1
Introduction

In the present dissertation, new and traditional commercial refrigeration systems were compared in terms of their energy efficiency and environmental impact. First, thermodynamic models were developed to simulate and compare the pumped CO_2 and the CO_2 booster cycles with the conventional supermarket direct expansion system. Performance was evaluated by means of the coefficient of performance (COP) and the annual energy consumption, with R404A, CO_2 and new low-GWP non-azeotropic blends considered as working fluids. Second, a lumped parameter model was developed to simulate the steady-state operation of a multi-compressor multi-evaporator direct expansion refrigeration system. A component-based solution method was employed, as well as multizone descriptions for the tube-and-fin heat exchangers. The environmental impact of the technology, operating with R404A and new low-GWP refrigerants, was measured in terms of the life cycle climate performance (LCCP).

1.1
General overview

1.1.1
Commercial refrigeration

According to Bansal [1], refrigeration plays an essential role in conserving food and granting thermal confort for the contemporary society. The International Institute of Refrigeration (IIR) reported [2] that 15% of all electricity consumed worldwide is associated with refrigeration. Hence, it is of most importance to obtain improved energy efficiency in operations of the kind, as a matter of reducing costs and environmental impact, as well as providing food quality [1].

Commercial refrigeration is, according to Arthur D. Little (ADL) [3], a broad equipment category, including central refrigeration systems for supermarkets, walk in refrigerators and freezers, ice machines, and a wide variety of self-contained refrigerated and frozen food display cases, beverage merchandisers and vending machines. Refrigerant emissions from the commercial domain are, as addressed by Girotto et al. [4], relatively high, with a study from Norway estimating that yearly refrigerant leakage from refrigeration systems represents 30% of the total original charge of refrigerant [5]. Globally, commercial refrigeration is the field with the largest refrigerant emissions, speaking for 40% of the total refrigerant emissions [6, 7]. The U. S. Department of Energy (DOE) reported [8] that, in 2006, commercial buildings consumed about 18% of the total primary energy used, Figure 1.1. In addition, Figure 1.2, provided by Navigant Consulting [9], shows a breakdown of the total power consumption by the commercial equipment segment. Supermarket refrigeration systems, which comprise compressor racks, display cases, condensers and walk-ins, account for half of the total energy used in commercial refrigeration.

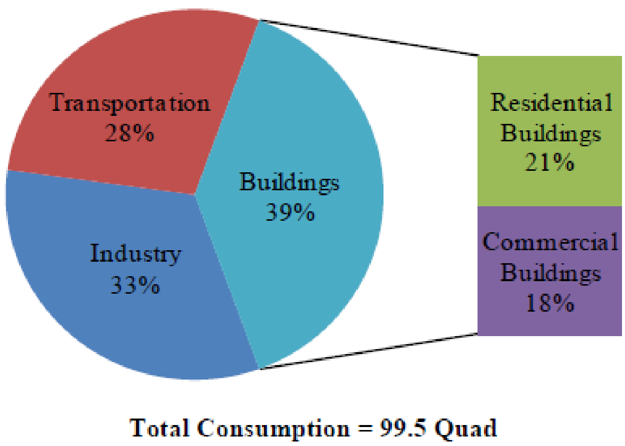


Figure 1.1: U. S. Primary energy consumption by sector in 2006, provided by the U. S. Department of Energy [8].

Supermarket refrigeration is the field with the largest refrigerant emissions, due to the thousands of fittings in the cooling systems, the large area where the refrigerant is moved and the high refrigerant charges [10]. Supermarkets are also one of the most energy-intensive types of commercial complexes [11]. In respect to this subsector, according to Billiard [12], power consumption ranges between 400 and 1,000 kWh/m^2/year. Lundqvist [13] suggested, additionally, that between 35 and 50% of the energy used in supermarkets is destined to the cooling plant. Electricity consumption in large supermarkets is estimated to be 4% of the national electricity use in the U. S. and France, according to Orphelin and Marchio [14]. Further, Baxter [15, 16] reported that typical supermarkets in the U. S., with sales area ranging between 3,700 and 5,600 m^2 approximately, respond for the consumption of about 2 to 3 million kWh annually for total store energy use.

A typical store in the U. S. uses about 39% of its total electricity consumption for medium and low temperature refrigeration, as presented in

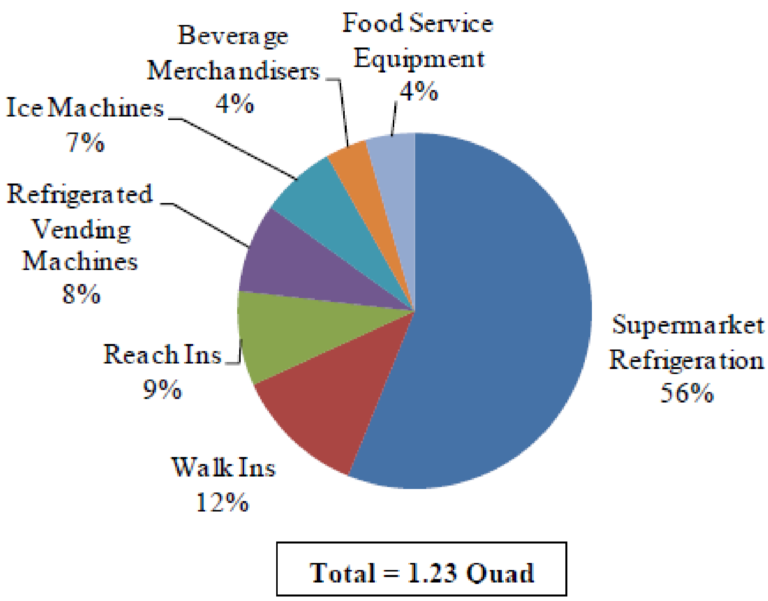


Figure 1.2: Annual power consumption of commercial refrigeration equipment, from Navigant Consulting [9].

Figure 1.3, provided by Energy Star [17]. A breakdown of the energy usage from this supermarket shows that, apart from the 39% consumed by the refrigeration equipment, 28% of the energy is used for fans and climate control, 23% for illumination, 5% for cooking, 3% for water heating and 3% for other uses (Arias [18]). Combining all these statistics with the vast expansion of supermarket complexes, which already cover 70% of the food sales market in Europe [7], it becomes clear why these commercial buildings are currently recognized as important power demand agents.

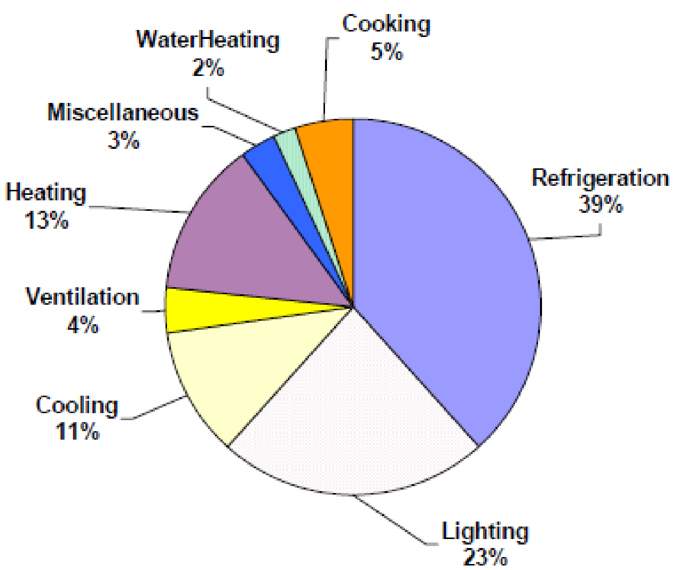


Figure 1.3: Typical electricity use of a store in the U. S. [17].

Baxter [15] describes a typical supermarket layout, as represented in Figure 1.4. Due to the large amount of perishable food products, refrigerated fixtures, which can be display cases or walk-in storage coolers, are located throughout the store. Display cases provide a temporary place not only to refrigerate food products, but also to merchandise them, whilst walk-in coolers are designed to store food products during that time period between receiving and putting the goods out for sale (Walker et al. [19]). The refrigeration of the display cases and walk-in coolers is done by a large vapor compression refrigeration system. Air coils are located in each case or cooler, with refrigerant being supplied to and removed from the coil by refrigerant piping. Compressors and condensers are located either in the machine room, usually in a remote part of the store, or on the roof of the building, above the machine room [19].

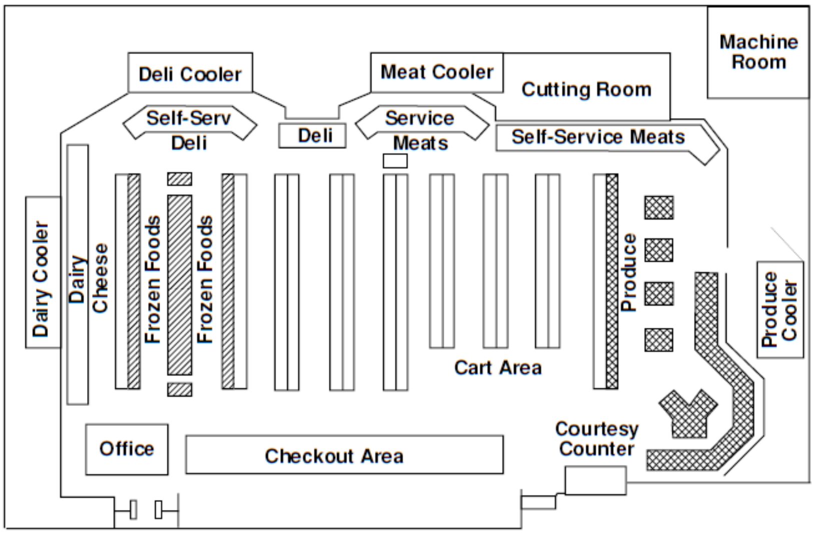


Figure 1.4: A representative supermarket layout, from Baxter [15].

