They theoretically investigated and compared two systems of "CO_2 only" booster configurations (i.e. PC and DMS) for the COP and the annual energy consumption with that of R404A direct expansion system climatic locations including New Delhi (India). The energy performance of a "CO_2 only" supermarket refrigeration plant assisted with a dedicated mechanical subcooling (DMS) using zeotropic mixture was proposed by Dai et al. [2018]. They suggested that the overall COP could be enhanced by 4.91% at the evaporating temperature -5°C and external temperature 35°C by adopting R32/R1234ze(Z) over pure R32 refrigerant. Gullo et al. [2018] found in-depth information on the current status and future perspective of transcritical R744 refrigeration systems for supermarket applications. They reviewed the state-of-art "CO_2 only" solutions for supermarket applications. They concluded that these solutions are being continuously cost and efficiency optimized, yielding promising results even in warm climates.

3. Description of the investigated systems

A schematic diagram of the standard CO_2 booster refrigeration system with both LT and MT loads is shown in Figure 1.

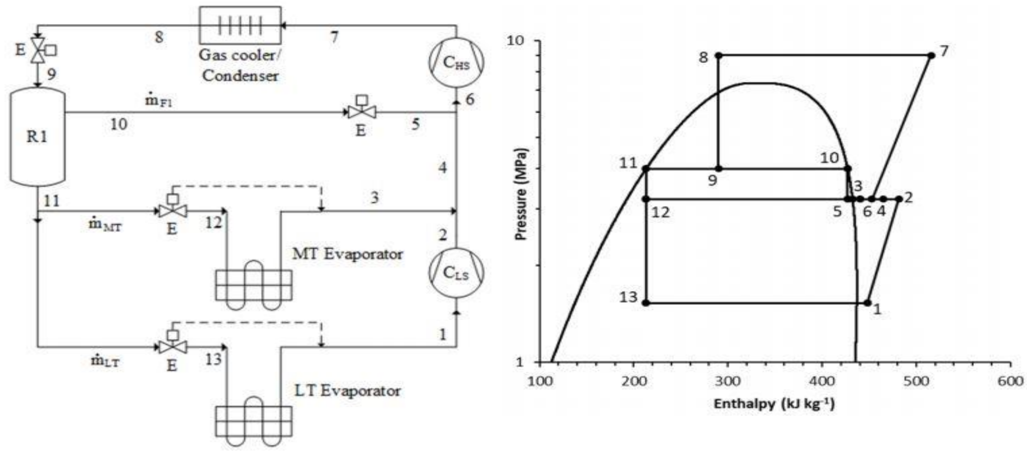

Fig. 1: Schematic and P-h diagrams of the standard CO_2 trans-critical booster refrigeration system [Purohit et al., 2017b].

Refrigerant leaving the LT evaporator is compressed by LS compressor and discharges the refrigerant into the suction manifold of the HS compressor. The LS compressor acts as a booster. The refrigerant from the MT evaporator enters the suction manifold of the HS compressor and joins here the LS discharge vapour. The flash gas is throttled to the MT evaporator pressure which maintains the discharge temperature of the HS compressor. Refrigerant leaving the HS compressor enters to the gas cooler which rejects heat to the ambient. Before entering the refrigerant into the evaporators, the refrigerant expands in expansion valve and receiver at an intermediate pressure. This expansion facilitates further cooling of the refrigerant.

Figure 2 shows a CO_2 booster system with parallel compression. Adoption of parallel compression in the CO_2 cycle reduces the load on the HS compressor, especially for high ambient conditions. The flash gas in the receiver at intermediate pressure is compressed back to the gas cooler using an parallel compression. This modification improves the COP of the system.

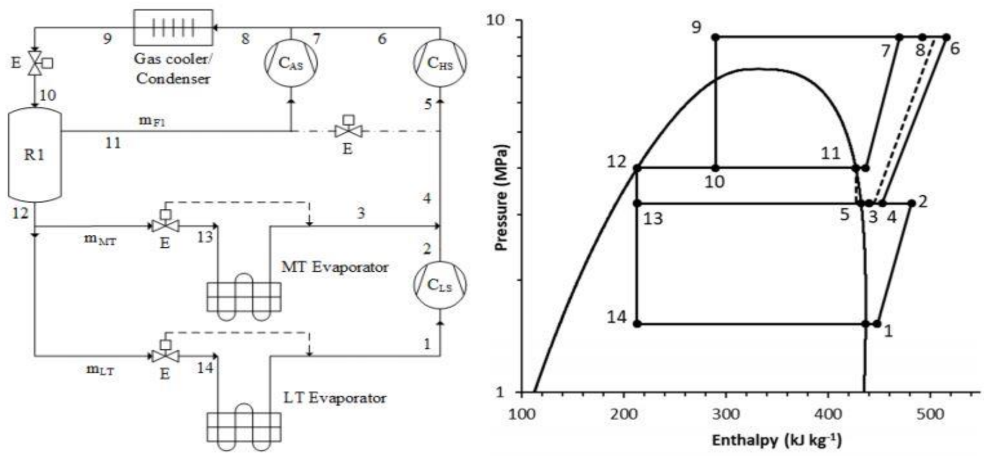

Fig. 2: Schematic and P-h diagrams of CO_2 trans-critical booster system with parallel compression [Purohit et al., 2017b].

