CHAPTER 2

Literature Review

This chapter attempts a comprehensive survey of reported studies related to CO_2 refrigeration systems and its development. Focus is on introduction of major cycle components and reported modifications in cycle configurations seeking performance improvement including, internal heat exchanger, ejector, work recovery expander, dedicated sub-cooler, flooded evaporator and twin-staging, cascading. The state of art of CO_2 refrigeration in supermarket application is discussed and the research gap areas are identified.

The literature review covers more than hundred research papers published between 1993 and 2018 in various reputed journals such as International Journals of Refrigeration, Applied Thermal Engineering, Energy Conversion and Management, Applied Energy, Energy, International Journal of Heat and Mass Transfer, ASHRAE transactions, HVAC&R Research, and conference proceedings namely IIR, ASHRAE, Atmosphere and IRAC for last 20 years.

2.1 CO_2 trans-critical refrigeration system background

CO_2 was introduced as refrigerant in 1870s, but due to lack of adequate technology, the coefficient of performance (COP) was low. Subsequent development of synthetic refrigerants with high COP like CFCs in 1930 and HCFCs in 1940s led to almost disappearance of CO_2 based systems. In 1992, Lorentzen revived CO_2 as a refrigerant through a trans-critical automotive air conditioning (Lorentzen 1994). The work was highly accomplished by the refrigeration research community as well as industry and thereafter, led to many developments in wide range of applications. Researchers are trying to design new and more efficient CO_2 systems by blending old concepts with state of the art technology.

A typical vapor compression trans-critical CO_2 cycle is shown in Fig. 2.1. Evaporation (4-1) takes place at sub critical pressure like other conventional vapor compression systems and heat rejection may take place at supercritical pressure (2-3) when the ambient is high, owing to its low critical temperature of 31.3°C. Thus, in high ambient operation, the cycle operates partly subcritical (low pressure side) and partly supercritical (high pressure side).

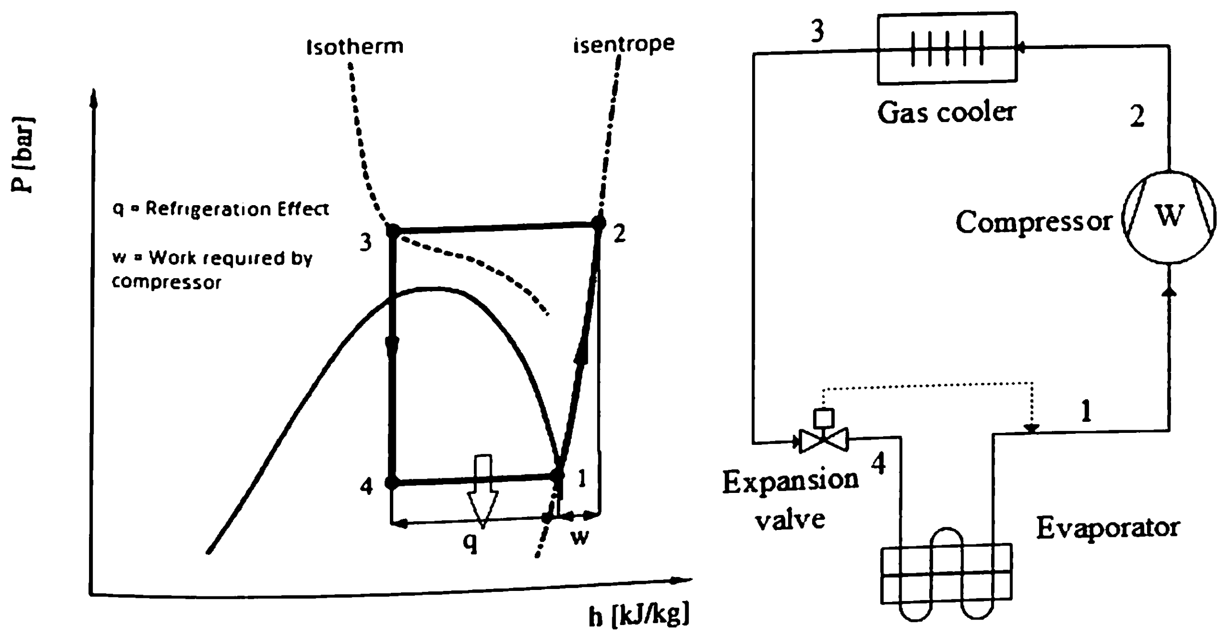


Fig. 2.1 Typical Ideal Trans-critical CO_2 Refrigeration Cycle

In supercritical heat rejection, no saturation point exists, consequently, the gas cooler pressure is independent of the refrigerant temperature at the gas cooler exist (state 3). Pettersen (1994) reported that the gas cooler pressure is an influencing parameter due to the s-shape of the isotherm in supercritical region. Since the throttling valve inlet condition determines the specific refrigeration effect, it is necessary to control the high side pressure for optimal operations. Further there is additional challenge for tropical regions to operate trans-critical CO_2 based systems under tight control of pressure to obtain effective heat rejection in the gas coolers. Several authors like Kauf (1999), Sarkar et al. (2004), Cabello et al., (2008), Yang et al., (2015), Sarkar and Joshi (2016), Gullo et al., (2016b), Hazarika et al., (2018) and Shao et al., (2018) had reported importance and methodology of achieving optimization of cycle performance with respect to gas cooler pressure.