According to Walker et al. [19], the cooling load requirements of display cases are key aspects to the design and operation of supermarket refrigeration systems. They not only greatly influence the power consumption of the commercial complex, but also play a major role in specifying the amount and size of condensers and compressors that must be installed [19]. The operating parameters of the vapor compression cycle responsible for providing cooling loads for the display cases is associated with maintaining the cases at a set temperature. Walker et al. reported [19] that display cases are maintained at air temperatures ranging from -32 to 2°C, depending on the type of good stored. There are two principal temperature levels in supermarkets: medium temperature (MT), for preservation of chilled food, and low temperature (LT), for frozen products [18].

The refrigeration load of the case, which is often given at a particular discharge air temperature, designates the thermal energy that should be removed as to mantain the product at the desired storage temperature. This thermal energy comes from a number of different sources (Walker et al. [19]): heat conduction through the walls; ambient air entrainment (both sensible and latent heat); heat exchange by radiation between the surrounding environment and the interior of the display case; and the heat load generated by the use of electricity in the display case for illumination of the product, operation of anti-sweat heaters and fan motors to circulate air around the case.

Figure 1.5: Open vertical display cabinet from a typical supermarket, provided by Arias [18].

Storage applications for display cases, as well as typical operating temperature ranges for food products, are listed in Table 1.1, adapted from Walker et al. [19]. The control of the food storage temperature is set by the discharge air temperature, i.e., the temperature of the air exiting the display case evaporator, as referred to in Table 1.1.

Walker et al. [19] reported that an independent survey of stores in southern California revealed that open multi-deck, medium temperature display cases represent roughly half of the refrigerated fixtures in a store, Figure 1.6. Additionally, Navigant Consulting verified [9] that medium temperature fixtures and storage coolers account for about 70 to 75% of the total supermarket cooling load, with open multi-deck cases corresponding to approximately 3/4 of that fraction. Figure 1.7 illustrates the results of Faramarzi [20], who verified that infiltration corresponds to the biggest component of refrigeration load in medium temperature open vertical display cases.

Table 1.1: Typical temperature requirements for major food storage applications, adapted from Walker et al. [19].

Temperature level	Storage application	Discharge air temperature (°C)
Medium (MT)	Beverage	2
	Deli	−0.5 to 1
	Produce	−0.5 to 1
	Dairy	−1 to 1
	Meat/seafood	−3 to −2
Low (LT)	Frozen food	−26 to −19
	Ice cream	−32 to −24

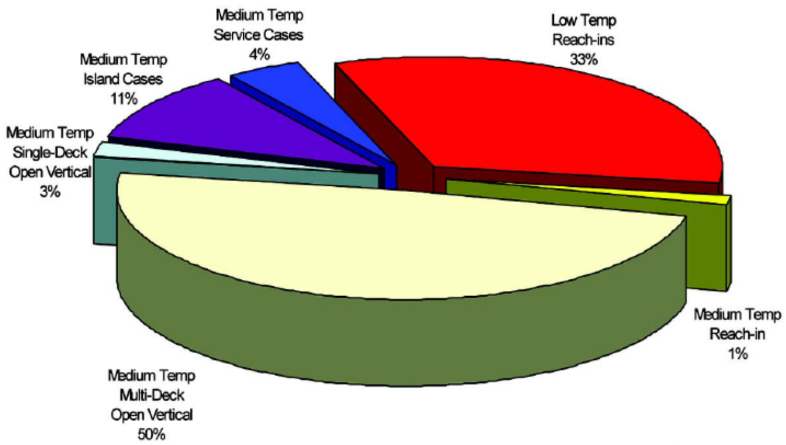


Figure 1.6: Breakdown of refrigerated fixture lineal meter by type in a typical supermarket (Walker et al. [19]).

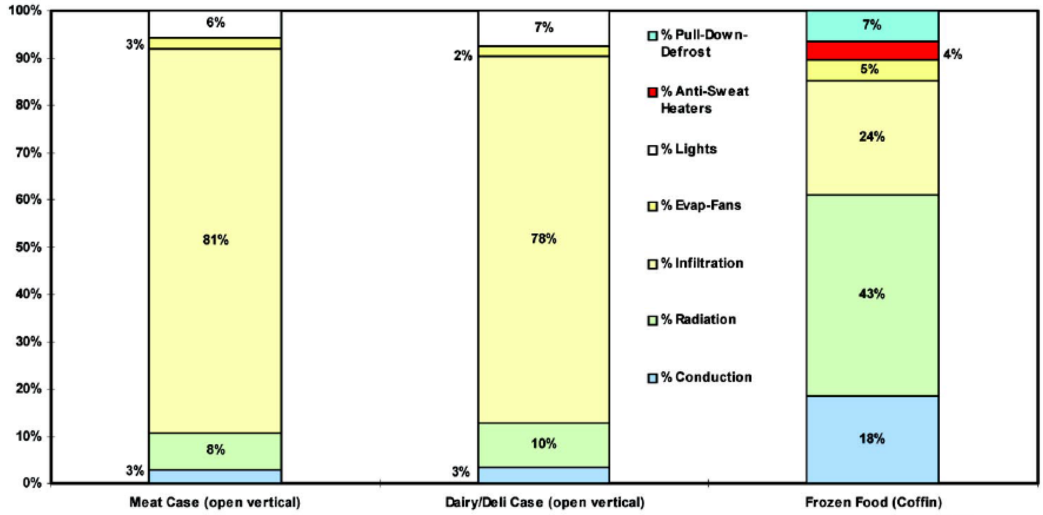


Figure 1.7: Breakdown of cooling load for meat, dairy, deli and frozen food fixtures, from Faramarzi [20].

Frost formation on the evaporator surface of display cases is an extremelly relevant feature, since, without periodic removel, frost can accumulate and block the airflow passages. Thus, the practice of providing defrost to the cases on a scheduled basis, commonly executed in supermarket refrigeration systems, is fundamental to the appropriate operation of the display case, as when it is not done or done incorrectly, it can damage the product stored [19].

In that sense, there is considerable potential for improvement in energy usage by supermarkets. Energy-saving technologies related to illumination, HVAC system and, in particular, given their central role in supermarket operation, the refrigeration system, have been studied and implemented in several stores to reduce energy consumption [18].

1.1.2 Environmental impact

During recent years, global environmental impacts, such as ozone layer depletion and global warming, have been directly associated with refrigeration systems [18]. In accordance with Bovea et al. [7], environmental problems are transferred to the refrigeration and air conditioning sector by means of two main aspects: the direct effect, due to the emissions of the most commonly used refrigerants, and the indirect effect, caused mainly by energy consumption of cooling facilities.

The concept of life cycle climate performance (LCCP), first proposed by the Technology and Economic Assessment Panel (TEAP) of the United Nations Environment Program (UNEP), calculates the so-called "cradle-to-grave" climate impacts of the direct and indirect greenhouse gas emissions [21]. The basic contributors to LCCP are, according to ADL [3], carbon dioxide emissions due to energy use and the direct warming impact of emissions. In that sense, the refrigeration system design, as well as the fluid selected as refrigerant, play key roles in the environmetal impact evaluation.

1.1.2.1 Refrigerants

Refrigerant selection is, as stated by Peixoto [22], primarily based on thermodynamic properties and other criteria, which include environmental impact, toxicity, safety, cost. The debate about what constitutes the "right choice" of refrigerant goes back to the beginning of refrigeration, and has been intensified in recent years. Currently, environmental impacts of refrigerants leaked into the atmosphere, as addressed by Kazachki and Hinde [23], have been yielding global and local environmental regulations in the supermarket industry that would be unimaginable two decades ago. Further, public awareness has been raised to an extent, due to unknown or potentially negative effects, where new directions are continually introduced, whilst existing regulations are becoming more and more restrictive [23]. Figure 1.8 depicts the evolution of refrigerant usage segmented in different periods.

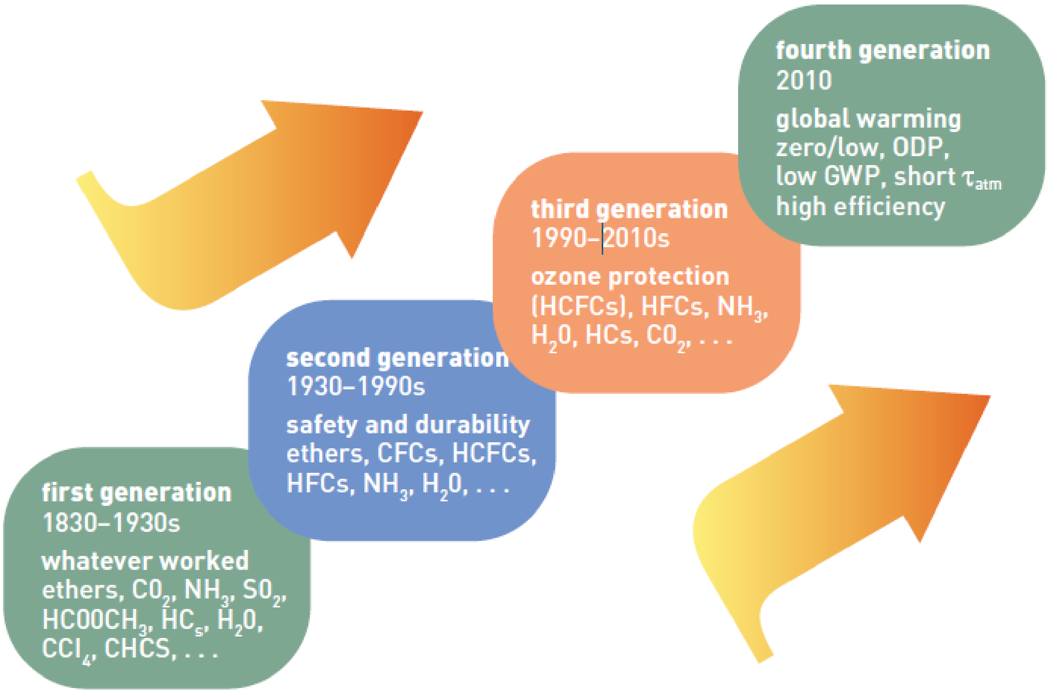


Figure 1.8: Evolution of refrigerants through four different generations, provided by Calm [24].

