4
Results

Provided with the theoretical analysis performed in Sections 2.2 and 3.2, the new refrigeration technologies are compared with the direct expansion (DX) system in terms of performance and energy consumption. Further, results are obtained for the systems operating with different refrigerants and in distinct geographic locations, in order to access a wider variety of trends.

Regarding the refrigerant choice, R404A, R407A, R407F and HDR81 are considered for the DX refrigeration system and the main cycle of the pumped CO_2, whereas the CO_2 booster technology and the pumped CO_2 secondary cycle operate with carbon dioxide. R404A has been regarded as an extremely useful refrigerant in a number of commercial refrigeration systems for over twenty years, as reported by Linde [27, 28]. Given the growing focus on environmental impact, R404A performance is compared to that of mixtures that can, eventually, replace it. In that case, R407A, R407F and HDR81 are chosen for representing ideal retrofit solutions for many existing R404A systems, besides combining environmental gains with low energy costs [27, 28]. CO_2 is considered for having emerged as one of the most promising environmentally friendly and energy efficient refrigerants in the food and refrigeration industry, due, in particular, to its favourable thermodynamic and transport properties, according to Bansal [1].

Locations chosen for the analysis are Atlanta (GA), Boulder (CO), and Philadelphia (PA), in the United States; Stockholm, in Sweden; and Manaus (AM) and Rio de Janeiro (RJ) in Brazil. These cities have been selected as they represent different climates, whereas three of them (Atlanta, Boulder and Philadelphia) are located near the stores considered by Kazachki [34] as reference for supermarket data.

Energy efficiency of the refrigeration systems is compared based on two parameters: cycle coefficient of performance and annual energy consumption. The COP is a direct measure of the efficiency of the refrigeration cycle for a given condition, being evaluated at a single operating condition. Annual energy consumption is, according to Kazachki [34], a reliable indicator of the performance of a supermarket refrigeration system, in particular regarding design and operational efficiency, for taking into account the range of local ambient temperatures during the year as well as their frequency.

Following Kazachki [34], both heat reclaim and evaporator defrost procedures are excluded from the analysis, as well as heating and air-conditioning loads, building fire and safety code, store lighting, plug loads, and the HVAC annual consumption. Including these processes in the analysis would require further input data, not adding, however, any differential aspect to the technologies under study.

4.1 Direct expansion refrigeration system

The performance of the pumped CO_2 and CO_2 booster technologies is compared with that of a DX refrigeration system. The model developed by Portilla [52] is considered for the evaluation of the efficiency of the DX system. Figure 4.1 depicts the schematics of the DX refrigeration cycle.

In Portilla's direct expansion model, refrigerant enters the compressor as saturated or superheated vapor at state 1, being compressed to a high-pressure gas at state 2. Pressure drop and temperature change take place in the discharge line, with the refrigerant exiting the device at the condenser inlet pressure.

Refrigerant, then, enters the condenser as superheated vapor at state 3, rejecting heat to the surrounding medium. As a result of this process, saturated or subcooled liquid leave the condenser at state 4, with temperature still above that of the surroundings. By passing through the liquid line, the refrigerant experiences once again pressure drop and temperature change, exiting as a subcooled liquid at high pressure.

The subcooled liquid refrigerant at state 5 is throttled to the evaporator inlet pressure as it passes through an expansion valve or capillary tube. Next, the refrigerant enters the evaporator as a low vapor quality saturated mixture at state 6, leaving as saturated or superheated vapor at state 7. During the complete evaporation of the refrigerant, heat is absorbed from the refrigerated space. Sensible heat gain (with temperature change) and pressure drop in the suction line complete the cycle, with the refrigerant leaving the evaporator as low pressure vapor and reentering the compressor at state 1.

4.2 Weather bin data

Based on Kazachki's study [34], the energy comparative analysis of the three refrigeration technologies is performed considering an estimation of the

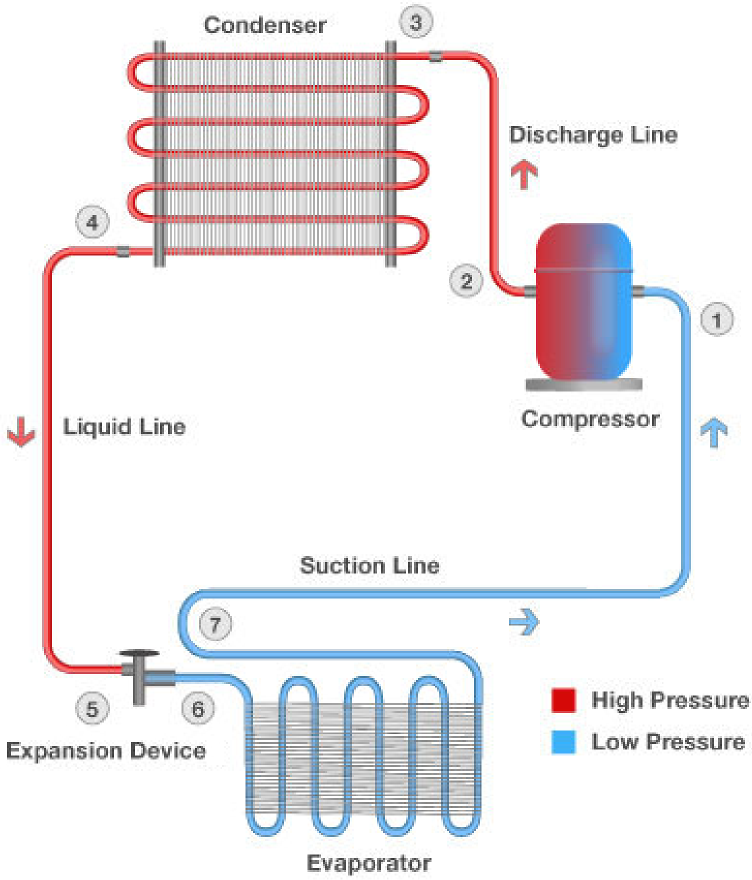


Figure 4.1: Scheme comprising the seven control volumes of the direct expansion refrigeration system [130].