2.2 Major components of CO_2 refrigeration cycle

The major components of the system are compressor, gas cooler, expansion device and evaporator. Due to trans-critical operation as well as unique properties of CO_2, design of system and the components are different compared to conventional vapor systems. The components are also subjected to high pressure, which necessitates a special attention for material selection and design. The compatibility of components to each other as well as maintenance and safety assurance are other critical issues to note in the CO_2 system.

2.2.1 Compressor

Försterling et al. (2002) reported experimental investigation of trans-critical CO_2 compressor for mobile A/C systems. The compressor as well as the whole refrigeration system for mobile air conditioning demands for compact design. Thus, two different compressor designs, pressure controlled reciprocating type and pressure & path-controlled swash plate type were tested, and it was concluded that the latter is more efficient. Kim et al. (2004) conducted a comparative study for several compressor designs including scroll, two stage twin rotary and two-cylinder reciprocating compressor for CO_2 compression and reported scroll compressor to be better. They also reported that due to high density of CO_2, the compressors design is compact compared to other refrigerants. Minetto et al. (2005) reported that the ratio of compressor's displacement volume is one of the most prominent factors affecting the COP and recommended the use of varying swept volume compressor for optimization of the cycle performance, when the ambient temperature is varying. Lambers and Kohler (2006) investigated Voorhees modified CO_2 compressors using indicator diagram. Voorhees compressors are based on the "multi-effect" principle or economizer principle. It was reported that by mapping with indicator diagram, the required compressor capacity may be estimated for a desired application. Rigola et al. (2006) compared performance of a semi-hermetic and a hermetic CO_2 compressor against conventional R134a compressor. In all investigated cases, performance of CO_2 semi hermetic compressor was found better. Jover et al. (2007) investigated CO_2 compressors for light commercial applications. They introduced a new design for reciprocating compressor with reduced compressor volume. Neto and Barbosa (2009) investigated phase change and volumetric flow behavior and mixing of CO_2 and synthetic lubricant oil in compressor. The motivation of the study is from observed high mutual solubility of CO_2 and lubricating oil used in compressor and possible property change of oil as well as refrigerant due to the same. They utilized two commonly used synthetic lubricants, namely, polyolester ISO 68 and an alkylbenzene ISO 32 and these were tested at temperatures 285, 298, 308, 328 and 348 K. The experimental results obtained were used to predict refrigerant solubility in oil, its liquid density and barotropic behavior with reasonable accuracy. Lee and Lam (2013) reported a comparative study of various generalized modelling approaches for scroll compressors. The constant polytropic efficiency approach (CPE) was found to be better for approximating the performance of the scroll CO_2 compressors. Kurtulus et al. (2014) proposed and tested a novel positive-displacement oil-free CO_2 compressor. The oil-free compressor uses the Sanderson rocker arm motion mechanism (S-RAM), which converts reciprocating to rotary motion. They reported overall isentropic efficiency of maximum 75.66% and minimum of 54.55% for the tested oil free compressor. Zhang et al. (2016) reported investigation on a single stage and a two stage CO_2 compressor, developed with same displacement. They reported that single stage CO_2 compressor depicted better performance in terms of isentropic and volumetric efficiency at lower pressure ratios while at higher pressure ratios the performance of both single stage and double stage compressors were similar. Further, the superheat was found to have no significant effect on the performance of either type of compressors. Wang et al. (2018) investigated stress deformations of a CO_2 scroll compressor using commercial software. They reported that under the combined load of pressure as well as temperature, the maximum stress and deformation was observed in the center of the gear head.