The first generation of refrigerants comprised, in agreement with Calm [24], familiar solvents and other volatile fluids, effectively including whatever worked and was available. Almost all of these refrigerants were flammable, toxic, or both, and some were also extremely reactive. The second era was characterized by a shift to fluorochemicals for their durability and safety. Chlorofluorocarbons (CFCs) and, later, hydrochlorofluorocarbons (HCFCs), dominated this period. Ammonia continued, though, to be used as one of the most popular refrigerants in large, industrial-scale systems [24].

In agreement with Calm [24], the association of released CFCs to the depletion of protective ozone propelled the third generation, aiming at stratospheric ozone preservation. The Vienna Convention and resulting Montreal Protocol forced the abandonment of ozone-depleting substances (ODSs). Fluorochemicals retained the primary focus, with emphasis on HCFCs and hydrofluorocarbons (HFCs). Eventually these modifications sparked renewed interest in “natural refrigerants” – particularly ammonia, carbon dioxide, hydrocarbons, and water. The majority of developed countries established limits to phase out propellant and blowing agent uses early, on requiring phase-out of

R-22 (the most widely used refrigerant today) by 2010 in new equipment, and the banning of all HCFC use in new equipment by 2020 [24].

The worldwide positive response to ozone depletion stands in contrast to the declining situation with climate change. In recent years, new discoveries and political debate on the subject of global warming have become current events. The Kyoto Protocol set bindings targets for greenhouse gas (GHG) emissions based on calculated equivalents of carbon dioxide, as stated by Calm [24]. Local laws and regulations to implement the Kyoto protocol may vary, though they usually prohibit avoidable releases of HFC and PFC refrigerants, also controlling or taxing, in some countries, their application. Lately, even harsher measures have been forcing shifts to a new generation of refrigerants, defined by attention to global warming [24].

Minor and Wells [25] reported that R404A has been the global industry standard HFC refrigerant for commercial refrigeration in both small and large systems. R404A is a near azeotropic mixture containing R125, R143a and R134a (44/52/4% by weight), non-toxic and non-flammable, as stated in A-Gas [26]. Figure 1.9, provided by Linde [27, 28], presents the most relevant characteristics of R404A.

	R404A	R407A	R407F
Constituents	R143a/R125/R134a	R134a/R125/R32	R134a/R125/R32
Preferred oil	Polyolester (POE)	Polyolester (POE)	Polyolester (POE)
ASHRAE safety classification	A1 – non-toxic & non-flammable	A1 – non-toxic & non-flammable	A1 – non-toxic & non-flammable
Boiling Point @ 1atm	-46.5°C	-45.5°C	-45.5°C
Critical Temperature	72.0°C	82.3°C	82.7°C
Critical Pressure	37.3 bar(a)	45.2 bar(a)	47.5 bar(a)
GWP	3922	2107	1824
GWP as % of R404A	100%	54%	46%

Figure 1.9: Main characteristics of R404A, R407A, and R407F, adapted from Linde [27, 28].

Mixtures such as R404A are traditionally termed non-azeotropic mixtures or, rather, zeotropic mixtures, which implies that, in phase equilibrium, the mass fractions of the vapor and liquid phases are different [29]. The most evident difference between the P-h diagrams of pure fluids and mixtures is that, as shown by Radermacher and Hwang [29], for instance, for distinct mass fractions, all the lines on this diagram incline. Secondly, due to the temperature glide, the isotherms in the two-phase zone present a negative slope.

For over twenty years, R404A has been an extremely useful refrigerant, in particular for applications of commercial refrigeration in supermarkets.

However, although it is non-toxic and has zero ozone depletion potential [21], its high global warming potential [21] is shifting the focus to the identification of low GWP alternatives that can provide similar cooling performance [24]. Replacement blends are, with a few exceptions, often less efficient than early choices, with compressor consumption reduction deriving primarily from improvements in equipment design rather than the properties of the new alternative fluids [24].

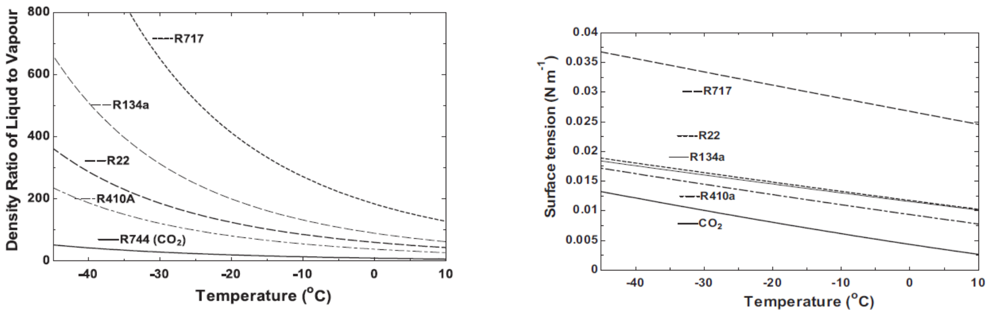
R407A and R407F are two ideal retrofit solutions for many existing R404A systems, according to Linde [27, 28]. Both refrigerants are HFC blends, and have shown to provide higher efficiency than R404A in many systems, besides presenting a reduction of about 50% in the global warming potential. R407A and R407F are non-toxic and non-flammable zeotropic mixtures with moderate glides, whilst also being compatible with the same oil, elastomers and plastics as R404A, which makes the changeover a relatively easy process [27,28].

Widely available on the market for over twenty years, R407A has been approved for use by a large number of compressor manufacturers, reducing the need for compressor heat protection for providing lower discharge temperatures than other R404A replacement candidates [27]. Regarding R407F, although it provides higher discharge temperatures than R404A, energy savings of up to 10% have been reported with many systems [28]. Figure 1.9, adapted from Linde [27, 28], depicts other relevant characteristics of R407A and R407F.

HDR21 and HDR81 are blends developed recently by Honeywell. Both refrigerants are 100% non-ozone depleting and non-flammable, possessing low toxicity, which makes them significantly safer in use than alternatives such as hydrocarbons and ammonia [30]. HDR81, the latest blend developed, can be applied in current R-404A equipment with little to no modification, yet offering GWP reduction of over 65% compared to R-404A, as reported by Yana Motta and Spatz [31]. Further, superior performance (9% better system efficiency) and discharge temperatures below the limits of the compressor were also verified [31] for the system operation with the mixture.

Carbon dioxide (CO_2) has emerged as a mainstream refrigerant to achieve low temperatures in the food and refrigeration industry. Lately, according to Bansal [1], the widespread use of CO_2 refrigerant, especially in supermarkets, has proved to be commercially attractive due to the large number of novel designs being used in the industry, including cascade, transcritical, transcritical booster, secondary loop, among other interesting designs and variations of refrigeration cycles.

When compared to other refrigerants, carbon dioxide presents excellent thermophysical properties at low temperatures. Higher liquid and vapour thermal conductivities, together with lower liquid viscosity and surface tension, make CO_2 an excellent refrigerant for two-phase boiling and condensation applications, as depicted in Figure 1.10 [1]. Nevertheless, pressures in CO_2 systems are typically 5 to 10 times higher than with conventional refrigerants, which turns safety into the major concern regarding its application as a refrigerant. In fact, the rupture of any piping, tubing, vessel or hose containing high pressure carbon dioxide can be fatal, since, if CO_2 concentration in the air exceeds 5%, breathing difficulties, leading to unconsciousness, may take place. Thefore, it is important, before commissioning a carbon dioxide plant, to undertake, as stated by Bansal [1], a "risk assessment" study, making sure that all safety precautions have been met.



1.10(a): Density ratio of liquid to vapor

1.10(b): Surface tension

Figure 1.10: Comparison of thermophysical properties of CO_2 at saturation temperature with those of other refrigerants, from Bansal [1].

1.1.2.2
Refrigeration systems

There is a great potential for the development of efficient systems in supermarkets, with typical efficiency improvements often involving new cycle configurations. Emerson [32] reported that the rate of refrigerant leakage from supermarket systems also varies considerably with the type of installation, hence affecting the design of refrigeration systems in supermarkets. As previously mentioned, a number of novel technologies have been implemented in the commercial sector. In that sense, using alternative systems together with low GWP refrigerants can significantly reduce the direct impact of environmentally emissions, as well as the indirect impact due to energy consumption.

Every supermarket comprises, according to Fischer [33], two main refrigeration systems: one is associated with low temperatures, whilst the other is related to medium temperature applications, as listed in Table 1.1.