Mechanical subcooling is a solution to make trans-critical CO_2 system feasible in warm climates. It uses a small mechanical vapour compression cycle that is coupled to the main cycle at the exit of the gas cooler in order to provide sub-cooling to the main refrigeration cycle. The mechanical sub-cooling cycle operates with a different refrigerant. Figure 3 shows a schematic diagram of the CO_2 booster system with a dedicated mechanical sub-cooling cycle. It has a positive impact on system performance and energy consumption.

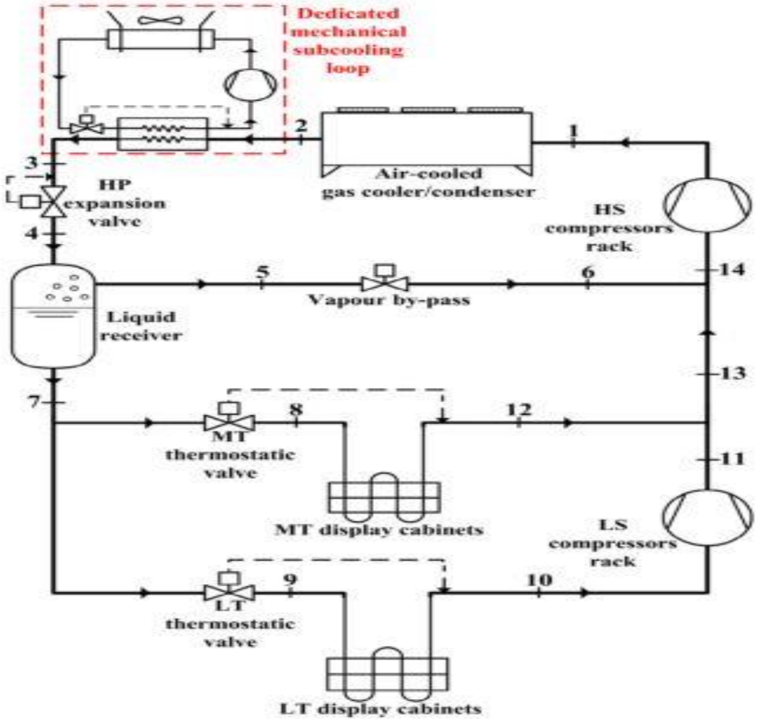

Figure 3: Schematic diagram of transcritical CO_2 booster system with dedicated mechanical subcooling [Gullo et al., 2016b and Purohit et al., 2017a].

4. Thermodynamic modelling

Thermodynamic models are developed assuming steady-state operation, no heat transfer or pressure loss in piping and components and isenthalpic expansion in expansion valves. For the simulation, the MATLAB is used and the required thermo-physical properties of the refrigerant are invoked using REFPROP V9.0. The modelling has included a standard booster (CB1), a booster with parallel compression (CB2), and four configurations of the booster with dedicated mechanical subcooling (CB3) systems.

4.1. Ambient conditions and operating parameters

The systems are analyzed for the warm climate (i.e. high ambient temperature) ranging from 32°C to 45°C of selected locations like Jodhpur, New Delhi etc. The optimum high side pressure is controlled by adjusting the expansion valve. The global efficiency correlations of LS and HS compressors are given in Table 1. These correlations are derived from BITZER software as the function of the pressure ratio for semi-hermetic reciprocating compressors, which respects technological constraints [Gullo et al., 2016b]. Thus, these constraints limit the maximum compressor discharge pressure to 10600 kPa. The applied limit on the compressor discharge pressure and temperature affects the COP, especially at high ambient temperature. When the system operates at greater than 10600 kPa gas cooler pressure, the performance of the system is dropped. With increasing in ambient temperature, the bypass flash gas generation rate is increased at the constant gas cooler pressure and results in down the performance [Purohit et al., 2017b]. The operating parameters for the different configurations of CO_2 booster refrigeration systems are given in Table 2.

Table 1: Compressor's global efficiency correlations [Gullo et al., 2017]

Compressor	Isentropic efficiency
LS (R404A)	$\eta_{LS} = (-0.0004 * (P_{Cond}/P_{LT})^2) - (0.0021 * (P_{Cond}/P_{LT})) + 0.6989$
HS (R404A)	$\eta_{HS} = (-0.0075 * (P_{Cond}/P_{MT})^2) + (0.0652 * (P_{Cond}/P_{MT})) + 0.5609$
LS (R744)	$\eta_{LS} = (-0.0012 * (P_{MT}/P_{LT})^2) - (0.0087 * (P_{MT}/P_{LT})) + 0.6992$
HS (Sub-critical, R744)	$\eta_{HS} = (-0.1155 * (P_{Cond}/P_{MT})^2) - (0.5762 * (P_{Cond}/P_{MT})) - 0.0404$
HS (Transition, R744)	$\eta_{HS} = (-0.1155 * (P_{GC\ or\ Cond}/P_{MT})^2) - (0.5762 * (P_{GC\ or\ Cond}/P_{MT})) - 0.0404$
HS (Trans-critical, R744)	$\eta_{HS} = (-0.0021 * (P_{GC}/P_{MT})^2) - (0.0155 * (P_{GC}/P_{MT})) + 0.7325$

Table 2: Operating parameters of the investigated system

Parameter	Value
LT load (kW)	65
MT load (kW)	120
LT evaporator temperature (°C)	-34.5
MT evaporator temperature (°C)	-8
Internal superheat for LT and MT (°C)	5
Gas cooler pressure (kPa)	10600
Approach temperature for gas cooler (°C)	2
Receiver pressure (kPa)	3500
Sub-cooling outlet temperature of the CO_2 cycle (°C)	20
Compressor efficiency of DMS cycle	0.85
Approach temp. for the condenser of DMS cycle (°C)	10
Evaporator temperature for DMS cycle (°C)	10