COP and annual energy consumption at six different geographic locations. The calculations are performed accross the range of ambient temperatures for each of the cities during the year, as both the power input and the number of operating hours vary with the ambient temperature. In order to maintain generality, the set of ambient temperatures is divided into temperature intervals ("bins").

Ambient temperatures and their variation over the year for Atlanta, Boulder and Philadelphia were obtained from ASHRAE's Weather Year for Energy Calculations 2 (WYEC2) [133] for every hour of the year. Typical year hourly data for Brazilian cities of Manaus and Rio de Janeiro were taken from the Solar and Wind Energy Resource Assessment project (SWERA) [134]. As for Stockholm's wheater statistical data, the International Weather for Energy Calculations (IWEC) [135], a result of ASHRAE Research Project 1015, is the source considered.

Tables C.1 to C.6, in Appendix C, show the bin hours for the six geographic locations considered.

4.3 Input data

Following Kazachki's approach for the theoretical analysis of alternative supermarket refrigeration technologies [34], specifications and operational features of an existing store layout are considered as the basis for defining a more simplified set of parameters, which realistically reflect currently-designed supermarket refrigeration systems. Ge and Tassou's performance evaluation of supermarket refrigeration systems [77] is also an important source of operational features in this study, particularly for specifications regarding the CO_2 booster system. Thus, according to Kazachki [34] and Ge and Tassou [77], the key conditions assumed for the energy analysis are described next, with a more detailed description of these parameters provided in Tables 4.1 to 4.3.

The dominant fraction of the necessary cooling load is that from display cases, coolers, and freezers, with small air conditioning units contributing with the additional portion. From Kazachki [34], it is assumed that the net refrigerating loads do not vary with the outdoor air ambient conditions, as they perform in an air-conditioned indoor environment. The medium temperature cooling load is assumed as 250.89 kW (856,079 Btu/h), yielding an evaporating temperature of -5.6°C (22°F), whilst the low temperature cooling load is set at 87.921 kW (300,000 Btu/h) with evaporating temperature of -27.8°C (-18°F). These cooling loads and evaporating temperatures closely match those present in the supermarket store selected as reference for Kazachki's study [34].

As a matter of fact, the cooling loads in display cases, coolers, and freezers may differ during the year, as an effect of variations in indoor dry-bulb temperature and relative humidity [34]. However, capturing these changes and implementing them into the analysis would require performance data from manufacturers of refrigerated fixtures, which are often not available or demand a large number of additional tests from the manufacturer, as reported by Kazachki [34]. Further, data on refrigeration loads and evaporating temperatures in terms of dry and wet bulb temperatures are not expected to become available in the near future, given the large number of refrigerated fixtures and the fast development of new and existing models [34].

In the DX and pumped CO_2 refrigeration systems, for both medium and low temperature applications, and all operating refrigerants, an efficiency (including electric motor efficiency) of 65% is assumed for the compressor (Kazachki [34]). Regarding the booster CO_2 technology, Ge and Tassou [77] suggested a number of semi-hermetic reciprocating compressors for the high pressure stage, and other semi-hermetic compressors for the low pressure stage. The following expressions were proposed by the authors to estimate the isentropic efficiency of each compressor, which is a function of the pressure ratio:

$$\eta_{cp,hs} = 0.7595 - 0.0328 R_p \tag{4.1}$$

$$\eta_{cp,ls} = 0.7178 - 0.0328 R_p \tag{4.2}$$

The volumetric efficiency is taken as 100% for all refrigeration technologies, temperature levels and operating refrigerants. Further, pressure drop and heat transfer are neglected in the discharge line for all refrigeration technologies and temperature levels.

In all the refrigeration systems, floating head pressure control strategy is used with temperature differences between condensing and ambient temperatures of 5.6°C (10°F) [34] for both the MT and LT temperature packs, while the minimum condensing temperature for each pack is fixed at 10°C [77]. Eq.(4.3) summarizes the determination of the condensing temperature:

$$T_{cd}\,[°C] = \begin{cases} 10 & \text{if } (T_{amb} + 5.6) \leq 10°C; \\ T_{amb} + 5.6 & \text{if } (T_{amb} + 5.6) > 10°C; \end{cases} \tag{4.3}$$

Ge and Tassou [77] identified that the optimun high side pressure of the booster CO_2 system is largely dependent on ambient air temperature, compressor performance characteristics and the effectiveness of the suction line heat exchanger. According to the authors, when ambient air temperature is low, all the CO_2 booster cycle operates in subcritical mode, which requires the control strategy for the high side pressure to be designed correspondingly. Further, the transition point for subcritical and transcritical cycles is considered at ambient temperature of 21°C [77]. Eq.(4.4) describes the general control strategy of high side pressure for supermarket CO_2 booster systems, according to Ge and Tassou [77]:

$$P_{gc}\,[kPa] = \begin{cases} 4497 & \text{if } T_{amb} < 0°C; \\ 135.2\, T_{amb} + 4434 & \text{if } 0 \leq T_{amb} \leq 20°C; \\ 7205 & \text{if } 20 < T_{amb} < 21°C; \\ 7500 & \text{if } 21 \leq T_{amb} \leq 27°C; \\ 234.26\, T_{amb} + 1154.1 & \text{if } T_{amb} > 27°C. \end{cases} \tag{4.4}$$

Kazachki [34] proposes a subcooling of 0°C in the condenser (or gas cooler) for all systems and temperature levels. Pressure drop is neglected in both medium and low temperature applications for DX's condenser and evaporator, pumped CO_2's refrigerant condenser and evaporator, and CO_2

booster's gas cooler (or condenser) and evaporators.