Centralised direct expansion refrigeration system. The centralised direct expansion (DX) system, encompassing two completely separate systems for low temperature (LT) and medium temperature (MT) loads, is, by far, the most common refrigeration technology in supermarkets throughout the world today [32]. A typical configuration of the system is illustrated in Figure 1.11, provided by Emerson [32]. Each system uses a central "multi-compressor pack" consisting of three to eight semi-hermetic scroll or reciprocating compressors, placed in a separate machine room, either at the back of the store, inside or outside the building, or even on its roof, in order to reduce noise and prevent costumer access [32, 34]. These multiple compressor racks usually operate at different suction pressures, so as to support the distinct evaporating temperatures at which cases, coolers and freezers are set. Condensers are usually air-cooled and, therefore, located outside. Liquid and suction lines run throughout the supermarket, feeding refrigerant to the refrigerated fixtures and returning vapor to the compressor pack, according to Kazachki [34]. The condenser pressure is typically controlled by varying the air flow, a measure which ensures that the condensing pressure is kept as low as possible [32].

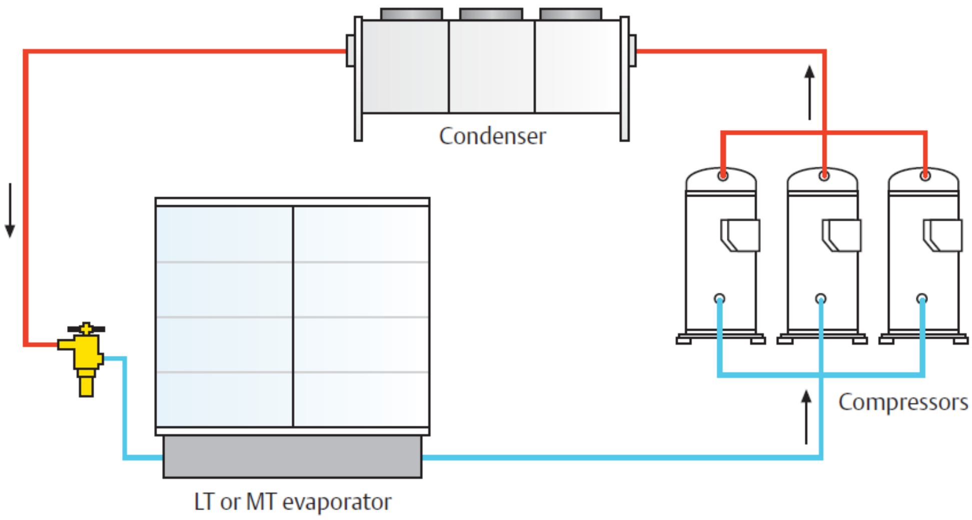


Figure 1.11: Schematic design of a centralised direct expansion (DX) refrigeration system, from Emerson [32].

During the 1980s, according to Bivens and Gage [35], centralised direct systems had annual refrigerant emissions in the range of 20–35% of the refrigerant charge. In the last thirty years, due to international and national regulations, more efficient systems and personnel training, annual emissions have decreased to approximately 10% of charge in new installations [18].

Distributed direct expansion refrigeration system. Unlike the centralised DX refrigeration system, which presents a central room containing multiple compressor racks, distributed systems use multiple smaller rooftop units, mounted within each condenser unit and connected to the refrigerated fixtures, resulting in considerably less refrigerant lines [32,34]. Each unit is located near the display case they serve (on the roof, behind a nerby wall or close to the cabinet in the sales area) [34]. Condenser units can be factory-assembled, allowing their quality to be carefully controlled and integrating the construction process [32]. Figure 1.12 provides the schematic design of a distributed DX system.

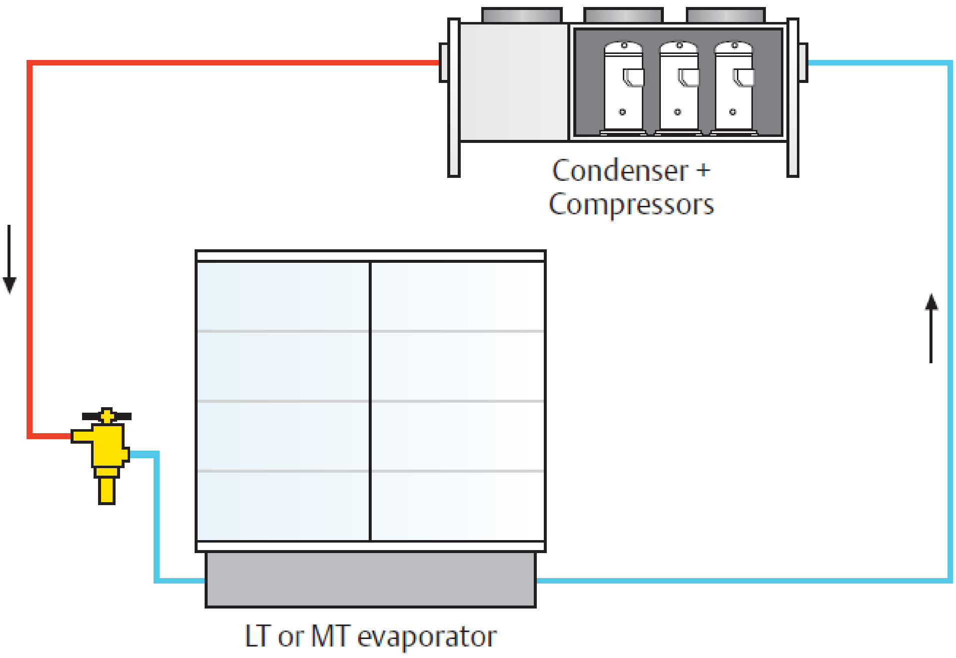


Figure 1.12: Schematic design of a distributed DX refrigeration system, from Emerson [32].

Distributed refrigeration systems usually use refrigerant charge smaller than centralised DX systems, while also reducing the leakage rate as a result of smaller diameter pipe sizing. However, their installation is not always practical, with the structure and location of the commercial complex playing a key role in the decision [32, 34].

Secondary coolant refrigeration system. A secondary coolant refrigeration system, presented by Emerson [32], is schematized in Figure 1.13. Heat from cases and coolers is transferred to the evaporators by circulating a secondary fluid, typically an aqueous solution, glycol, CO_2 or other heat transfer fluid.

These systems require a pump to circulate the coolant around the store area. Further, there is an extra heat exchanger, which demands a temperature difference to drive the heat transfer. Thus, the evaporating temperature must be lower than the secondary fluid temperature, leading to an increase in compressor power consumption [32].

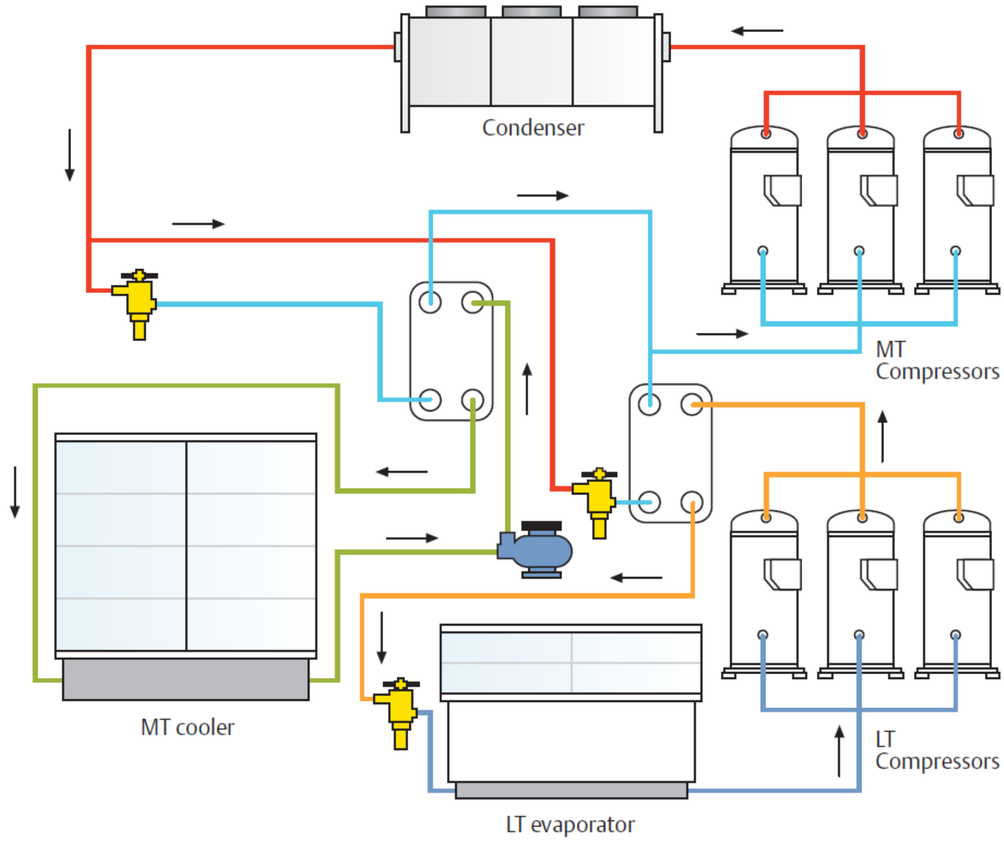


Figure 1.13: Schematic design of a Secondary Coolant refrigeration system, from Emerson [32].

According to Wang et al. [36], there are two categories of secondary refrigerants: single-phase and two-phase fluids. Antifreeze solutions, corrosion inhibitors and biocides are the most popular choices for single-phase refrigerants. Two-phase secondary fluids, however, take advantage of the high latent heat during the phase change process [36]. When carbon dioxide is utilized as a two-phase secondary coolant, the refrigeration cycle is called pumped CO_2 system. As it happens with general secondary coolant technologies, the pumped CO_2 system can take many forms, as reported by Sawalha [37].