2.2.2 Gas Cooler

Pettersen et al. (1998) investigated compact heat exchangers for CO_2 A/C systems. They reported higher overall efficiency for micro-channel-based heat exchangers. Problems like refrigerant distribution in evaporator manifolds and heat transfer tubes were found to be less significant. Park and Hrnjak (2004) studied the effect of gas cooler size on system performance. They reported that about 80% of the total gas cooler capacity was covered in the first pass or the inlet part of the gas cooler. As a result, while the first tube is very hot and participates in convective heat loss to ambient air, the rest have much lower contribution, reducing effective convection. Aimed at establishing a database for designers, Hwang et al. (2005) investigated fin and tube type gas cooler for two different ambient conditions (29.4°C & 35°C). They also reported that gas cooler capacity was higher at ambient temperature of 29.4°C than at 35°C. Park and Hrnjak (2007) investigated the adverse effect of possible heat conduction between the tubes in a gas cooler and concluded that it could be reduced by cutting some fins, when the conduction is significantly high. They also demonstrated up to 3.9% improvement in gas cooler capacity by cutting some of the fins strategically. Ge and Cropper (2009) investigated air cooled finned tube CO_2 gas coolers. In their work three different circuit arrangements were tested and they reported that the performance can be improved by increasing the number of circuit. But in their work effect of fan power and system performance were not considered. Sarkar et al. (2010b) reported experimental investigations on water cooled gas cooler for simultaneous heating and cooling application in Indian climatic conditions. They reported that the system COP increases with increase in mass flow rate of water in coaxial gas coolers. However, this work is limited to a single set of environmental conditions. Gupta and Dasgupta (2010) reported simulation based performance evaluation of air cooled finned tube gas coolers in Indian context. They carried out simulations to see the sensitivity of the gas cooler performance against the variation in operating temperature, pressure, air velocity and various other physical design parameters. Fronk and Garimella (2010) reported both experimental and theoretical work on compact micro channel water cooled CO_2 gas cooler. They also addressed optimization of the model. Yu et al. (2012) developed a tube in tube heat exchanger model applicable for supercritical CO_2 and water. In their work, effect of inlet pressure on variation of heat transfer rate was found to be prominent. Sanchez et al. (2012) investigated coaxial gas cooler using finite element modeling, taking water as coolant. In their study several correlations for estimating the CO_2 heat convection coefficient have been studied to aid selection of the best fits and influence of the various parameters on gas cooler thermal effectiveness. Yu et al. (2014) conducted experimental study to investigate the performance and heat transfer process in tube in tube counter flow heat exchanger. They reported that the local heat transfer rate of heat exchanger peaks near the pseudo critical region due to drastic rise of specific heat. Zhang et al. (2015) investigated mixed convection heat transfer in supercritical CO_2 flowing vertically upwards in a helically coiled tube. The coupling effects of the buoyancy force, centrifugal force, and variations in the physical properties were investigated to determine the temperature and heat transfer coefficient distributions along the circumference. Gupta and Dasgupta (2014) reported simulation and performance optimization of finned tube gas cooler for trans-critical CO_2 refrigeration system in Indian context and proposed some design guidelines for ambient temperature within range 30°C to 55°C. Ge et al. (2015) studied both lumped and distributed models for an air-cooled CO_2 gas cooler. The effect of coil sizes and pipe arrangements on system performance was evaluated. To maintain a fixed cooling capacity at various ambient conditions, necessary conditions for a larger size gas cooler was suggested. Marcinichen et al. (2016) presented investigation focusing on reducing the volume of CO_2 gas cooler leading to reduction in charge of refrigerant in circuit. They reported up to maximum of 14% reduction in charge of refrigerant in case of adoption of proposed optimized design of gas cooler. Rossetti et al. (2018) reported numerical investigation on CO_2 finned tube gas cooler using equivalence modelling. They reported three dimensional effects of the air maldistribution on the temperature, heat transfer coefficient and cooling power.

2.2.3 Expansion valve/Capillary tube

The basic device for expansion in a CO_2 refrigeration system is a manual or electronic expansion valves. Capillary tubes are also well suited for small capacity applications. Casson et al. (2003) proposed expansion system with differential valve, liquid receiver and a thermostatic valve. Their simulation results highlighted self-adjusting capability even at temperature variation of the secondary fluid. One of the early paper investigating capillary tubes for expansion process in the CO_2 based refrigeration system was by Madsen et al. (2005). Various lengths and diameters of capillary tubes were tested. They reported that capillary tubes were advantageous when the evaporation temperature is constant and that the gas cooler outlet temperature does not deviate much from design condition. Huang et al. (2007) presented experimental investigation on expansion valve. They recommended avoiding oversize of downstream line of safety valve and installation of device to eliminate static electricity in downstream line. Agarwal and Bhattacharya (2008) presented a steady state model developed to evaluate performance of capillary tubes for simultaneous heating and cooling system. The system performance was reported to be marginally improved with a capillary tube at higher gas cooler exist temperature. A guideline for selection of optimum capillary tube size was also reported. Wang et al. (2012) developed a separated flow model of coiled adiabatic capillary tubes. The results helped formulating modified friction factor for better and reasonable predictions. Assessment of common correlations for calculating mass flow rate through capillary tubes in trans-critical cycles was done by Cecchinato et al. (2009). They replaced the expansion valve with capillary tubes for high pressure application, to optimally respond to changes in gas cooler heat sink temperature. Capillary tubes of various diameters were experimentally investigated under wide range of inlet pressures and temperatures to obtain a more reliable design. Da Silva et al. (2009) published a study on expansion of trans-critical CO_2 through adiabatic capillary tubes. They investigated the effect of operating conditions and the tube geometry on the CO_2 mass flow rate. They concluded that the trans-critical CO_2 flow is nearly similar to the subcritical capillary tube flows, except for the regions in the vicinity of the critical point. Da Silva et al. (2011) also reported investigation on diabatic capillary tubes for trans-critical refrigeration cycle. They studied the effect of operating conditions and the tube geometry on the CO_2 heat and mass flow rates. Further, a mathematical model for predicting mass flow rate was presented and validated against experimental data, and a good agreement between experimental and computed values of mass flow rates was claimed with error margin below 10%. In addition, the model could estimate the suction line outlet temperature within $\pm 3°C$ accuracy. They also concluded that, the heat exchanger dimensions and its position played a dominant role on the heat flux. Trinchieri et al., (2014) investigated three capillary tubes of various lengths to optimize the dimensions of tube for a cooling load. They also claimed that with appropriate refrigerant charge optimization, it was possible to optimally utilize capillary tubes as expansion device even for a large range of operating conditions. Baek et al. (2013) investigated control method of gas cooler pressure using electronic expansion valve (EEV) and outdoor fan speed. Hou et al. (2014a) investigated mass flow rate characteristics of supercritical CO_2 flowing through an electronic expansion valve (EEV). They reported that it is possible for the systems to have both choked and non-choked flow due to the influence of outlet pressure of EEV on the mass flow rate. In the same year, another study by Hou et al. (2014b) was reported highlighting influence of EEV opening on the system performance. They reported that EEV opening has profound effect on the discharge from the compressor and subsequently on gas cooler outlet pressure. Song et al. (2017) reported investigation on adaptability of a capillary in a CO_2 trans-critical refrigeration system. They reported COP of system with capillary to lie in range from 82% to 98% in relative to system with EEV. Consequently, they suggested to adopt capillary in CO_2 refrigeration system which is simpler in construction, low cost and non-controller in comparison to EEV. The effect of inlet pressure as well as temperature, outlet pressure, EEV flow area as well as geometry on the CO_2 mass flow rate characteristics in EEV was investigated by Liu et al. (2018). They concluded that the influence of needle degree on the CO_2 mass flow rate is significant and cannot be neglected. They also developed mathematical relation to accurately predict the CO_2 mass flow rate through an EEV.