The impact of heat gains or losses in the liquid refrigerant lines on subcooling, according to Kazachki [34], are to be neglected for both the DX and pumped CO_2 refrigeration systems. Temperature changes in liquid lines before and after the heat exchanger are also neglected for the CO_2 booster [77]. The effect of pressure drop in liquid line follows the same pattern.

According to Kazachki [34], heat gains in DX and pumped CO_2 return lines are to be neglected, as well. For the CO_2 booster, heat gains are also neglected in the suction line before and after the heat exchanger, as suggested by Ge and Tassou [77]. A compressor return gas temperature of 7.2°C (45°F) is considered for the DX refrigeration system, with a useful superheat of 5.6°C (10°F) assumed in the pumped CO_2 refrigerant evaporator [34]. For the CO_2 booster, a useful superheat of 5.6°C (10°F) is also proposed for both the medium and low temperature evaporators. Useful superheat in the DX refrigerated fixtures is 2.8°C (5°F) for medium temperature applications, and 8.3°C (15°F) for low temperature ones, according to Kazachki [34].

A pressure drop of 1.1°C (2°F) is taken for the DX low temperature system and medium temperature system suction lines, whilst in the pumped CO_2 medium temperature system and low temperature system, the pressure drop is neglected due to the short return lines and the downstream movement of oil [34]. For the CO_2 booster, a pressure drop of 1.1°C (2°F) is considered in the suction line after the heat exchanger, while in the suction line before the heat exchanger the pressure drop is neglected.

In the pumped CO_2 refrigeration system, for both medium temperature and low temperature applications, an efficiency (including electric motor efficiency) of 60% is assumed for the circulation pump, from Kazachki [34]. The pressure drop in the secondary coolant evaporator is considered as 3.9°C 7°F), whereas heat gains or losses and pressure drop are neglected in the high pressure liquid line and the low pressure vapor line [34]. No useful superheat is considered in the secondary fluid evaporator, whilst no subcooling is applied at the secondary fluid condenser, as proposed by Kazachki [34]. The temperatrure difference in the intermediate heat exchanger is assumed to be 5.6°C (10°F) [34].

Regarding the CO_2 booster refrigeration system, according to Danfoss [70], receiver control is generally inexistant in refrigeration systems. A simple investigation of the intermediate pressure showed that it must be as low as possible, in order to reduce the amount of liquid in the gas bypass line. Further, the receiver pressure is constant regardless of the ambient temperature. Danfoss [70] suggested the receiver pressure to be set at 3000 kPa. Accordingly, Ge and Tassou [77] also defined 3000 kPa as the intermediate pressure for the CO_2

booster when evaluating its performance. Thus, the value is also considered for the CO_2 booster here analyzed. Moreover, pressure drop in the high and medium pressure sides of the intermediate heat exchanger is neglected, as well as any heat gain for the superheated vapor.

A summary of the key conditions for each refrigeration technology is presented in Table 4.1, for direct expansion system, Table 4.2, for pumped CO_2 system, and Table 4.3, for CO_2 booster system.

Table 4.1: Input data for direct expansion refrigeration system analysis

Parameter	MT	LT
Cooling load	250.89 kW	87.921 kW
Compressor isentropic efficiency	0.65	0.65
Compressor volumetric efficiency	1.00	1.00
Temperature change in discharge line	0°C	0°C
Pressure drop in discharge line	0 kPa	0 kPa
Condensing temperature	Eq.(4.3)	Eq.(4.3)
Condenser outlet subcooling	0°C	0°C
Pressure drop at condenser	0 kPa	0 kPa
Temperature change in liquid line	0°C	0°C
Pressure drop in liquid line	0 kPa	0 kPa
Evaporating temperature	−5.6°C	−27.8°C
Evaporator outlet superheat	2.8°C	8.3°C
Pressure drop at evaporator	0 kPa	0 kPa
Compressor suction temperature	7.2°C	7.2°C
Drop of sat. temperature in suction line	1.1°C	1.1°C

4.4 Coefficient of performance

The performance of the refrigeration technologies operating with distinct refrigerants and in different geographic locations is firstly evaluated by means of the coefficient of performance (COP). Since two temperature levels, medium and low, are being considered for each technology, reflecting a realistic supermarket [34], and in DX and pumped CO_2 such condition requires two separate systems, the COP expression is presented for each refrigeration cycle.

For the DX refrigeration system, coefficient of performance is determined by the ratio between cooling load in the medium temperature system plus cooling load in the low temperature system and compressor power consumption in the medium teperature system plus compressor power consumption in the low temperature system. Eq.(4.5) expresses the COP of the DX technology operating in medium and low temperature levels.