The balance between the advantages and penalties of the secondary coolant technology is key to its successful implementation in commercial buildings. On the one hand, the presence of a circulating pump and an intermediate heat exchanger draw extra energy [23]. On the other hand, in accordance with Emerson [32], since heat exchangers can be placed near the compressors, when assembled with the condenser, this package can be delivered as a manufactured unit similar to an air conditioning chiller. As a consequence, a significantly reduced refrigerant charge and leakage is present.

Additional benefits of the secondary coolant technology are, as reported by Kazachki and Hinde [23]: improved product quality and reduced shrink in fresh foods; opportunity to apply more efficient and environmental-friendly (yet with flammability issues) refrigerants; and reduced demand and dependence on qualified technicians for installation and operation. Further, the short liquid supply lines and vapor return lines result in reduced heat exchange between refrigerant and ambient, controlled vapor superheat, and negligible pressure drop in these lines. The authors [23] even suggested that secondary coolant systems consist in the only supermarket refrigeration technology to substantially reduce refrigerant charge and to achieve the coveted zero-leak condition.

Cascade refrigeration system. With the cascade technology, illustrated in Figure 1.14, a centralised DX system is used for medium temperature loads, whilst the low temperature applications have a separate circuit that rejects its heat into the suction stage of the MT system [32]. In fact, since the LT cycle has a low condensation temperature, carbon dioxide alone or combined with a brine may be applied in subcritical mode without exorbitant pressures [1, 32].

Regarding the benefits of the cascade system, since the lower boiling point refrigerant has a higher saturation pressure at low temperatures, the ingress of air is kept under control. In addition, smaller compressors are required for the same refrigeration effect, given the high density of suction vapor [1].

In contrast, the temperature difference necessary to drive the heat transfer across the extra heat exchanger produces minor losses in energy efficiency, if compared to a DX system [32]. Further, cascade refrigeration systems tend to involve a complex electronic and refrigeration control system [1].

Transcritical CO_2 refrigeration system. According to Bansal [1], this configuration was the natural progression from the first subcritical cascade cycles. A transcritical CO_2 cycle operates in significantly higher pressures than HFC systems. The circuitry configuration is, in agreement with Evans [38], similar to that of the DX refrigeration technology, except for the heat rejection at a gas cooler at ambient temperatures above 23°C, since the carbon dioxide leaving the compressor is supercritical, a state where vapor cannot condense and one

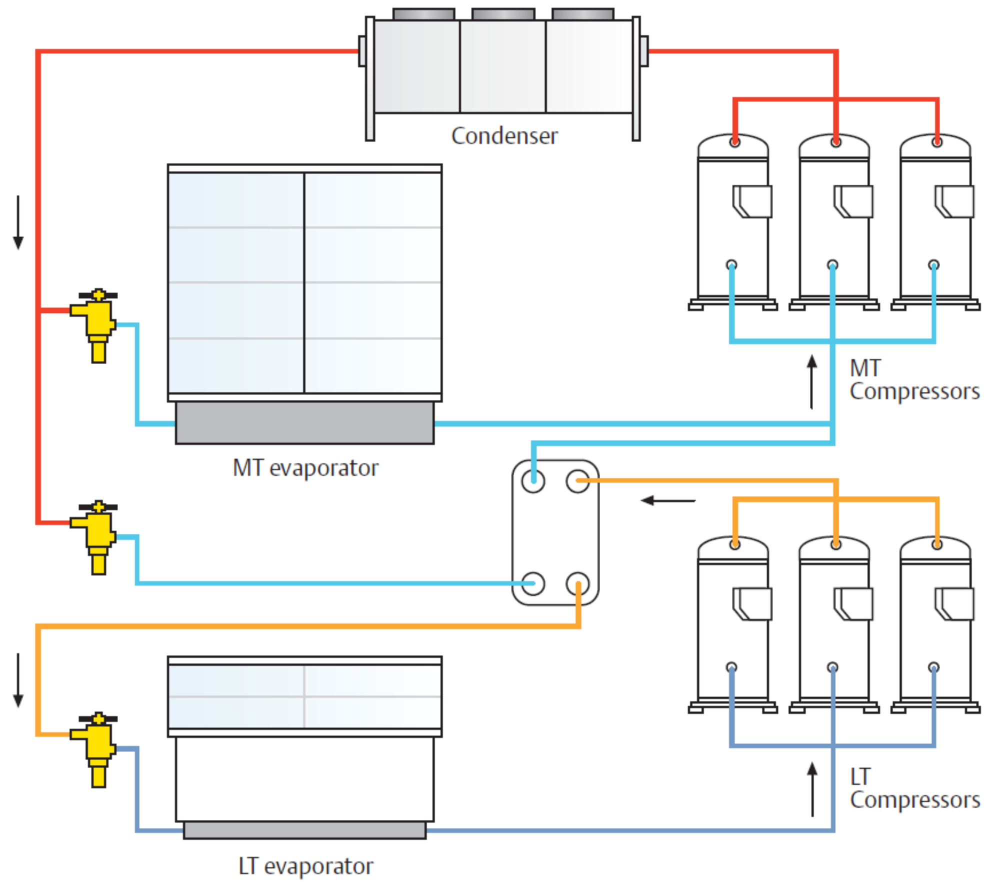


Figure 1.14: Schematic design of a cascade refrigeration system, from Emerson [32].

cannot distinguish liquid from vapor [38].

Sawalha [37] reported that control of the pressure in the gas cooler (or condenser when in subcritical conditions) is essential for the system to achieve its best performance, with an optimum value for the COP of the technology existing for each ambient temperature. The author also observed that this technology is better suitable for cold climates, where the operation of the system will mostly be in the subcritical region [37], improving the efficiency of the cycle.

A clear disadvantage of the transcritical CO_2 system is, as addressed by Evans [38], the low performance when operating in warmer ambients. In such case, the heat rejection pressure of the high-stage cycle is considerably higher than what it would be if any other common refrigerant was utilized, enforcing, thus, transcritical operation [37].

CO_2 booster refrigeration system. Booster systems use CO_2 in both the transcritical/subcritical cycles, depending on the ambient temperature, and supply both the low and medium temperature loads. A direct connection between the LT and MT circuits exists in the form of a flash tank or separator, with the low temperature cases regulated by a booster compressor [1]. At ambient temperatures above 23°C, roughly, the compressors discharge the gas above the CO_2 critical pressure, and heat rejection takes place at a gas cooler, reducing the temperature of the discharge gas without condensing it into liquid [32]. When passing through a pressure reduction valve, a portion of the cooled fluid condenses, with liquid and gas being separated in the flash vessel mantained at intermediate pressure. The liquid is then distributed to the display cases by the lines, whilst the gas is delivered at the suction stage of the medium temperature compressors via an expansion device [32]. Figure 1.15 schematically represents the CO_2 booster refrigeration system.

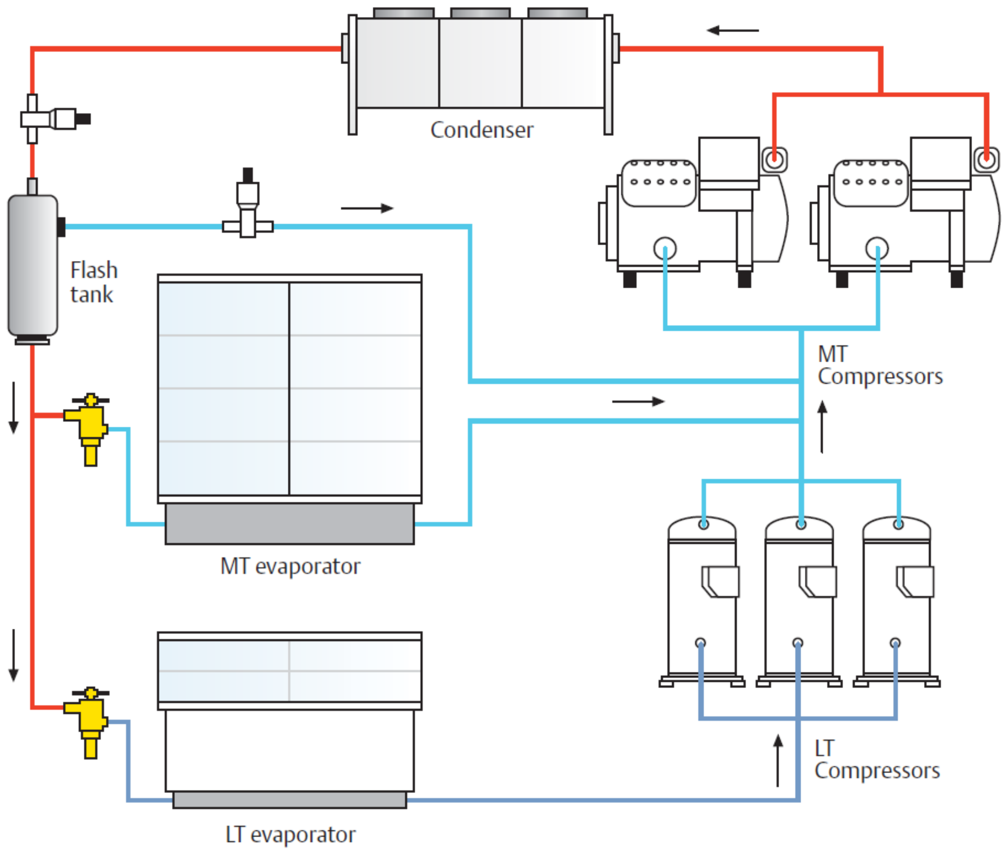


Figure 1.15: Schematic design of a CO_2 booster refrigeration system, from Emerson [32].