4.2. Control strategy

For the smooth operation of the CO_2 system with varying ambient temperature, a suitable control strategy is required. The control strategies for different cycle regions and temperature are given in Figure 4. In the available literature, the research on control strategy is limited. The control strategy was suggested by Gullo et al. [2016b]. This control strategy was also adopted by Purohit et al. [2017b] for optimum exergetic and economic analysis on various modifications of CO_2 booster refrigeration systems. The control strategy has four zones i.e. two zones in sub-critical condition, one in transition condition and one in trans-critical condition. From the figure 4, it is seen that the full trans-critical operation, mode IV, commences at the temperature greater than 28°C, where the gas cooler outlet temperature was held at 2°C higher than the ambient temperature, while the gas cooler pressure was optimized. Figure 5 shows the different operating zones for the R744 booster refrigeration system.

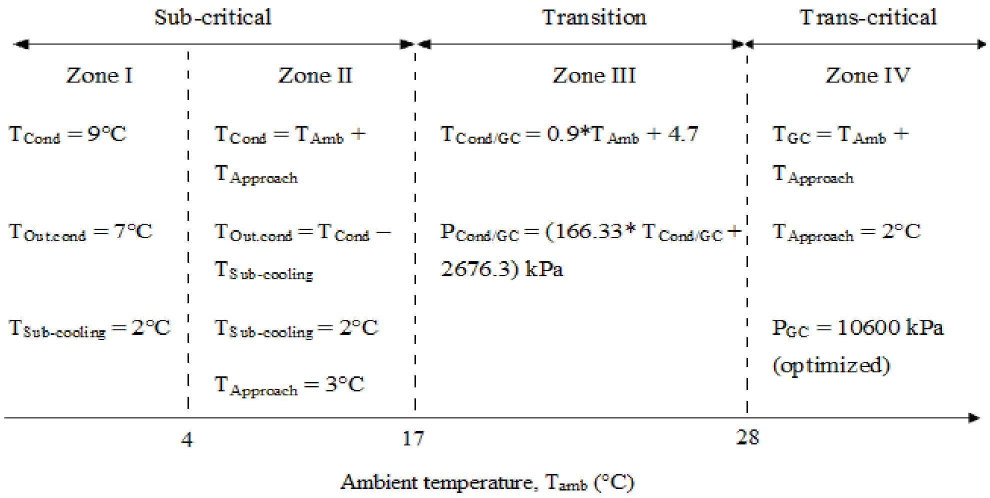

Figure 4: Control strategy for the booster systems [Purohit et al., 2017b].

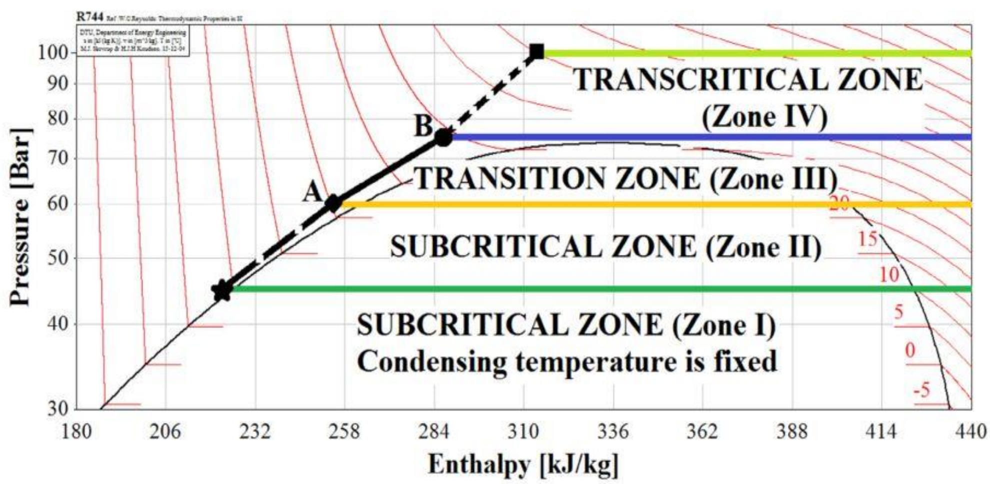

Figure 5: Definition of the operating zones for the R744 booster refrigeration system [Gullo et al., 2016b].

4.3. Thermodynamic analysis for standard CO_2 booster cycle (CB1), CO_2 booster with PC (CB2) and CO_2 booster with DMS (CB3)

In this paper, the analysis has included a standard CO_2 booster (CB1), a CO_2 booster with parallel compression (CB2), and four configurations of the CO_2 booster with dedicated mechanical subcooling (CB3). The CB3 system has four different configurations in this investigation that uses refrigerants like as R134a, R290, R1234ze and R1234yf.