2.2.4 Evaporator

Many researchers like Kim and Bullard (2001), Elbel and Hrnjak (2004) etc. took early initiative to focus on evaporator for CO_2 trans-critical system. They conducted analytical study for different design of evaporators. Kim and Bullard (2001) developed a detailed finite volume model for a multi-slab micro-channel evaporator for a CO_2 mobile air conditioning system and validated the model for a two-slab prototype evaporator to study air side phenomena such as the effects of condensate and inclination angle. Kim et al. (2004) reported that the overall size required for CO_2 micro-channel evaporator with louvered fins is less than that of an R134a evaporator. To avoid maldistribution of the two-phase flow in micro-channel of the evaporator, Elbel and Hrnjak (2004) proposed a flash gas by pass system. Cho and Kim (2007) experimentally investigated in tube evaporation heat transfer characteristics and reported that the average heat transfer coefficient improvement for micro fin tube was better compared to the smooth tube. Onaka et al. (2010) investigated dimethyl ether and CO_2 mixture as refrigerant for trans-critical cycle. They reported that in tube evaporation heat transfer coefficient decreased with increase in CO_2 volume fraction. Jin et al. (2011) developed and validated finite volume model for the CO_2 mobile air conditioning system. Effects of selecting various refrigerant side heat transfer and pressure drop models on the performance of evaporators were discussed. Fang et al. (2013) reviewed and analyzed 34 existing correlations (heat transfer coefficient) using 2956 experimental data points and reported discrepancy in the existing correlations. Patino et al. (2014) did comparative analysis of a CO_2 evaporator model using two approaches, one of them utilizing classical empirical correlations approach and other using flow pattern mapping approach. They reported that flow pattern mapping approach to be better. Yamasaki et al. (2017) experimentally investigated CO_2 dry ice behavior in an evaporator. They reported that for ultra-low temperature application, adoption of swirl promoters is necessary to prevent down-stream blockage of the channel.

2.3 CO_2 refrigeration system performance enhancement

2.3.1 Internal heat exchanger (IHX)

The concept of adoption of IHX in basic trans-critical CO_2 cycle was first reported in an experimental work by Lorentzen and Pettersen, (1993) on a car air-conditioning system. Large number of investigations subsequently have been reported on CO_2 refrigeration cycle with IHX, which include both theoretical as well as experimental study. The key parameters investigated include length of IHX, effectiveness of IHX and effect of IHX on overall cycle efficiency. IHX guarantees superheated vapor at the compressor inlet. Further, in sub-critical mode of operation, it ensures that refrigerant enters in liquid state to the expansion valve. Two prominent and competing effects are associated with introduction of an IHX in the cycle, which critically affects the cycle performance. Firstly, it enhances proportion of liquid at the inlet of evaporator, resulting into increase in specific cooling capacity. Secondly, it contributes towards increasing superheat at the low-pressure side, which leads to higher compressor discharge temperature and increase in specific power consumption. However, with increase in suction temperature at compressor inlet, the refrigerant mass flow rate in the circuit reduces as reported by Domanski et al., (1994). Boewe et al., (2001) experimentally investigated CO_2 refrigeration cycle with IHX for mobile air-conditioning application. They reported up to 25% improvement in COP over the basic cycle. The effect of IHX length on the performance of CO_2 refrigeration cycle was analytically investigated by Rozhentsev and Wang, (2001). Liu et al., (2005) experimentally investigated a CO_2 automotive direct expansion air conditioning system. They reported that system COP is highly sensitive to lubricant property, refrigerant charge and high side pressure. Cho et al., (2007) experimentally investigated the effect of electronic expansion valve (EEV) and IHX on the performance of IHX cycle. They reported a maximum of 9.1% and 11.9%

improvement in COP and refrigeration effect, respectively. Cabello et al., (2008) presented results from an experimental evaluation of a single stage CO_2 refrigeration cycle equipped with IHX for warm climate. They reported that under-estimation of the optimal gas cooler pressure tends to reduce the energy efficiency at considerably higher rate compared to over-estimation of gas cooler pressure. An experimental investigation of trans-critical CO_2 spilt-system with IHX for residential air conditioning was reported by Aprea and Maiorino (2008). The authors reported increase in COP due to IHX. Aprea and Maiorino (2009) also compared the performance of the same system with sub-critical R134a refrigeration system. They reported that development of robust control system for the gas cooler heat rejection pressure are necessary for success of large scale spilt systems. Rigola et al., (2010) extensively studied the effect of adoption of IHX in the basic CO_2 cycle using an experimentally validated numerical model. They reported improvement in performance of cycle with IHX and that the extent of improvement tends to increase with increase in ambient temperature. An experimental investigation of CO_2 residential air conditioning system with IHX was reported by Tao et al., (2010a). In their investigation, the effect of inlet conditions at gas cooler side and the evaporator setting were found to be the main influencing parameters. They reported up to 6% improvement in COP within the range of investigation. In another study by Tao et al., (2010b), exergetic analysis was reported based on experimental data. They observed about 30.7% of the total exergy loss in gas cooler, which is maximum among components, while, the minimum was 3% occuring in IHX. Sánchez et al., (2010) experimentally investigated effect of operating parameters on the compressor superheat. They reported that the extent of superheat tends to increase with increase in discharge pressure and decrease in evaporator temperature. Torrella et al., (2011) experimentally investigated the effect of evaporator temperature as well as ambient temperature on the performance of CO_2