According to Bansal [1], one of the main advantages of the booster technology is that, given the flash gas concept, high pressure is required only between the compressor and the separator, being limited, thus, to the machine room. Although significantly high costs and oil managenet issues have been reported [1] for this system, it has been embraced by the food and refrigeration industry due to benefits like smaller pipes, low pumping power, only one liquid line and high thermal efficiency of CO_2.

In addition, these systems tend to operate subcritically most of the year, in moderate climates, with reduced capacity of the condenser and increased refrigeration effect [1]. However, in agreement with Chiarello et al. [39], for high ambient conditions, such as in Brazil, performance may be greatly harmed given the expanded transcritical operation, demanding, in such conditions, extra adjustments.

1.2 Objective and methodology

The main objective of the present work was to compare the performance of commercial refrigeration systems operating with a variety of refrigerants, including low-GWP fluids. The technologies were compared by means of the coefficient of performance, annual energy consumption and life cycle climate performance, for different climate conditions. In order to do so, two thermodynamic models (Part I) and one lumped parameter model (Part II) were developed.

Thermodynamic models for the secondary coolant system operating with carbon dioxide as a secondary refrigerant, namely, pumped CO_2 system, and for the CO_2 booster refrigeration system, were built. Both simulation tools are capable of predicting system performance with pure compounds, azeotropic or non-azeotropic mixtures as working fluids. Subsequently, COP and annual energy consumption analysis were performed, in order to compare these technologies with the conventional supermarket direct expansion system. To extend the scope of comparisons, the simulations included R404 and some of its low-GWP potential replacements as primary-cycle refrigerants.

A lumped parameter model for the steady-state operation of a multi-compressor multi-evaporator DX system was, then, developed. The simulation tool, equipped with component-based solution, employs air-source heat exchangers with tube-and-fin configuration, as well as a compressor model characterized by the use of either map curves or performance data. Similarly to the thermodynamic models, this simulation tool was adapted to run with pure fluids, zeotropic or azeotropic compounds as refrigerants. The lumped parameter model developed was used to perform an LCCP analysis, so as to compare the environmental impact of a typical R404A DX refrigeration system with that of potential low-GWP retrofit solutions.

1.3 Literature review

1.3.1 Pumped CO_2 refrigeration system

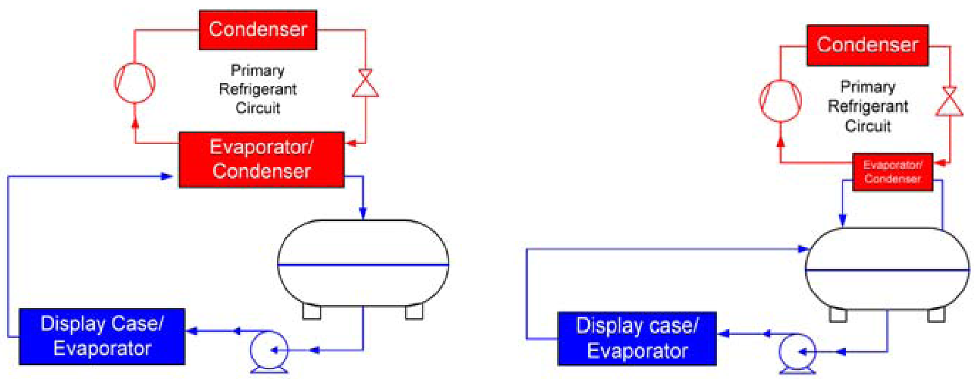
Wang et al. [36] presented a comprehensive review of secondary loop refrigeration systems within commercial refrigeration, including a discussion of the benefits and disadvantages of employing carbon dioxide as a two-phase secondary fluid. Inlow and Groll [40] evaluated the performance of secondary loop systems operating with different secondary fluids, comparing it to that of a R22 conventional system. They concluded that a secondary colant system with CO_2 as a two-phase secondary refrigerant and ammonia as the primary refrigerant provides a COP equivalent to that of existing DX systems using R22. Further, according to the authors [40], indirect systems using single-phase refrigerants do not perform as well due to higher pressure losses and lower heat transfer performance. The operation of the pumped CO_2 refrigeration system in low-temperature applications was investigated by Hinde et al. [41] with field installations. Main features included small pipe size, excellent heat transfer properties, good material compatibility and small refrigerant charge. Drawbacks included, according to the authors, high operating pressures, low availability of components, and added pumping power [41].

A few other analyses of succesfull field installations of indirect systems, in low temperature level, utilizing CO_2 as a two-phase secondary refrigerant, have been performed [18, 41–49]. Pearson [43] considered two-phase CO_2 as secondary fluid in supermarket systems for the Swedish market. Pachai [44] addressed a pumped CO_2 secondary system installed in more than fifty shops in Sweden. The primary refrigerant was a R290/R170 mixture, and the secondary refrigerants, for low and intermediate temperature levels, were CO_2 and propylene glycol, respectively. Hinde et al. [41] reported that at least nine low-temperature CO_2 secondary systems were operational in the U. S. and Canada by early 2008, with R404A or R507 as primary refrigerants, and CO_2 and propylene glycol as the low and medium temperature secondary coolants, respectively. The stores ranged in size from small neighborhood markets to large supercenters and warehouse-style stores, where cooling capacities ranged from 22 to 160 kW.

Celik [50] estimated the effect of individual cycle components on system capacity and performance of Pumped CO_2 refrigeration systems. Cycle optimization was conducted as well, by using mass flow rate ratio, intermediate pressure coefficient and power ratio. Kaga et al. [51] developed a compact variable capacity cooling system with an inverter compressor, using R600a as the primary refrigerant and carbon dioxide as the secondary fluid, which was circulated by "thermosiphon" effect. This enhanced system presented COP values ranging from 2.1 to 2.6, with refrigerating capacity going from 170 to 230W. Further, the energy consumption was 5% lower than that of a R134a baseline DX system [51].

A number of studies from the literature compared indirect CO_2 system solutions to conventional DX installations [18, 31, 37, 40, 41, 46–49, 52–56]. Results were, overall, conflicting, though the majority reported that the secondary coolant technology had a slightly higher energy consumption than the traditional system.

Sawalha [37] theoretically analyzed different aspects of the application of carbon dioxide in supermarket refrigeration, including its usage as secondary fluid. A pumped CO_2 system is one of the technologies modeled and simulated by the author, with further performance evaluation and system optimization. Sawalha provided basic schematic diagrams for two possible arrangements for pumped CO_2 secondary circuits, as shown in Figure 1.16. Nevertheless, the author developed computer models, thermodynamic analysis and optimization for the arrangement in Figure 1.16(a) only. Results indicated that the power consumption of the indirect technology investigated was 13% higher than that of the DX system with the same refrigerant in the primary loop.



1.16(a): Secondary circuit crossing the receiver once

1.16(b): Secondary circuit crossing the receiver twice

Figure 1.16: Basic schematics of two arrangements for CO_2 secondary circuits (from Sawalha [37]).

A thermodynamic model was developed for the evaluation of energy consumption, total equivalent warming impact (TEWI) [21] and LCCP of CO_2 secondary fluid systems by Portilla [52]. The author considers the same configuration for the secondary loop as Sawalha [37], Figure 1.16(a), and compares the results with those of the direct expansion technology. Portilla's results [52] indicated that the secondary coolant system with R407F as primary refrigerant and CO_2 as secondary refrigerant provides reduction of almost 22% in environmental impact, if compared to the R407F DX system. In that case, annual power consumption of the Pumped CO_2 configuration investigated was only 1% higher than that of the Direct Expansion system.

Hinde et al. investigated [41] energy consumption and TEWI of a low temperature CO_2 secondary coolant system. Compared to a typical supermarket HFC DX baseline system, the pumped CO_2 system saved 3 to 12% energy, depending on climate and level of subcooling. The impact on the total TEWI was primarily a result of the reduction in refrigerant charge, providing emissions up to 70% lower [41]. On the other hand, results obtained by Kruse [53] showed higher energy consumption for the secondary coolant technology. The author followed a strategy similar to that of Hinde et al. [41] in comparing the energy consumption of a DX system and an indirect refrigeration system with a CO_2 secondary loop.

Simulations of direct and indirect refrigeration systems in a supermarket stores were performed by Arias [18], using a computer program with the ability to simulate building heating and cooling loads. Results indicated that, in the city of Sala (Sweden), a carbon dioxide indirect system with heat recovery and float condensing temperature was the most energy efficient, had the lowest impact on the environment and the lowest cost, during a study period of 15 years [18].

The secondary loop refrigeration system installed in an American superstore of 11,000 square meters floor area, operating with propylene glycol and CO_2 as secondary fluids for medium and low temperature applications, respectively, was also analyzed [47]. The indirect technology reduced the charge amount of R507 by 90%, with an estimated 7% reduction in the carbon footprint of this store also reported. Christensen [46] investigated a secondary refrigeration system using carbon dioxide as primary and secondary refrigerant in supermarket applications. Tests and field measurements were carried out and compared with the original cabinet, with CO_2 proving to be a good alternative to the HFCs, according to the author [46]. Also in a field study, Heinbokel [49] reported that the CO_2 indirect system had a 6% higher energy consumption if compared to a conventional system, defined as a DX R404A single stage, while a brine system had a 20% higher consumption.

1.3.2 CO_2 Booster refrigeration system

The CO_2 booster technology is a recent development, so reports on the subject are scarce, especially those providing experimental results or field measurements.