Table 4.2: Input data for pumped CO_2 refrigeration system analysis

Parameter	MT	LT
Compressor isentropic efficiency	0.65	0.65
Compressor volumetric efficiency	1.00	1.00
Temperature change in discharge line	0°C	0°C
Pressure drop in discharge line	0 kPa	0 kPa
Refrigerant condensing temperature	Eq.(4.3)	Eq.(4.3)
Refrigerant condenser outlet subcooling	0°C	0°C
Refrigerant pressure drop at condenser	0 kPa	0 kPa
Temperature change in liquid line	0°C	0°C
Pressure drop in liquid line	0 kPa	0 kPa
Refrigerant evaporator outlet superheat	5.6°C	5.6°C
Refrigerant pressure drop at evaporator	0 kPa	0 kPa
Temperature change in suction line	0°C	0°C
Drop of saturation temperature in suction line	0°C	0°C
Cooling load	250.89 kW	87.921 kW
Pump isentropic efficiency	0.60	0.60
Temperature rise in high pressure liquid line	0°C	0°C
Pressure drop in high pressure liquid line	0 kPa	0 kPa
Secondary fluid evaporating temperature	−5.6°C	−27.8°C
Secondary fluid evaporator outlet superheat	0°C	0°C
Secondary fluid drop of sat. temperature at evaporator	3.9°C	3.9°C
Temperature rise in low pressure vapor line	0°C	0°C
Pressure drop in low pressure vapor line	0 kPa	0 kPa
Secondary fluid condenser outlet subcooling	0°C	0°C
Difference of temperature in intermediate HX	5.6°C	5.6°C

$$COP_{DX} = \frac{\dot{Q}_{ev,MT} + \dot{Q}_{ev,LT}}{\dot{W}_{cp,MT} + \dot{W}_{cp,LT}} \tag{4.5}$$

In the pumped CO_2 refrigeration system, COP is calculated dividing the sum of the cooling load in the medium temperature system and in the low temperature system by the sum of the compressor and pump power consumption in the medium teperature system and in the low temperature system. The coefficient of performance of the pumped CO_2 technology, considering systems for medium and low temperature applications, is calculated with Eq.(4.6).

$$COP_{Pumped} = \frac{\dot{Q}_{ev,MT} + \dot{Q}_{ev,LT}}{\left(\dot{W}_{cp,MT} + \dot{W}_{pp,MT}\right) + \left(\dot{W}_{cp,LT} + \dot{W}_{pp,LT}\right)} \tag{4.6}$$

Regarding the CO_2 booster refrigeration system, performance is obtained with the ratio between the cooling load in the medium temperature evaporator plus the cooling load in the low temperature evaporator and the power consumption in the high stage compressor plus the power consumption in the low stage compressor. Note that, whilst in the case of DX and pumped CO_2

Table 4.3: Input data for CO_2 booster refrigeration system analysis.

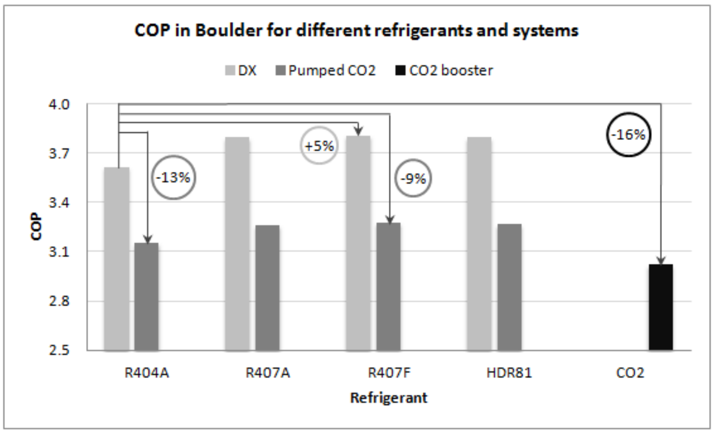
Parameter	Value
Medium temperature cooling load	250.89 kW
Low temperature cooling load	87.921 kW
High stage compressor isentropic efficiency	Eq.(4.1)
High stage compressor volumetric efficiency	1.00
Low stage compressor isentropic efficiency	Eq.(4.2)
Low stage compressor volumetric efficiency	1.00
Temperature change in discharge line	0°C
Pressure drop in discharge line	0 kPa
Gas cooler pressure	Eq.(4.4)
Gas cooler outlet temperature	Eq.(4.3)
Pressure drop at gas cooler	0 kPa
Temperature change in liquid line upstream heat exchanger	0°C
Pressure drop in liquid line upstream exchanger	0 kPa
Temperature change in liquid line downstream heat exchanger	0°C
Pressure drop in liquid line downstream heat exchanger	0kPa
Temperature rise in suction line downstream heat exchanger	0°C
Drop of sat. temperature in suction line downstream heat exchanger	0°C
Temperature rise in suction line upstream heat exchanger	0°C
Drop of sat. temperature in suction line upstream heat exchanger	1.1°C
Suction HX temperature change on superheated vapor side	0°C
Transcritical refrigerant pressure drop at suction HX	0 kPa
Superheated vapor pressure drop at suction HX	0 kPa
Receiver pressure	3000 kPa
Medium evaporating temperature	−5.6°C
Medium temperature evaporator outlet superheat	5.6°C
Pressure drop at medium temperature evaporator	0 kPa
Low evaporating temperature	−27.8°C
Low temperature evaporator outlet superheat	5.6°C
Pressure drop at low temperature evaporator	0 kPa

cycles different systems are required to access the two temperature levels, for CO_2 booster one sole system can supply both medium and low temperature requirements. Thus, Eq.(4.7) can be used to determine the COP for the CO_2 booster technology.