trans-critical refrigeration cycle having IHX. They reported that the energy efficiency of the cycle with IHX tends to rise with rise in ambient temperature. A maximum of 12% rise in COP was recorded for ambient temperature 34°C at evaporator temperature of -15°C. Aprea et al., (2013) presented an experiment based comparative study of exergetic performance of R134a system vs a CO_2 system. The system performance with R134a was reported better for the range of ambient temperature investigated (20°C to 45°C). Further, it was reported that irrespective of the fluid, IHX is able to augment the exergetic efficiency of the system. De Carvalho et al. (2015) investigated a two-stage variable capacity CO_2 refrigeration system equipped with IHX. They suggested that for a wide range of compressor speed, a cycle without IHX operates satisfactory with a single combination of refrigerant charge and expansion restriction. It was also reported that the cycle with IHX performs better only at lower compressor speed. Singh et al., (2016) reported that a system with IHX has lower mass flow rate and perform better, especially at high ambient temperature. Ituna-Yudonago et al., (2017) presented a simulation based study on the behaviour of IHX in a CO_2 refrigeration cycle under variable boundary conditions for both steady state and transient state operations. They reported that, the transient period is much higher when the inlet temperature is lower and that the transient period is inversely proportional to the thermal effectiveness of the IHX.

2.3.2 Ejector expansion

An ejector-expansion device can effect significant reduction in expansion losses of the basic trans-critical cycle (Ahammed et al., 2014 a&b). Li and Groll (2006) reported experimental study to evaluate ejector as an expansion device in trans-critical cycle. They constructed a mathematical model to investigate the effect of ejector design parameters on the system performance for application in U.S. Military Environmental Control Unit (ECU). They concluded that system with ejector has about 11% higher COP and 9.5% higher cooling capacity over the basic trans-critical system. Drescher et al. (2007) experimentally investigated CO_2 trans-critical system using ejector as expansion device. They reported reduction in operating pressure and improvement in performance. Liu and Groll (2008) designed and constructed a controllable ejector expansion device for a trans-critical CO_2 experimental air conditioning system. They reported optimum value of diameter for the ejector motive nozzle throat and the mixing section. They also concluded that the suction nozzle efficiency depends on the suction nozzle inlet pressure and ejector throat area. The possibility of overcoming throttling losses by introducing ejector in the basic cycle was explored by Elbel and Hrnjak, (2008). Their prototype ejector could recover up to 14.5% of the throttling losses. An experimental study on trans-critical CO_2 based two phase ejector system was reported by Masafumi et al., (2009). The authors claimed that the performance of the cycle is significantly influenced by its operating temperature and pressure. Further, they reported substantial improvement in COP over the basic cycle using ejector. In the same year, an investigation on an Isentropic Homogenous Equilibrium (IHE) two-phase supersonic flow of CO_2 through ejector was reported by Nakagawa et al. in 2009. They concluded that the decompression phenomenon and the IHE solution in nozzle were a function of both inlet temperature and divergence angle of the diffuser. Nakagawa et al. (2011) investigated the effect of mixing length of the two-phase ejector on the performance of system equipped with IHX. They reported that mixing length plays a vital role on system performance. COP improvement over conventional system was demonstrated through the optimized mixing length. It was also observed that, improper sizing of mixing length can even lower the COP. Lee et al. (2011) investigated air conditioning system using ejector as expansion device. They reported 15% enhancement in performance of the system. Liu et al. (2012) reported an attempt towards enhancement of the performance of the system equipped with variable frequency compressor using a controllable ejector. They reported that ejection ratio, suction pressure ratio, cooling capacity and COP reaches their maximum level when the distance between ejector motive nozzle exit and mixing section entry is about 3 times the mixing section diameter. Banasiak et al. (2012) performed an experimental and numerical investigation of the optimum ejector geometry. Based on a simplified, one-dimensional ejector model, an optimization of the ejector geometry was obtained and the ejector efficiency was found to be dependent on the mixer length and the diameter, as well as on the diffuser divergence angle. Maximum increase in COP of 8% over a system with a conventional expansion valve was reported. Lucas and Koehler (2012) investigated trans-critical CO_2 cycle using ejector as an expansion device. They studied system performance at various inlet conditions of ejector for a range of evaporator and gas cooler outlet temperatures and reported 17% improvement in COP with an ejector efficiency of 22%. Lawrence and Elbel (2014) reported a comparative study of performance of CO_2 and R134a in two-phase ejector cycle for automotive air conditioning applications. They concluded that, ejectors in CO_2 cycle are more effective than in R134a cycle. They also discussed the possible reason for the difference in ejector performances between the two refrigerants. Lee et al. (2014) improvised the same system using ejector having an entrainment ratio greater than 0.76 and reported enhancement in performance. They also suggested a control strategy for ejector refrigeration system. Liu et al. (2016) examined simultaneous heating and cooling performance of an ejector expansion CO_2 trans-critical system. They showed that the cycle could be operated optimally by adjusting the ejector internal geometry and the compressor frequency according to indoor and outdoor air temperatures. Zheng and Deng (2017) reported experimental investigation on a trans-critical CO_2 ejector expansion refrigeration system equipped with two stage evaporation. Adoption of two stage evaporator lead to improvement in COP as well as cooling capacity. In addition, effect of variable area ratio on entrainment ratio was equal to the effect of change in cooling load of the evaporators.