Table 3: Thermodynamic equations for different CO_2 booster systems with standard cycle (CB1), with PC (CB2) and with DMS cycle (CB3)

Component Name	Standard CO_2 booster	CO_2 booster with PC	CO_2 booster with DMS cycle
LS Compressor	$\dot{W}_{LS} = \dot{m}_{LT} * (h_2 - h_1)$	$\dot{W}_{LS} = \dot{m}_{LT} * (h_2 - h_1)$	$\dot{W}_{LS} = \dot{m}_{LT} * (h_2 - h_1)$
Mixing of refrigerant (MT with LT)	$(\dot{m}_{LT} + \dot{m}_{MT}) * h_4 = \dot{m}_{LT} * h_2 + \dot{m}_{MT} * h_3$	$(\dot{m}_{LT} + \dot{m}_{MT}) * h_4 = \dot{m}_{LT} * h_2 + \dot{m}_{MT} * h_3$	$(\dot{m}_{LT} + \dot{m}_{MT}) * h_4 = \dot{m}_{LT} * h_2 + \dot{m}_{MT} * h_3$
Mixing of refrigerant (FG with LT and MT)	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_6 = \dot{m}_{FG} * h_5 + (\dot{m}_{LT} + \dot{m}_{MT}) * h_4$	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_7 = \dot{m}_{FG} * h_6 + (\dot{m}_{LT} + \dot{m}_{MT}) * h_5$	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_6 = \dot{m}_{FG} * h_5 + (\dot{m}_{LT} + \dot{m}_{MT}) * h_4$
HS compressor	$\dot{W}_{HS} = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_7 - h_6)$	$\dot{W}_{HS} = (\dot{m}_{LT} + \dot{m}_{MT}) * (h_5 - h_4)$	$\dot{W}_{HS} = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_7 - h_6)$
Gas cooler	$\dot{Q}_{GC} = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_7 - h_8)$	$\dot{Q}_{GC} = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_7 - h_8)$	$\dot{Q}_{GC} = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_7 - h_8)$
DMS cycle	-	-	$\dot{W}_{DMS} = \dot{m}_{DMS} * (h_b - h_a)$ $\dot{Q}_{Cond} = \dot{m}_{DMS} * (h_b - h_c)$ $h_c = h_d$ $\dot{m}_{DMS} * (h_a - h_d) = (\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * (h_8 - h_9)$
Expansion valve	$h_8 = h_9$	$h_8 = h_9$	$h_9 = h_{10}$
Receiver	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_9 = (\dot{m}_{FG} * h_{10}) + (\dot{m}_{LT} + \dot{m}_{MT}) * h_{11}$	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_9 = (\dot{m}_{FG} * h_{10}) + (\dot{m}_{LT} + \dot{m}_{MT}) * h_{11}$	$(\dot{m}_{LT} + \dot{m}_{MT} + \dot{m}_{FG}) * h_{10} = (\dot{m}_{FG} * h_{11}) + (\dot{m}_{LT} + \dot{m}_{MT}) * h_{12}$
Flash gas bypass valve	$h_{10} = h_5$	-	$h_{11} = h_5$
Parallel compressor	-	$\dot{W}_{PC} = \dot{m}_{FG} * (h_{10} - h_6)$	-
MT valve	$h_{11} = h_{12}$	$h_{11} = h_{12}$	$h_{12} = h_{13}$
LT valve	$h_{11} = h_{13}$	$h_{11} = h_{13}$	$h_{12} = h_{14}$
MT evaporator	$\dot{Q}_{MT} = \dot{m}_{MT} * (h_3 - h_{12})$	$\dot{Q}_{MT} = \dot{m}_{MT} * (h_3 - h_{12})$	$\dot{Q}_{MT} = \dot{m}_{MT} * (h_3 - h_{13})$
LT evaporator	$\dot{Q}_{LT} = \dot{m}_{LT} * (h_1 - h_{13})$	$\dot{Q}_{LT} = \dot{m}_{LT} * (h_1 - h_{13})$	$\dot{Q}_{LT} = \dot{m}_{LT} * (h_1 - h_{14})$
Refrigeration effect	$\dot{Q}_{Ref} = \dot{Q}_{LT} + \dot{Q}_{MT}$	$\dot{Q}_{Ref} = \dot{Q}_{LT} + \dot{Q}_{MT}$	$\dot{Q}_{Ref} = \dot{Q}_{LT} + \dot{Q}_{MT}$
Power consumption	$\dot{W}_{Tot} = \dot{W}_{LT} + \dot{W}_{MT}$	$\dot{W}_{Tot} = \dot{W}_{LT} + \dot{W}_{MT} + \dot{W}_{PC}$	$\dot{W}_{Tot} = \dot{W}_{LT} + \dot{W}_{MT} + \dot{W}_{DMS}$
COP	$COP = \dot{Q}_{Ref}/\dot{W}_{Tot}$	$COP = \dot{Q}_{Ref}/\dot{W}_{Tot}$	$COP = \dot{Q}_{Ref}/\dot{W}_{Tot}$

where h is enthalpy in kJ/kg, $\dot{m}$ is mass flow rate of refrigerant in kg.s^{-1}, $\dot{W}$ is work input rate or power consumption in kW, $\dot{Q}$ is heat transfer rate in kW and COP is coefficient of performance.

5. Results and discussion

In this section, the following parameters are compared for all investigated systems: COP, total power input, optimal gas cooler pressure and flashed mass flow rate. The COP is computed as the ratio of the total cooling capacity to the total power input. The total cooling capacity or refrigeration load is the sum of MT and LT evaporators load. The total power input is calculated as the sum of power consumed by the LS compressor and HS compressor for the CB1 system. While for CB2, the total power input is calculated by the sum of power consumed by the LS compressor, HS compressor and parallel compressor. For CB3, the total power input is calculated by the sum of power consumed by the LS compressor, HS compressor and DMS compressor. The COP and total power input of the all investigated CO_2 supermarket's refrigeration systems are compared with R404A multiplex DX system.

Optimal pressure for gas cooler corresponds to the pressure where the system is operated at maximum COP for a given ambient temperature. The flashed mass flow rate of refrigerant is isenthalpically expanded by bypass expansion valve in CB1 and CB3; whereas the flashed mass flow rate of refrigerant is isentropically compressed by the parallel compressor in CB2 system.