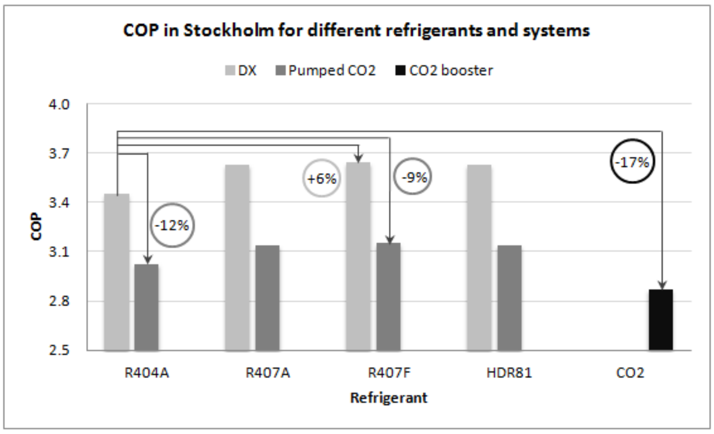
$$COP_{Booster} = \frac{\dot{Q}_{ev,mt} + \dot{Q}_{ev,lt}}{\dot{W}_{cp,hs} + \dot{W}_{cp,ls}} \tag{4.7}$$

Figures 4.2 and 4.3 show predictions in the coefficient of performance by geographic location, refrigeration technology and system temperature level. Table D.2 in Appendix C provides a more detailed set of results table organized by location, technology and temperature level.

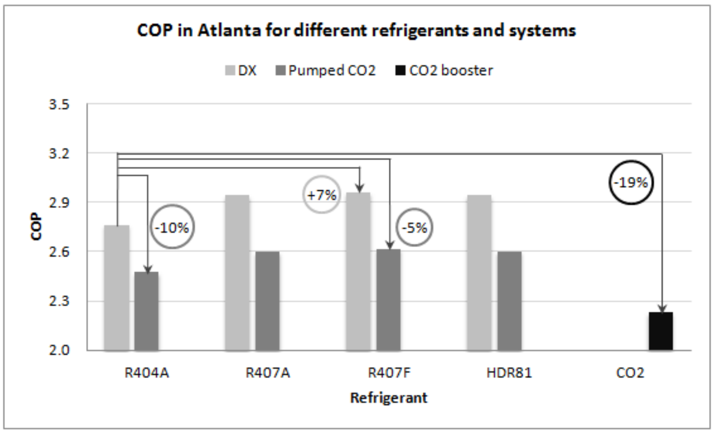
Comparing the three technologies in terms of cycle performance, it becomes apparent that the direct expansion system has superior COP in all



4.2(a): Boulder, USA

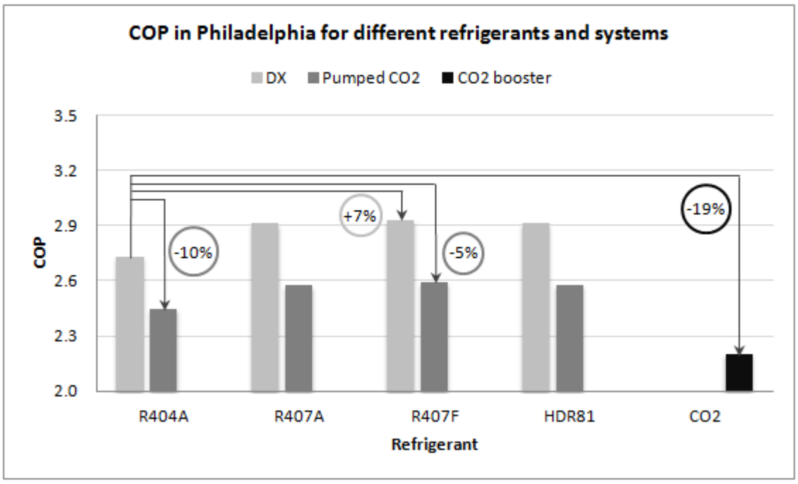


4.2(b): Stockholm, Sweden

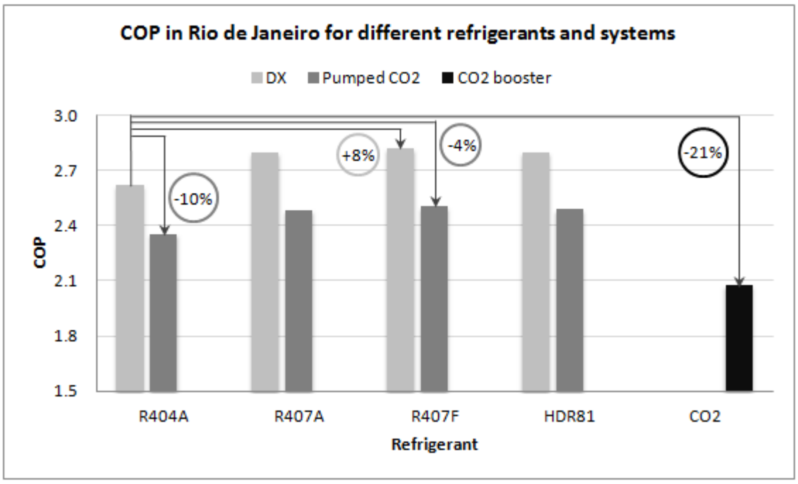


4.2(c): Atlanta, USA

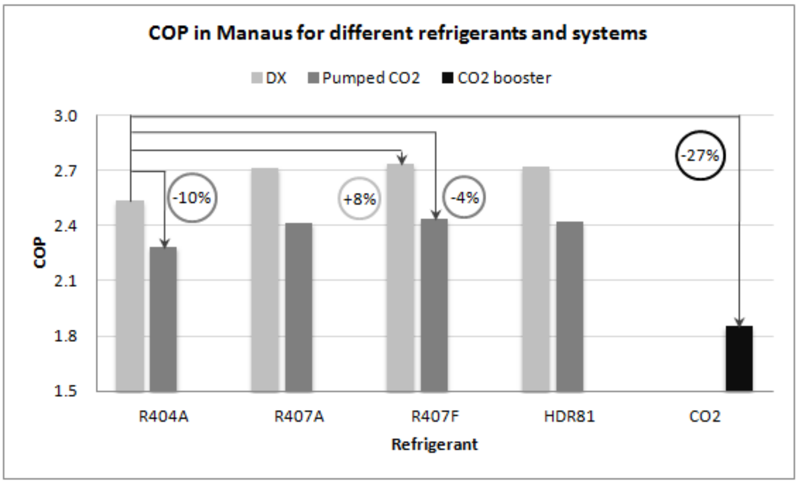
Figure 4.2: Part 1 of the COP analysis. The first percentage, left side, refers to the difference in COP between DX and pumped CO_2, both systems operating with R404A as main fluid. The second percentage is the difference between the DX system COP operating with R404A and R407F. The third percentage is the difference between the COPs of R404A DX and pumped CO_2 with R407F as primary-cycle refrigerant. The last percentage, right side, refers to the difference in COP between the R404A DX and the CO_2 booster technologies.