Voorhees [57] was the first to introduce a multistage CO_2 transcritical cycle, showing that, by using two-stage compression, the COP could be improved, and the necessary high pressure, reduced. A few other authors reported on the benefits of considering the CO_2 Booster refrigeration system, instead of a simple transcritical CO_2 cycle with one compression only [1,58–60]. Yang et al. [58] observed that multistage CO_2 cycles could yeld performances 9% higher than those of single-compression systems. Agrawal et al. [59] showed that staging not only enhanced the performance, but also improved the design.

Bansal [1] presented a review of the application of carbon dioxide refrigerant in low temperature refrigeration systems, along with some discussion on the new system designs being used in the food industry, of which the CO_2 transcritical booster was included. The author suggested that this technology, although costlier than the single stage system, due to additional elements and control, is gaining interest of the food retail industry, for the commercial availability of its components and other benefits, like smaller pipes and corresponding insulation, low pumping power and high thermal efficiency of CO_2, one liquid line and less occupation of machine room space. In his review of cycle modifications for CO_2 refrigeration and heat pump systems [60], Sarkar also highlighted the benefits of considering a multistage CO_2 transcritical system.

Field measurements of CO_2 booster systems are scarce, given the novelty of the technology [61, 62]. Danfoss [61] covered the technical design of a 10kW (LT) and 24kW (MT) transcritical CO_2 booster refrigeration system with gas bypass for a small discount supermarket. The system was, later, designed and built, presenting an energy consumption 4% inferior to that of a conventional R404 DX system, and has been in operation since 2007 without major problems. Madsen [62] described the design and construction of a booster system installed in a small supermarket. The electricity energy consumption of the system was monitored and compared with that of stores using R404A systems, with results indicating that the CO_2 booster was more efficient than the conventional technology by 4%, as well.

Optimization of parameters associated with the CO_2 booster system has been performed in the literature as well [63–70]. Baek et al. [64] optimized the pressure ratio of the transcritical carbon dioxide booster cycle with intercooling, whilst the optimum discharge pressure for flash gas bypass system was given by Agrawal et al. [59]. Cavallini et al. [66] experimentally studied a CO_2 booster system with intercooling, observing the existance of an optimum upper cycle pressure for transcritical heat rejection.

The energy efficiency of the CO_2 Booster refrigeration system was investigated by Ge and Tassou [69], who identified and examined, through thermodynamic analysis, the effects of parameters affecting system efficiency in the transcritical cycle, at higher ambient air temperature. The effects of varying high side refrigerant pressure, ambient air temperature, refrigerant intermediate pressure, medium and low evaporating temperatures, superheating degree, effectiveness of suction line heat exchanger, and compressor efficiency were all evaluated. In the end, the optimal high side pressure in the transcritical cycle was also established and derived as a function of ambient air temperature, effectiveness of the suction line heat exchanger and compressor efficiency [69]. Danfoss [70] provided an application guide listing a number of control measures to optimize the CO_2 booster system, including controls for gas cooler, receiver, injection, compressor capacity and high pressure.

Thermoeconomic theory was used by Ommen and Elmegaard [71] to establish the cost of cooling in MT and LT refrigerated display counters, considering both subcritical and transcritical operating criteria. The authors verified that the cost of reaching temperatures of frozen goods, in the circumstances considered, was more than twice the cost rate for equivalent chilled temperatures. Another study, by Agrawal and Bhattacharyya [72], showed that flash gas inter-cooling is not economical for CO_2 Booster systems.

Enhanced multistage CO_2 systems are also subject of study in the literature [69, 73–76]. Ge and Tassou [69] reported that lower ambient temperatures and sizeable heat recovery benefit the CO_2 Booster system performance, with the technology presenting recovery potential even during operation at subcritical conditions, due to the higher cycle pressures and temperatures. Colombo et al. [73] also investigated heat recovery for retail applications, using an enhanced booster CO_2 transcritical system with high and medium temperature. Results indicated that this system can provide a large potential reduction in CO_2 equivalent emissions compared with a conventional HFC-based system, when sustainable heating and cooling solutions are present.

Another study, by Cecchinato et al. [74], showed that more elaborate cycles (double-throttling, open flash tank and split cycle) presented great improvement, especially for the heaviest operating conditions (the lowest evaporating temperature and the highest external temperature). Further, through theoretical thermodynamic analysis, Cho et al. [76] concluded that the two-stage CO_2 cycle with vapor injection yielded an 18.3% improvement in performance.

A few authors compared the CO_2 booster technology with typical direct expansion systems efficiency-wise [32, 55, 56, 77–80]. Ge and Tassou [77] used the supermarket simulation software "SuperSim" to predict and compare the annual performance of a supermarket CO_2 booster refrigeration system with that of a conventional R404A multiplex system. By optimizing the control of both systems to yield maximum seasonal efficiencies, the authors [77] verified that, for weather conditions in the North of England, the two refrigeration systems lead to similar energy consumption. Mikhailov [78] compared COP and TEWI of different refrigeration system designs, with results also indicating that a CO_2 Booster system provides best efficiency in moderate and cold climates.

Emerson [32] analysed different refrigerant/technology combinations under three main angles: energy consumption, environmental impact and investment cost. Fourteen combinations of system configurations and refrigerants were investigated, including a CO_2 booster transcritical system. Main conclusions of the study indicated that, in transcritical mode, coefficients of performance (COP) of the booster are lower than conventional vapour compression systems, with significant penalties in warmer climates like Southern Europe. In contrast, the carbon dioxide booster was verified to be an excellent configuration to lower the TEWI in Northern Europe, though moving to this new technology would require a full store architecture change [32].

An LCCP analysis was performed by Abdelaziz et al. [79] to investigate the R744 transcritical booster system as a replacement for a HFC multiplex DX system. The CO_2 booster system showed a 77% emissions reduction compared to R404A, though such result came at the cost of higher energy consumption. Fricke et al. [80] also compared the traditional multiplex DX system and the transcritical CO_2 booster system in terms of life cycle climate performance, as well as energy consumption. The authors concluded that a typicall CO_2 booster refrigeration system, coupled with high-efficiency display cases and walk-ins, could achieve energy consumption and carbon dioxide emissions reductions of 39% and 76%, respectively, when compared to the traditional multiplex DX system.

1.3.3
Modeling of steady-state operation of vapor compression system

Steady-state simulation, according to Winkler [81], describes the steady-state behavior of a system at a set of specific design points, as well as predicting its performance at off-design conditions. Since vapor compression cycles are evaluated and designed by means of steady-state performance, it becomes clear why most of the studies on the system make use of simulation in steady-state operation [82, 83].

Numerous simulation models of steady-state operation of vapor compression simulation models can be found in the literature [59, 84–107]. A broad review was provided by Winkler [81]. The vast majority of these simulators were first developed for academic purposes, thus their applications were restricted to the principal investigators. However, several of them progressed to now commercially available software packages [81].

The first simulation tool developed to support the design of vapor compression cycles was the Oak Ridge National Laboratory (ORNL) Heat Pump Design Model [89]. The simulation model was based on previous studies from ORNL [86] and the Massachusetts Institute of Technology (MIT) [85], and was limited to steady-state air-to-air heat pump applications. Nevertheless, the tool has remained relevant by means of various improvements proposed in the subsequent decades. In fact, the present work used the tube-and-fin heat exchangers of referenced model [89], coupled with many additional and updated routines modifications, to simulate a multi-compressor multi-evaporator Direct Expansion refrigeration system [81].

CYCLE_D, a program developed at the National Institute of Standards and Technology (NIST) [88], aimed to provide theoretical analysis of vapor compression systems. The simulation tool was a result of a series of revisions of previous models (CYCLE-11 and CYCLE-7 [90]), with considerable focus in the performance evaluation of pure and mixed refrigerants [81]. The thermodynamic model provided by the Research Center for Refrigeration Technology and Heat Pumps (FKW), Cycle Calculation Program KMKreis, calculated design points for nine distinct cycle configurations. Analogously to CYCLE_D, it was a simple simulation tool, not designed to provide very detailed analysis of vapor compression system configuration, though widely used since it was first available [81].

Some of these steady-state vapor compression simulation models were component-based, that is, additional components, often developed by the users, could be inserted with ease into the original configuration. In this approach, the component models are treated as "black-box" elements, with the system solver only requiring knowledge of how they are connected , as reported by Winkler [81]. A component-based simulation scheme presents increased flexibility, since it can be employed to analyze arbitrary system configurations, though penalties in robustness are observed, given that the solver cannot assume a functional form of the mathematical equations being solved by the component models [81].

The first component-based solution scheme was developed to handle the simulation of a variety of absorption system configurations [108]. Stoecker [109, 110] suggested benefits and drawbacks of considering a component-based simulation, presenting, later on, concepts for component-based model for thermal system analysis [81]. Parise [95] employed and solved, sequentially, simple component models to simulate a vapor compression system, with Herbas et al. [98] improving the tool by implementing more detailed component models [81].

A more detailed simulation model, ACMODEL, was provided by Herrick Laboratories at Purdue University [100], as well. Although the program was built in a modular and object-oriented format, the component library was limited to a single finite difference-based heat exchanger model and a single ARI map-based compressor model [111]. The Advanced Refrigeration Technologies (ART), a program developed by the Thermal Systems Research and Modeling (IMST) of the Universidad Politécnica de Valencia [103], offered the option of adding a number of accessories to the simulation model, from piping to liquid-line suction-line heat exchanger. The tool, similarly to ACMODEL [100], was organized in a modular structure, allowing for different component models to be selected from a library and to be run by the program. Originally introduced to simulate water-to-air heat pumps, the program was later enhanced [104] to include air-to-air heat pumps [81].