5.1. Comparison in term of COP for supermarkets refrigeration systems

Figure 6 compares the resulting COPs of all investigated CO_2 systems at different ambient temperatures over R404A multiplex DX system. It can be noticed that the CB3 has the highest COPs, whereas the CB1 and CB2 have lower COPs than CB3 system. Hence, the CB3 system is a suitable solution to the supermarkets in a warm climate. From Figure 6, it can be seen that the COP for all systems has a negative trend with an increase in ambient temperature in mode IV. All investigated configurations of CB3 system have almost same COPs for the entire range of investigation (ambient temperature, 32°C to 45°C). At an ambient temperature of 32°C the COP for CB1, CB2, CB3 with R134a, R290, R1234ze and R1234yf refrigerants in DMS cycle are found to be 1.4087, 1.5021, 1.6253, 1.6240, 1.6254 and 1.6228 respectively. While at the highest investigated ambient temperature of 45°C the COP for CB1, CB2, CB3 with R134a, R290, R1234ze and R1234yf refrigerants in DMS cycle are found to be 0.9014, 1.0433, 1.4542, 1.4486, 1.4539 and 1.4394 respectively.

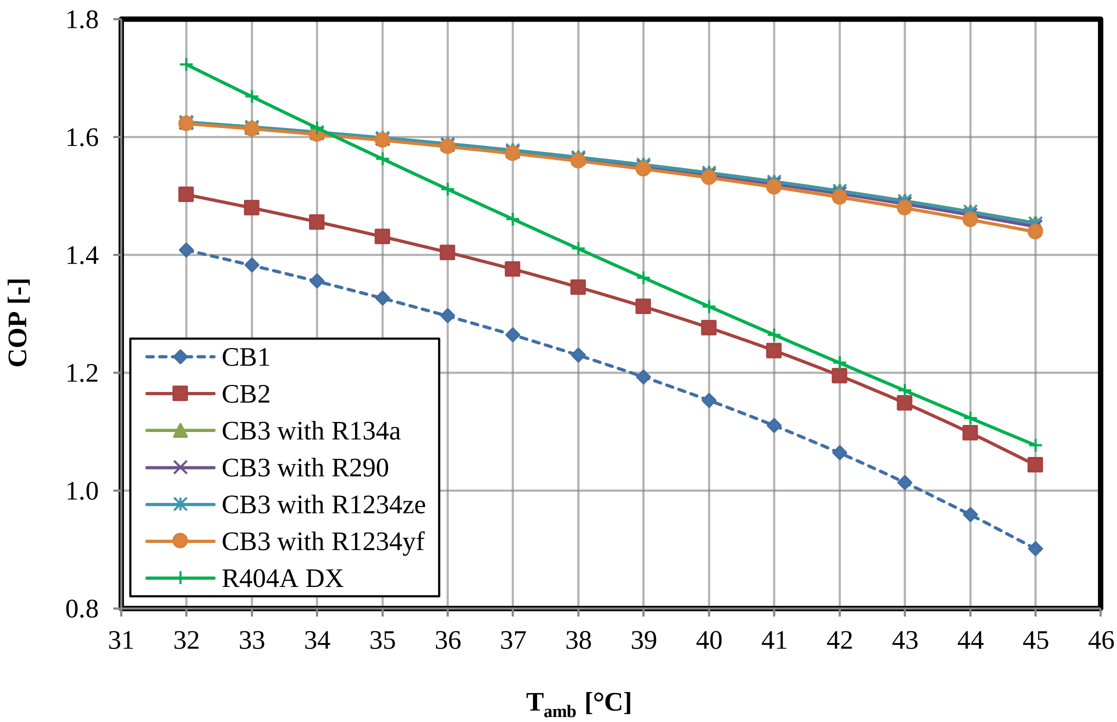

Figure 6: Comparison in term of COP for CO_2 supermarkets refrigeration systems at different ambient temperatures.

In comparison with an R404A multiplex DX system, all the configurations of CO_2 booster equipped with the DMS cycle lead over the range of 35°C to 45°C ambient temperatures. At the highest investigated ambient temperature of 45°C, the improvement in COPs of the systems CB3 with R134a, R290, R1234ze and R1234yf over an R404A multiplex DX system are found to be 34.99%, 34.47%, 34.96% and 33.61%, respectively.

5.2. Comparison in term of total power input

Figure 7 shows the comparison of the total power input of all investigated systems at different ambient temperatures. It is observed that the power saving for the CB3 system is highest followed by CB1 and CB2 systems. The adoption of DMS cycle on CO_2 booster system leads to maximum power saving by about 28.26% at highest investigated ambient temperature of 45°C. The maximum power saving for CB3 with R134a, R290, R1234ze and R1234yf refrigerant in DMS cycle over R404A multiplex DX system is found to be 25.92%, 25.63%, 25.90% and 25.15% at 45°C ambient temperature, respectively.