4.3(a): Philadelphia, USA



4.3(b): Rio de Janeiro, Brazil



4.3(c): Manaus, Brazil

Figure 4.3: Part 2 of the COP analysis of supermarket refrigeration technologies operating with distinct refrigerants in different locations. Presentation of percentage values follows the same pattern described in Figure 4.2.

geographic locations, if compared to the pumped CO_2 and the CO_2 booster, regardless of the working fluid considered.

Replacing R404A in the DX with any of the blends impacts positively the COP, though R407F is the mixture for which the rise in efficiency is the greatest, increasing from 5% in locations with low mean ambient temperatures (Boulder, Figure 4.4(b)) to 8% in warmer climates (Rio de Janeiro and Manaus, Figures 4.5(b) and 4.5(c), respectively). For the pumped CO_2 technology, R407F is also the best choice as main cycle refrigerant, with trends similar to that of the DX systems.

Secondary-loop technologies similar to the pumped CO_2 have worse cycle performance in geographic locations with colder mean ambient temperatures, if compared to DX cycle. In Boulder and Stockholm, mean ambient temperatures lower than 17°C, the COP of the pumped CO_2 system operating with R404A is 13% and 12% lower than that of the R404A DX, Figs. 4.2(a) and 4.2(b). Considering the pumped CO_2 with R407F as primary fluid, both percentages are reduced to 9%. For warmer regions, where mean ambient temperatures are higher than 24 °C, the difference between COPs of pumped CO_2 and DX systems COP is 10%, when both systems have R404A as the main fluid, and 4%, when R407F is selected for the pumped CO_2 primary cycle and R404A for the traditional DX, Figures 4.3(b) and 4.3(c).

In regard to the CO_2 booster refrigeration system, contrarily to the secondary-coolant technology, higher COPs are verified in geographic locations with low mean ambient temperature, with the performance in Boulder being only 16% inferior to that of the R404A DX circuit, Figure 4.2(a)). On the other hand, in warmer conditions like that of Manaus, the COP of the CO_2 booster technology is 27% lower than the DX system, Fig. 4.3(c)).

In order to better compare the technologies and refrigerants, an analysis based on annual energy consumption follows.

4.5 Annual energy consumption

Annual energy consumption is calculated based on the power input and weather bin data [34]. The total annual energy consumption corresponds to the total energy consumption in all bins. To calculate the energy consumption in each bin, the system total power per bin is multiplied by the number of hours in that bin.

In the DX refrigeration system, the total power is the sum of compressor power consumption in the medium temperature system and in the low temperature system. Regarding the pumped CO_2 technology, total consumption is obtained with the sum of compressor and pump power consumption in the medium temperature system and in the low temperature system. For the CO_2 booster refrigeration circuit, power input is the sum of high stage compressor and low stage compressor power consumption.

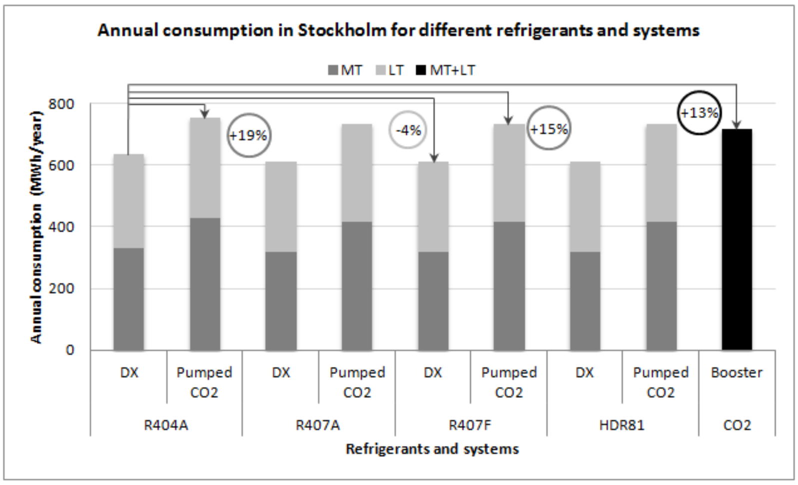
Figures 4.4 and 4.5 show patterns in the annual energy consumption by geographic location, refrigeration technology and system temperature level. Table D.1, in Appendix D, provides a more detailed set of results table organized by location, technology and temperature level.

Analysing Figures 4.4 and 4.5, one can observe that in all geographic locations, for a specific operating refrigerant, the direct expansion refrigeration system has the lowest energy consumption, if compared to pumped CO_2 and CO_2 booster. Regarding the DX system working fluid, although all three blends present lower consumption than R404A, R407F is the one with the best performance in all climate conditions. In colder regions, as Stockholm, the impact of replacing R404A with R407F is less significant, with annual consumption down only 4%, Figure 4.4(a). In Manaus, where fewer hours of low ambient temperatures are verified, the reduction in annual consumption, when using R407F instead of R404A, can reach 8%, Figure 4.5(c). Considering the selection of the pumped CO_2's main cycle refrigerant, a similar trend is observed when moving between different climate conditions.