2.3.3 Work recovery expander

The concept of incorporation of a work recovery expander in place of an expansion valve was proposed by Lorentzen (1994). Thereafter, many researchers have investigated the expander technology both theoretically and experimentally, reviews on the same can be found in literature (Zhang et al., 2013; Singh and Dasgupta, 2016). Work from expander can be recovered and utilized either by coupling with the compressor drive shaft using a suitable mechanical linkage or by generating electricity out of expander work, which then may be indirectly used to compensate for the power demand of the cycle.

Heyl and Quack (1999) introduced a free piston expander which had full pressure expansion operation and they termed it as first-generation expander. The design was further improved to second generation free piston expanders incorporating a cylinder connected with double rocker arm (Nickl et al., 2002). A third-generation expander with three stage expansion together with one stage compression was presented by Nickl et al., (2003). In the year 2005, system was further modified by incorporating a liquid–vapor separator vessel between the second and third stage of expansion (Nickl et al., 2005). With improvement in cycle, the COP of the system progressively improved. Baek et al. (2005) investigated a piston-cylinder work producing expansion device and demonstrated 10% improvement in system performance by using expander in place of the expansion valves in a trans-critical CO_2 cycle. Riha et al. (2006) investigated expander and compressor as a unit and reported that for successful operation of the unit with fluctuating load, a few valves are needing to be added to the cooling system stream to guarantee safe operation. Peng et al. (2006) tested and validated the design of a slider-based free piston expander, in which the slider was utilized to control the expander inlet and outlet flow. The experimental studies showed that the isentropic efficiency of the expander could reach up to 62%. Further investigation showed that the design lacked robustness, which resulted in serious vibration problem at higher frequencies. Fukuta et al. (2006a) reported experimental investigation on a scroll type work recovery expander. They developed expander using scroll compressor for a water heater. It was reported that leakage reduces the volumetric efficiency of such systems and a maximum of 55% system efficiency was achieved. In the same year, Fukuta et al. (2006b) reported design of a vane type work recovery expander to drive a secondary vane type compressor. An intercooler between a first-stage compressor and a second-stage compressor was also provided which reduced the second-stage compressor work. Operating characteristics of the combined expander and compressor unit was determined. Zeng et al. (2007) tested a rolling piston expander and reported that the P-V curve agrees with experimental results and they also emphasized the importance of expansion ratio as it affects systems performance. Yang et al. (2008), developed and investigated a vane type expander for the trans-critical cycle. They found that the leakage through the seal arc was about 41% of the total leakage and that the other leakages were mainly associated with the end surface gaps and the vanes. They also measured volumetric efficiency of the expander at various speeds and concluded that the same increases with increase in rpm. Fukuta et al. (2008) reported CO_2 trans-critical expansion process through vane expander including detailed investigation on inflection points using single piston expander equipped with glass windows. The inflection point is that where the expansion process enters two-phase region. They concluded that it takes relatively long time for inflection or there is delay in flash during trans-critical expansion process for low initial expansion temperature. The performance of the vane expander and expansion process was investigated further by Fukuta et al. (2009). The authors reported that the prototype of vane expander had large leakage loss and the performance could be improved by supplying a vane back pressure. Yang et al. (2009a) introduced springs in the vane slots to ensure tighter contact between the vane and the cylinder wall. They also introduced seal arc to reduce the leakage from the inlet port to the outlet port. The new design lead to increase in volumetric and isentropic efficiencies, however, introduced more vibration. Later, Yang et al. (2009b) investigated the vane dynamics to understand the vane movement and the contact conditions between vane tip and the cylinder wall. They concluded that the centrifugal force was much lower than the gas force and that the same had little influence on the vane movement in the slots. They identified that the impact of the charging gas on the vanes nearer the suction port was responsible for the loss of contact between vanes and surface of cylinder wall, which in turn necessitated further investigation of the suction port design. To overcome vane movement problem, Jia et al. (2011) introduced high pressure gas at the bottom of the vane slots as support. This neutralized impact of the charging gas, also the adverse effects of vibration and friction losses were avoided, which was prominent in the spring loaded design. Tian et al. (2012) reported a leakage test conducted on rolling piston expander and concluded that the lubricant film thickness plays an important role in estimating the leakage. They included four classic leakage models for study and concluded that the laminar leakage model is the one most suitable for present application. Fukuta et al. (2013) investigated expansion process in detail by utilizing a high-speed camera. They also monitored heat transfer coefficients within expansion chamber with a range 2-7 kW m^{-2} K^{-1} in two phase region. A black-out phenomenon was reported near the critical point during expansion process. Subiantoro and Ooi, (2013) presented an economic analysis in favor of adoption of work recovery expander in a basic vapor compression cycle. They reported that expanders are suitable for CO_2 based cycles and have a relatively short payback period. Local electricity tariff and isentropic efficiency of work recovery expander are reported to be the two most prominent factors affecting the payback period. In the year 2014, a small capacity device with reciprocating piston type expander was examined by Fukuta et al., (2014). The authors concluded that the supply and discharge of refrigerant by reciprocating motion of the piston and the expander can be driven and operated with a small flow rate. Hou et al. (2014c) designed and tested a micro-turbo expander having refrigeration capacity of 15 kW. Their study showed that replacing the throttle valve with an expander can decrease the exergy loss and improve the COP. Liu et al., (2017) investigated six different two-stage CO_2 trans-critical refrigeration configurations equipped with work recovery expander. Configuration with two-stage expander and intercooling was reported to be better in terms of performance and equipment selection.