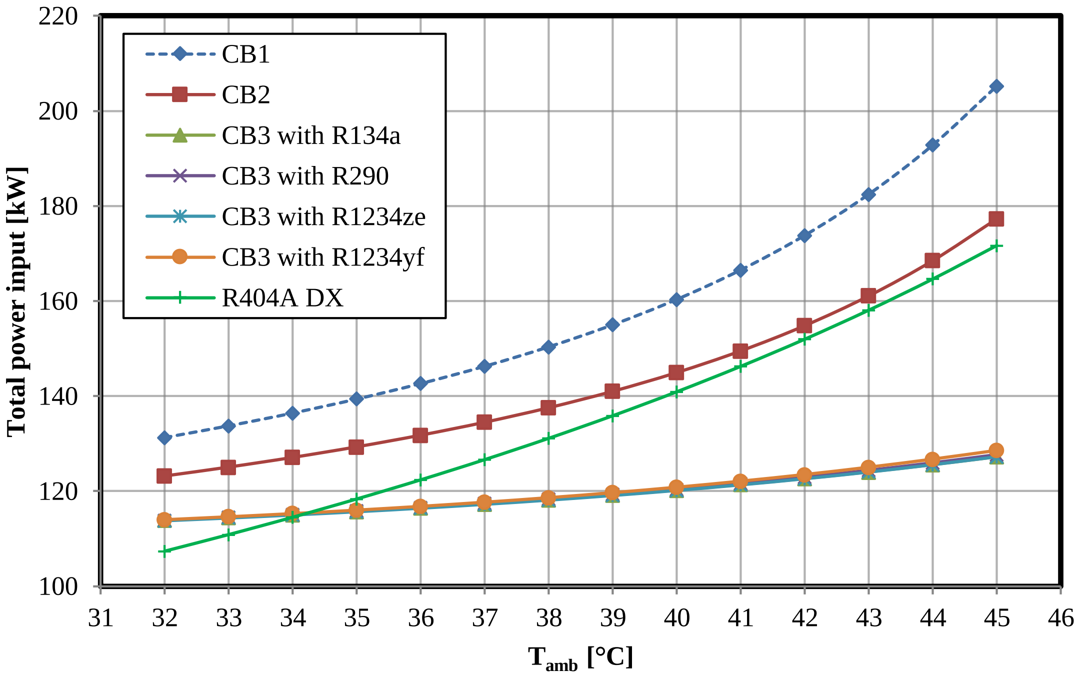

Figure 7: Comparison in term of total power input for CO_2 supermarkets refrigeration system at different ambient temperatures.

5.3. Optimal gas cooler pressure

Optimal gas cooler pressures for all the investigated systems at different ambient temperatures are shown in Figure 8. It is observed that the optimal gas pressures for CB1 and CB2 show an almost linear rise with an increase in ambient temperature. The highest operating gas cooler pressure is observed for the CB1 system. The CB3 system is found with the lowest optimal gas cooler pressure. For this system, the optimal gas cooler pressure first increases linearly for up to 35°C and then it suddenly decreases and remains constant for 36°C to 45°C. The decrease in mechanical sub-cooling capacity is the reason for constant pressure.

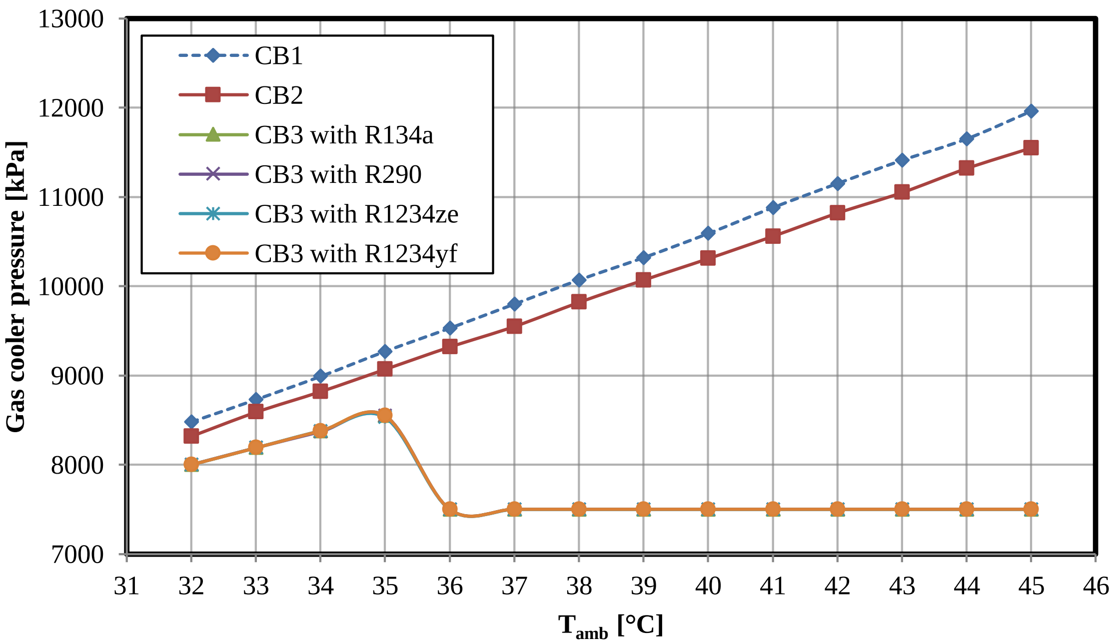

Figure 8: Optimal GC pressure for CO_2 supermarkets refrigeration system at different ambient temperatures.

5.4. Effects on flashed gas mass flow rate

Variation in flash gas generated for all the investigated systems is shown in Figure 9. With an increase in ambient temperature, the flashed gas mass flow rate at receiver increases due to the fixed gas cooler pressure for CB1 and CB2 systems. While the flash gas generation remains constant and the same for all investigated configurations of CB3 at different ambient temperatures.