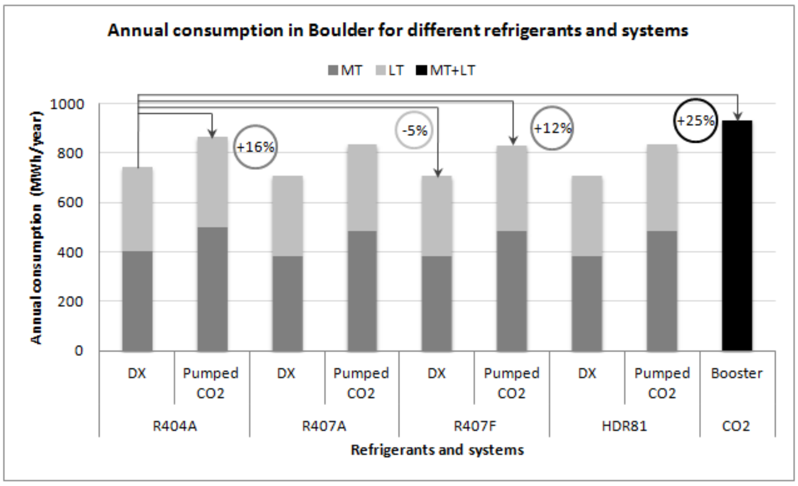
In Stockholm and Boulder, geographic locations with colder climates, the pumped CO_2 technology has an annual energy consumption 19% and 16% higher than that of the direct expansion, respectively, when both systems are operating with R404A, Figures 4.4(b) and 4.4(a). If refrigerant R407F is considered as the working fluid in the primary cycle of the pumped CO_2, the difference between the annual consumption of the technologies, in the same locations, is reduced to 15% and 12%, respectively.

In the climate conditions of Philadelphia and Atlanta, the difference between annual consumption of the pumped CO_2 and the DX systems, both operating with R404A, is lower, if compared to that of Stockholm and Boulder, representing, respectively, 15% and 14%, Figures 4.4(c) and 4.5(a). Considering R407F as a primary-cycle refrigerant in the CO_2 indirect technology, the variation in energy consumption is smaller, only 11% to 9% lower than that of the direct expansion system.

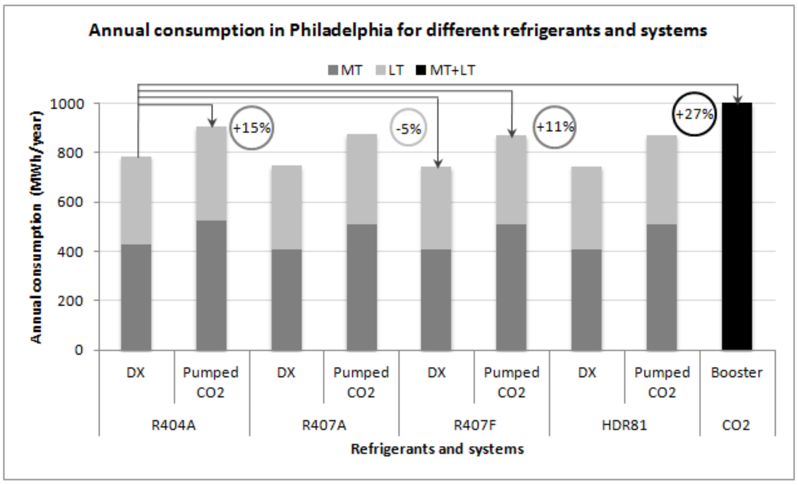
Secondary-coolant technologies with arrangement similar to that of the pumped CO_2 refrigeration system perform the best in geographic locations with warmer climates and small variation of temperatures along the year, as it can be seen in Figures 4.5(b) and 4.5(c). In Rio de Janeiro and Manaus, the energy consumption of the secondary CO_2 system operating with R404A