2.3.4 Dedicated sub-cooling

Installing a separate dedicated sub cooler, which provides sub-cooling of the high temperature CO_2 coming out from the gas cooler of the basic CO_2 cycle is also a prominent modification. Schoenfield et al. (2012) investigated CO_2 trans-critical refrigeration system with thermoelectric sub-cooler. They claimed substantial improvement in refrigeration capacity and overall cycle COP. Sarkar (2013) analytically investigated CO_2 trans-critical refrigeration system equipped with thermo-electric sub-cooler. He reported 25.6% improvement in COP and 15.4% reduction in operating gas cooler pressure. Llopis et al., (2015a) theoretically investigated adoption of dedicated mechanical sub-cooler in a CO_2 refrigeration system. They reported maximum of 20% improvement in COP and 28.8% improvement in cooling capacity. In the next year, Llopis et al., (2016) experimentally evaluated the performance of CO_2 refrigeration cycle with and without mechanical sub-cooler and confirmed the improvement in both COP and cooling capacity. Bush et al., (2017) reported transient behaviour of CO_2 refrigeration system with dedicated mechanical sub-cooler through analytical study. Llopis et al., (2018) has presented comprehensive review on sub-cooling technologies applied especially to the CO_2 refrigeration system,

2.3.5 Flooded evaporator

The concept of flooded evaporator essentially ensures optimal use of heat transfer area and has been explored by many researchers to improve the evaporator performance. Aganda et al., (2000) observed that conventional evaporators do not permit optimal use of heat transfer area and a part of evaporator is always dominated by single phase vapour flow. The situation improves when a flooded evaporator is used, it ensures wet region throughout the flow passage leading to substantial increase in heat transfer. This also enables the system to be operated at a higher evaporation temperature. Precise control of refrigerant mass flow through the evaporator is also possible without using expensive electronic expansion valves (EEV) in flooded evaporator configuration. However, a flooded system needs an additional pump or a two-phase ejector to recirculate the refrigerant. A trans-critical CO_2 system with flooded multi-evaporator and ejector system was investigated by Minetto et al. (2014). Performance of this system was compared with dry-expansion evaporator systems and substantial energy savings and smoother operation were claimed. Karampour and Sawalha (2018) presented elucidated study of practical implementation of various techniques to flood the CO_2 evaporators, including adoption of liquid ejector, pump and IHX.

2.3.6 Twin-staging

Twin-staging is commonly employed in refrigeration systems when there is a large difference between evaporating and condensing temperatures. Although employing twin-staging increases the initial cost over single-stage systems, it alleviates a few practical problems and can save on total compressor work. In twin-stage compression, the refrigerant flows in series through two or more compressors. Intermediate pressure is an important and critical parameter which affects the performance of a twin-stage system. The performance of CO_2 systems is enhanced considerably by adopting twin-staging as the pressure difference across the CO_2 compressor is large. Common variants of twin-stage cycles are two-stage compression inter-cooling cycle (IC), two-stage flash gas bypass cycle (FGB), two-stage flash inter-cooling cycle with flash gas removal (FGI) and two-stage parallel compression cycle (PC).