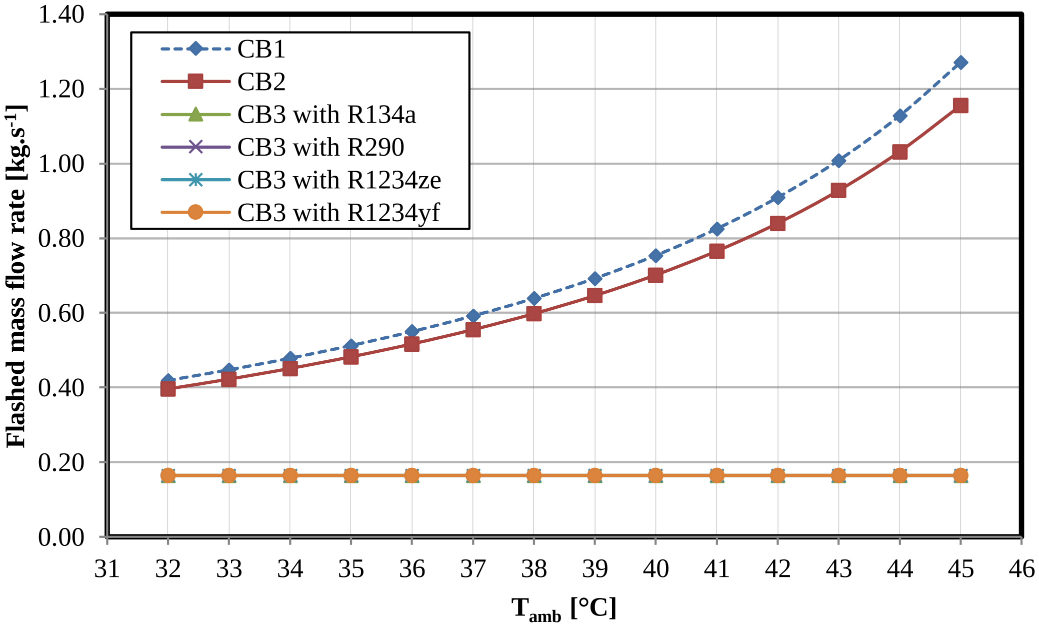

Figure 9: Flashed gas mass flow rates for CO_2 supermarkets refrigeration system at different ambient temperature.

6. Conclusions

The COP and power consumption of all the investigated refrigeration systems have been theoretically evaluated in this chapter. The investigated systems have involved CO_2 standard booster (CB1), booster with PC (CB2) and booster with DMS (CB3) systems for supermarkets. Their performance has been compared over R404A multiplex direct expansion refrigeration system at different ambient temperatures for a warm climate. From the simulation, the adoption of the DMS in the booster system has been found more energetically beneficial than the CB1 and CB2 systems at high ambient temperatures. All investigated configurations of CB3 system have almost same COPs for the entire range of investigation (ambient temperature, 32°C to 45°C). At the highest investigated ambient temperature of 45°C the COP for CB1, CB2, CB3 with R134a, R290, R1234ze and R1234yf refrigerants in DMS cycle are found to be 0.9014, 1.0433, 1.4542, 1.4486, 1.4539 and 1.4394 respectively.

The adoption of DMS cycle on CO_2 booster system leads to maximum power saving by about 28.26% at highest investigated ambient temperature of 45°C. The maximum power saving for CB3 with R134a, R290, R1234ze and R1234yf refrigerant in DMS cycle over R404A multiplex DX system is found to be 25.92%, 25.63%, 25.90% and 25.15% at 45°C ambient temperature, respectively. The highest operating gas cooler pressure is observed for CB1. With an increase in ambient temperature, the flashed gas mass flow rate at receiver increases due to the fixed gas cooler pressure for CB1 and CB2 systems. While the flash gas generation remains constant and the same for all investigated configurations of CB3 at different ambient temperatures.

Based on the simulation study, the following future aspects are:

1. To investigate the performance of the transcritical CO_2 booster system with DMS cycle for each configurations by experimentally.
2. A booster system with both parallel compression or DMS cycle and multi-ejector may be improved system.
3. Integrated CO_2 transcritical refrigeration system may be improved system for both refrigeration and air-conditioning.

7. References

[1] Beshr M., Aute V., Sharma V., Abdelaziz O., Fricke B., Radermacher R., "A comparative study on the environmental impact of supermarket refrigeration systems using low GWP refrigerants", International Journal of Refrigeration, 56 (2015) 154-164.

[2] Beshr M., Bush J., Aute V., Radermacher R., "Steady state testing and modeling of a CO2 two-stage refrigeration system with mechanical sub-cooler", In: Proceeding of the 12th IIR Gustav Lorentzen Natural Working Fluids Conference, Edinburgh, UK, 1146 (2016).

[3] Chiarello M., Girotto S., Minetto S., 'CO_2 supermarket refrigeration system for hot climates", In: Proceeding of the 9th IIR Gustav Lorentzen Conference on Natural Refrigerants, Sydney, Australia, 39 (2010).

[4] Dai B., Liu S., Li H., Sun Z., Song M., Yang Q., Ma Y., "Exergetic performance of transcritical refrigeration cycles with mechanical subcooling using zeotropic mixture as refrigerant", Energy, 150 (2018) 205-221.

[5] Finckh O., Sienel T., "Market introduction of commercially viable CO_2 supermarket refrigeration systems", In: Proceeding of the 9th IIR Gustav Lorentzen Conference, Sydney, Australia, (2010).

[6] Fricke B., Zha S., Sharma V., Newel J., "Laboratory evaluation of a commercial CO_2 booster refrigeration system", In: Proceeding of the 16th International R&AC conference, West Lafayette, USA, 1691 (2016).