4.4(a): Stockholm, Sweden

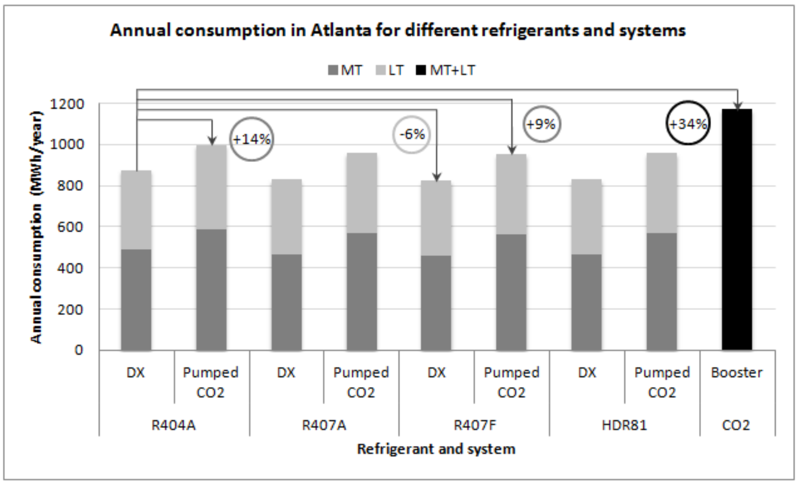


4.4(b): Boulder, USA

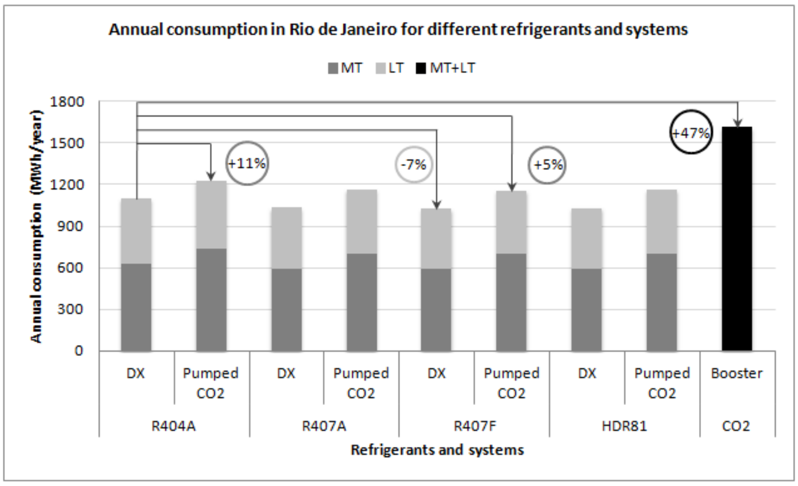


4.4(c): Philadelphia (PA), USA

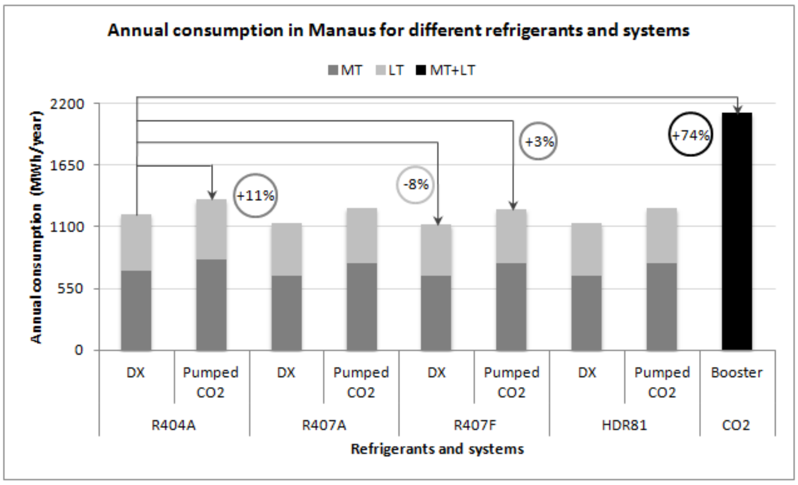
Figure 4.4: Part 1 of the annual consumption analysis of refrigeration technologies operating with distinct refrigerants in different locations. Presentation of percentage values follows the same pattern described in Figure 4.2, though referring, here, to annual energy consumption.



4.5(a): Atlanta, USA



4.5(b): Rio de Janeiro, Brazil



4.5(c): Manaus, Brazil

Figure 4.5: Part 2 of the annual consumption analysis of refrigeration technologies operating with distinct refrigerants in different locations. Presentation of percentage values follows the same pattern described in Figure 4.2, though referring, here, to annual energy consumption.

is only 11% higher than that of the R404A DX. The pumped CO_2 solution performs even better when operating with R407F as primary refrigerant: its annual energy consumption is only 5% and 3% higher than that of the R404 direct expansion system, respectively, for the same locations.

Regarding improvements for the energy consumption of pumped CO_2 technology, Kazachki and Hinde [23] suggested that the reduced pressure drop and the reduced heat transfer through the short insulated pipe lines allowed for taking full advantage of the floating condensing pressure. Whereas in a DX refrigeration system the liquid refrigerant lines may consist of kilometers of pipe, in secondary-coolant systems the same lines are only a few meters to a few tens of meters [23]. Moreover, the use of electronic expansion valves in DX circuits is cost-prohibitive given the large number of required valves and associated electronics, whilst secondary-coolant technologies work well with electronic expansion valves which don't require seasonal readjustment when operating over a wider range of condensing pressures, as reported by Kazachki and Hinde [23].

Regarding the CO_2 booster, the influence of climate conditions in the power consumption is characteristically different. In geographic locations with a large number of hours of low ambient temperatures, such as Stockholm, the difference between annual energy consumption of the CO_2 booster and the DX systems is the lowest: 13%, Figure 4.4(a). According to Ge and Tassou [77], the better performance of the CO_2 system at low temperatures is due to the operation in subcritical cycles, with a largely reduced compression ratio.

In Atlanta and Philadelphia, where a broader temperature variation accross the year is verified, the energy consumption of CO_2 booster refrigeration systems is, respectively, 27% and 34% higher than that of R404A DX systems, Figures 4.4(c) and 4.5(a)). In such climates, the special features of the booster technology have less of an impact if compared to the application in colder conditions.

In warmer climates, such as Manaus, from an energy point of view, the use of CO_2 booster refrigeration system can be counterproductive, with annual consumption 74% superior to the R404A DX technology, Figure 4.5(c). As reported by Ge and Tassou [77], during periods of high ambient temperatures, the higher pressure and transcritical operation can lead to higher energy consumption for the CO_2 booster, when compared to the DX system operating in medium and low temperature applications.

However, the implementation of advanced control techniques can result in improvement of the efficiency of CO_2 cycles, according to Ge and Tassou [77]. Making maximum utilization of pressure control reduces significantly heat recovery opportunities for the R404A DX system, whilst the booster CO_2 lends itself for heat recovery even during subcritical conditions, mainly due to higher cycle pressures and temperatures [77]. Regarding such heat recovery potential of the CO_2 booster refrigeration system, simulations for a specific supermarket show [77] that energy recovery can provide 40% energy savings in the space heating energy requirement, if compared to a R404A DX system operating with floating head pressure control and no heat recovery.

In reality, as discussed in Kazachki [34], the decision of which technology to select also depends on consideration of conditions such as ease and cost of installation, operation and maintenance, and the supermarket's established practices.