NIST implemented ACSIM [112] as a vapor compression system simulation tool to allow for the design of evaporator and condenser circuitries. The program required the user to input an ARI compressor map and a constant system superheat [81]. The component-based model VapCyc, introduced [105] and, later, improved [106] by Richardson et al., allowed for the simulation of steady-state vapor compression cycles with additional components.

CoolPack, a collection of simulation models for refrigeration systems, was developed by the Department of Mechanical Engineering (MEK) at the Technical University of Denmark (DTU). The software consists of three sets of programs: Refrigeration Utilities, EESCoolTools, and a transient element called Dynamic. For the EESCoolTools, Engineering Equation Solver (EES), a software developed by Klein and Alvarado, was employed for the development of the simulation models. The dynamic elements are modeled and solved using a DAE-type simulation program called DALI, developed by the Department of Energy Engineering at the Technical University of Denmark [113].

1.3.4 Life cycle climate performance analysis of supermarket refrigeration systems

The Technology and Economic Assessment Panel (TEAP) of the United Nations Environment Program (UNEP) was the first to propose [21] the life cycle climate performance (LCCP) as a tool to assess the direct and indirect environmental impacts of greenhouse gas emissions [3]. Hwang [114] presented a detailed review on concepts, calculation methods and available tools for the LCCP Analysis. The author described the indicator as a rigorous approach to identifying and quantifying direct and indirect environmental impact. Wang et al. [115] analyzed and investigated differente indexes for estimating the global warming impact, including GWP, TEWI and LCCP, considering the concept, contents, difference and correlation among them. Results indicated LCCP to be the most accurate and reliable method.

Many other authors reported the LCCP analysis to be a good indicator of energy efficiency and environmental impact for commercial refrigeration [116–122]. Neksa [118] suggested that the best way to address indirect emissions and, thereby, energy efficiency, was through life cycle performance evaluations. Among the indexes that have been proposed to measure the global warming impact, Riva et al. [119] classified the LCCP philosophy as the most "convincing". Pham and Rajendran [120] adopted the life cycle climate performance as the methodology to review the status worldwide on technical and policy search for next-generation refrigerants. Jayakumar [121] considered LCCP as a more holistic approach to refrigerant selection, stating that any choice made to move away from today's refrigerants should be LCCP neutral at minimum.

The number of publications that perform LCCP analyzes based on theoretical or experimental data of commercial refrigeration systems is limited, since it hasn't been long since the concept was first introduced [21]. However, it is growing rapidly [3,31,52,55,79,80,123–127]. The Intergovernmental Panel on Climate Change (IPCC) provided an overall review [116] on publications that employ the life cycle cimate performance as an environmental indicator to compare supermarket refrigeration systems and alternative refrigerants.

ADL [3] used the LCCP methodology to investigate environmental performance of specific HFCs and non-HFCs technologies in applications for commercial refrigeration. The report [3] presented a number of results regarding the selection of alternative refrigerants, including the suggestion of employing ammonia as a primary refrigerant in secondary loop configurations, as long as proper design of the mechanical machine room were to be developed.

Further, conclusions of the study [3] included the observation that refrigerant emissions contribute much more to the LCCP in typical supermarkets DX cycles than in smaller, factory assembled self-contained equipments.

Harnisch et al. [124] calculated the LCCP for different types of supermarket refrigeration systems in Germany using a straightforward model, which accounted for production, emissions and energy consumption. Portilla [52] also employed the LCCP method to compare the environmental impact of alternative supermarket refrigeration technologies, based on the simulation of thermodynamic models operating with non-azeotropic refrigerants. A more elaborated LCCP study of three R-22 alternatives in commercial heat pump systems was performed by Spatz [125]. The analysis employed detailed system modeling for energy use, by means of compressor maps, tube-to-tube modeling for heat exchangers, and analytical models or correlations for expansion devices. Results indicated the indirect effect as the dominant factor in the LCCP of heat pumps [125].

In order to provide a clear understanding of the performance of HFCs in commercial walk-in refrigeration systems, Hwang et al. [126], based on measured data, compared their environmental impacts, from estimates for power consumption and LCCP.

Motta and Spatz [31] performed LCCP analysis of technologies used in supermarket refrigeration with newly developed refrigerants in operation. The LCCP evaluation indicated that superior energy-efficiency and lower GWP refrigerants had strong potential to reduce the carbon footprint of current and future systems. Fricke et al. [80] presented energy and LCCP analyses of a variety of supermarket refrigeration systems, with different refrigerants, in a number of climate zones across the United States. Results revealed that high-efficiency supermarket refrigeration systems could reduce carbon dioxide emissions by as much as 78%, compared to the baseline multiplex DX system.

The ORNL [127] developed a vapor compression system design and evaluation tool for supermarket refrigeration, based on the LCCP concept, in an Open Source environment. The ORNL LCCP model is able to assess a variety of vapor compression system applications, including supermarket refrigeration cycles. Abdelaziz et al. [79] utilized this LCCP design tool to evaluate the performance of a typical multiplex DX commercial refrigeration system with alternative refrigerants. Minor system modifications in the circuitry were also considered, with ambient temperature and system load determined from a widely used building energy modeling tool (EnergyPlus [128]).

Possible improvements on the LCCP analysis have also been reported by a few authors. Riva et al. [119] highlighted the lack of standardisation that often makes LCCP values not clearly comparable. Further, according to the authors [119], a conclusion about the environmental impact cannot be obtained unless the economic aspect of limited financial resources is taken into consideration. Using a supermarket refrigeration system as an example, they [119] showed how the concept of eco-efficiency could complements the LCCP analysis. Following the same strategy, Voigt [56] studied the global warming impact of various types of supermarket refrigeration systems based on eco-efficiency methodology.

According to Hwang [114], in order to establish a harmonized LCCP evaluation methodology, applicable for refrigeration and air conditioning systems, the International Institute of Refrigeration (IIR) formed a working party to focus on increased research and development on the area. The IPCC [116] suggested that, although a powerful tool, results for LCCP were too sensitive to input assumptions, requiring careful validation. ADL [3] considered inappropriate to use, when evaluating LCCP values, a 100 year integration time horizon in conjunction with certain compounds. According to the study [3], CO_2 has a lifetime over 100 years, though many of the HFCs do not, which could compromise potential comparisons.

1.3.5
Main contributions of the present work

In Part I, regarding the CO_2 Booster system modeling, a fundamental attribute of the tool developed is the ability to address operation in both subcritical and transcritical conditions, whilst most of the studies are limited to transcritical operation. When discussing Secondary Coolant refrigeration systems, not many authors accounted for the carbon dioxide as a two-phase secondary refrigerant and, of those who did so, even less adopted the circuit configuration presented in Figure 1.16(b), as in the present work.

All the thermodynamic models that have been developed in this study are similar in the capability to simulate the operation of non-azeotropic refrigerant mixtures with any degree of temperature glide. The inclusion of pressure drop and heat transfer effects along the liquid and vapor lines is also uncommon in the literature revised. Another particularity is the large number of degrees of freedom that potential users of the present model will find when entering input information for the models: a number of different parameter options are considered for each device.

Further, one can observe that most of the authors who compared novel supermarket technologies to conventional installations have done so considering the DX R404A as a reference. One of the contributions of this study was to consider commercial Direct Expansion systems operating with new low-GWP refrigerants, in order to extend the scope of the comparisons. In that sense, part of the uniqueness of the present work is associated with the inclusion of some of the most recent low-GWP fluids developed, notably the blends HDR21 and HDR81, for which open source data is not available yet. Additionally, annual energy consumption calculations were performed in six different locations, including cities in the U. S., Europe and Brazil, so as to account for a wide variety of climate conditions.

In Part II, when looking at supermarket refrigeration systems design, one of the main singularities of the lumped parameter DX model developed was the implementation of multiple compressor and evaporator packs. The great majority of the studies have not addressed this particularity, as it requires the development of a more complex component-based solution. In addition, regarding the models for air-source fin-and-tube heat exchangers, the updated correlations considered for heat transfer coefficient and pressure drop were key factors, as well as the inclusion of several options of tube internal surface and fin pattern.

Not many papers have been found in the literature using the LCCP as an indicator for evaluating environmental impact. Also, the development of such a complex component-based simulation model for the environmental assessment is infrequent, with most of the studies employing existing tools, or utilizing simple models. In fact, following the suggestion of the IPCC [117], who ephasized the sensitiveness of the LCCP methodology to the input data, all parameters involved in the present LCCP analysis were either calculated by the simulation model (refrigerant charge and energy consumption), or obtained through exhaustive literature review (leakage rate, indirect emission factor, etc.).

To the author's knowledge, no previous studies involving all of these aspects in the comparison of commercial refrigerion technologies operating with low-GWP fluids could be found in the literature reviewed.

1.4 Structure and organization

This study was divided in two parts. Part I is associated with the development, verification and comparison of the thermodynamic models.

- Chapter 2 presents an overview, the mathematical model, the numerical solution and a model goodness verification for the simulation tool developed for the pumped CO_2 system;

- Chapter 3 follows the same pattern of the previous chapter, now for the simulation model developed for the CO_2 booster system;
- Chapter 4 presents the performance comparison between the thermodynamic models developed and a supermarket DX system, based on COP and annual energy consumption.

Part II encompasses the development, validation and analysis of the lumped parameter model.

- Chapter 5 presents an overview, the mathematical model, solution scheme and validation for the component-based simulation tool developed for the multi-compressor multi-evaporator Direct Expansion system;
- Chapter 6 describes the LCCP analysis of the lumped parameter model when operating with different refrigerants, for two distinct case studies.

Finally, the present work ends with concluding remarks and suggestions for future studies, in Chapter 7.