[7] Ge Y. T., Tassou S. A., "Thermodynamic analysis of transcritical CO_2 booster refrigeration systems in supermarket", Energy Conversion and Management, 52:49 (2011) 1868-1875.

[8] Gullo P., C2307658_521831_392_401ortella G., Polzot A., "Energy and environmental comparison of commercial R744 refrigeration systems operating in warm climates", In: Proceeding of the 4th IIR Conference on Sustainability and the Cold Chain, 7th – 9th April, Auckland, New Zealand, (2016a).

[9] Gullo P., Elmegaard B., Cortella G., "Energy and environmental performance assessment of R744 booster supermarket refrigeration systems operating in warm climates," International Journal of Refrigeration, 64 (2016b) 61-79.

[10] Gullo P., Tsamos K. M., Hafner A., Ge Y. T., Tassou S. A., "State-of-the-art technologies for transcritical R744 refrigeration systems: A theoretical assessment of energy advantages for European food retail industry", Energy Procedia, 123 (2017) 46–53.

[11] Gullo P., Hafner A., Banasiak K., "Transcritical R744 refrigeration systems for supermarket applications: Current status and future prospective", International Journal of Refrigeration, 93 (2018) 269-310.

[12] Gullo P., Hafner A., Banasiak K., "Thermodynamic performance investigation of commercial R74 booster refrigeration plants based on advanced exergy analysis", Energies, 354:12 (2019).

[13] Hafner A., Hemmingsen A. K., "R744 refrigeration technologies for supermarkets in warm climates", In: Proceeding of the 24th IIR International Congress of Refrigeration, 16th – 22nd August, Yokohama, Japan, 168 (2015).

[14] Hafner A., "Integrated CO_2 system refrigeration, air conditioning and sanitary hot water", In: Proceeding of the 7th IIR Ammonia and CO_2 Refrigeration Technologies Conference, 11th – 13th May, Ohrid, Macedonia, (2017).

[15] Llopis R., Sancheg D., Sanz-Kock C., Cabello R., Torrella E., "Energy and environmental comparison of two-stage solutions for commercial refrigeration at low temperature: Fluids and systems," Applied Energy, 138 (2015) 133-142.

[16] Llopis R., Nebot-Andres L., Cabello R., Sanchez D., Catalan-Gil J., "Experimental Evaluation of a CO_2 Transcritical Refrigeration Plant with Dedicated Mechanical Subcooling", International Journal of Refrigeration, 69 (2016) 361-368.

[17] Minetto S., Cecchinato L., Corradi M., Fornasieri E., Zilio C., "Theoretical and experimental analysis of a CO_2 refrigerating cycle with two-stage throttling and suction of the flash vapour by an auxiliary compressor", In: Proceeding of IIR International Conference-Thermophysical properties and Transfer process of Refrigerants, Vicenza, 31st August – 2nd September, 37 (2005).

[18] Minetto S., Girotto S., Salvatore M., Rossetti A., Marinetti S., "Recent installations of CO_2 supermarket refrigeration system for warm climates: Data from field", In: Proceeding of the 3th IIR Conference on Sustainability and the Cold Chain, 23rd – 25th June, London, UK, 119 (2014).

[19] Mylona Z., Kolokotroni M., Tsamos K. M., Tassou S. A., "Comparative analysis on the energy use and environmental impact of different refrigeration systems for frozen food supermarket application", Energy Procedia, 123 (2017) 121-130.

[20] Ommen T., Elmegaard B., "Numerical model for thermoeconomic diagnosis in commercial transcritical/subcritical booster refrigeration systems" Energy conversions of Management, 60 (2012) 161-169.

[21] Pardinas A. A., Hafner A., Banasiak K., "Integrated R744 ejector supported parallel compression racks for supermarkets: Concept and study state simulation of configurations", In: Proceeding of the 7th IIR Ammonia and CO_2 Refrigeration Technologies Conference, 11th – 13th May, Ohrid, Macedonia, (2017).

[22] Purohit N., Gullo P., Dasgupta M. S., "Comparative assessment of low-GWP based refrigerating plants operating in hot climates," Energy Procedia, 109 (2017a) 138-145.

[23] Purohit N., Gupta D. K., Dasgupta M. S., "Energetic and economic analysis of trans-critical CO_2 booster for refrigeration in warm climate condition", International Journal of Refrigeration, 80 (2017b) 182-196.

[24] Santosa IDMC, Sudirman, IGNS Waisnawa, Sunu PW, Temaja IW, "Simulation of trans-critical CO_2 refrigeration system with booster hot gas bypass in tropical climate", Journal of Physics, 953 (2018).

[25] Sharma V., Fricke B., Bansal P., Zha S., "Evaluation of a transcritical CO_2 supermarket refrigeration system for the USA market", In: Proceeding of the 6th IIR Ammonia and CO_2 Refrigeration Technologies Conference, Ohrid, Macedonia, 16th -18th April, (2015).

[26] Tsamos K. M., Ge Y. T., Santosa I., Tassou S. A., "Experimental investigation of gas cooler/ condenser designs and effects on a CO_2 booster system", Applied Energy, 186:3 (2017a) 470-479.

[27] Tsamos K. M., Ge Y. T., Santosa I., Tassou S. A., Bianchi G., Mylona Z., "Energy analysis of alternative CO_2 refrigeration system configurations for retail food applications in moderate and warm climates", Energy Conversion and Management, 150 (2017b) 822-829.