Elbel and Hrnjak (2004) compared conventional direct expansion (DX) system with flash gas bypass (FGB) system experimentally. They reported improvement in both cooling capacity and COP by 9% and 7% respectively using a FGB system. Hwang et al. (2004) also investigated performance of CO_2 cycles with a two-stage compressor. Total four different cycle options were tested, these are: a basic cycle, a cycle with a suction line heat exchanger, a cycle with an intercooler, and a two-stage split cycle. It was reported that the two-stage spilt cycle, the cycle with intercooler and the cycle with suction line heat exchanger shows 35%, 22% and 18% improvement respectively in COP, over the basic CO_2 cycle at 7.2°C evaporator temperature. Bell (2004) reported a promising modification in CO_2 system by incorporating a parallel compressor for handling the vapour generated at the intermediate pressure vessel up to the gas cooler pressure. Based on theoretical and experimental investigation, he observed improvement in the cycle COP over a wide range of operation. An experimental study on two stage trans-critical CO_2 system was reported by Cavallini et al. (2005). Experiments were conducted for various gas cooler outlet pressures, while keeping a few variables fixed like evaporator pressure, evaporator outlet superheating and gas cooler outlet temperature. For typical air conditioning application dependence of the system performance on investigated parameters was reported. Cho et al. (2009) investigated the performance of a two-stage trans-critical CO_2 system with gas injection. Improvement in performance of the two-stage system is reported due to gas injection and inter-cooling. But at the same time, mass flow rate in the gas cooler increases. Hence, authors emphasized that in a gas injection cycle, optimum control by EEV is required. Sarkar and Agrawal (2010a) presented a theoretical study and discussed optimization of parallel compression cycle. They reported that the parallel compression technique is more effective at lower evaporator temperature and claimed improvement of up to 47.3% in cycle performance. Wang et al. (2011) investigated a cycle with two gas coolers and another cycle with an intercooler. They reported that performance of intercooler is better than that with double gas cooler model for the tested conditions. A two-stage compression CO_2 cycle with high back pressure was investigated by Qi et al. (2011). The high back pressure was useful to have better lubrication and to lower the oil flow rate. It was concluded that for a two-stage compressor, there exists an optimum compression ratio for each suction pressure. Fabrication of a multipurpose test rig system was reported by Chesi et al. (2012). It was equipped with two double stage compressors, a multi tube-in-tube heat exchanger as gas cooler, an evaporator and a few EEVs. The motive was to evaluate the performance of IHX equipped with a passive load management system. Several layouts were investigated, that include multi-stage compression, economizer, flash vapor by-pass and parallel compression. They also attempted to address some of the vital issues related to overall plant design strategy including component design, correct choice of measuring instruments and thermal management system. A trans-critical CO_2 system using a configuration of parallel compression with flash tank was investigated by Chesi et al., (2014). They isolated the parameters that influenced system performance in an actual cycle and identified the conditions required for a typical commercial refrigeration application. The CO_2 refrigeration system was tested sub critically and it was reported that if the heat exchanger was designed as a condenser and operated as a gas-cooler, the energy performance of the refrigeration plant could improve under certain subcritical and supercritical conditions (Sánchez et al., 2014). Fazelpour and Morosuk, (2014) reported an exergo-economic analysis of CO_2 trans-critical cycle. Based on their study, use of economizer was suggested to be beneficial for warm climatic operations. However, Sharma et al., (2014a) reported that the performance of CO_2 booster system with parallel compressor was comparatively lower than that of R404A direct expansion system at higher ambient temperature. Fritschi et al. (2016) fabricated a fully instrumented test rig to compare the conventional carbon dioxide refrigeration machine (flash gas injection) with parallel compressor. They reported optimum intermediate pressure between 45-50 bar for evaporation temperature above -10°C. They concluded that with increase in gas cooler outlet temperature the parallel compressor system in CO_2 trans-critical cycle outperforms the conventional system. Rahmati and Gheibi (2017) reported investigation on two-stage trans-critical CO_2 refrigeration cycle with multi inter-cooler. Based on exergy perspective, they reported maximum of about 31% exergy destruction rate in compressor. In addition, adoption of IHX was reported to be beneficial especially at elevated ambient.

2.3.7 Cascading

In cascade configuration, low-GWP refrigerants such as R290, R1234ze(E) and R717 are employed in the primary circuit, while CO_2 is in secondary circuit. The modification allows benefits in terms of both energy and environmental perspective as compared to conventional systems. Getu and Bansal (2008) formulated a multivariate relation for NH_3/CO_2 cascade system to estimate maximum COP and optimal mass flow rate ratio of refrigerants in the primary circuit to the secondary one. A NH_3/CO_2 cascade system was investigated by Bingming et al., (2009) with CO_2 in the lower temperature side and its performance was found better than single or two stage NH_3 systems. Yamaguchi and Zhang (2009) constructed CO_2 refrigeration system for a novel application. The proposed system could achieve refrigeration below the CO_2 triple point of -56.6°C. The system consisted of one trans-critical and another trans-triple point cycle arranged in cascade. They found reasonable COP for investigated cases. Dry ice particles were also traced in the circuits and owing to this problem, Yamaguchi et al. (2011), specifically investigated dry ice blockage in the CO_2 solid-gas flow based cascade system. They also conducted visualization study of the liquid CO_2 flowing in an expansion tube through a throttle needle valve. It was concluded that to prevent sedimentation of dry ice, the system needs to be operated at heating power above 900 W and condensation temperature above -20°C. Rezayan and Behbahaninia (2011) reported that the performance of NH_3/CO_2 cascade system is influenced by condensing temperature of NH_3, evaporating temperature of CO_2, condensing temperature of cascade condenser and approach temperature of cascade condenser. An experimental setup on NH_3/CO_2 cascade system was designed and built as reported by Dopazo and Fernández-Seara (2011). They reported up to 19.5 % higher COP for cascaded configuration over the two-stage NH_3 solution. Fernández-Seara et al. (2012) investigated the freezing process in plate freezer of the same system. It was concluded that CO_2 in low freezing process can very well serve as alternative to conventional refrigerants. Da Silva et al. (2012) explored a cascade refrigeration system for supermarket application. They compared the performance of the cascade system with direct expansion system using conventional refrigerants, R404A and R22 employed in the high-pressure side. They observed reduction in refrigerant charge and electric energy consumption for the cascade system. A cascade system (R134a/CO_2) was also investigated to operate at low evaporation temperature for commercial applications by Sanz-Kock et al. (2014). The performance of the system was analyzed by observing compressor performance, cooling capacity and COP. It was reported that CO_2 as refrigerant is well suited for low temperature application. Zhang et al. (2015) reported an investigation on the performance of CO_2/propane auto-cascade refrigerator. They utilized a fractionation heat exchanger (FHEX) to improve the separation efficiency of CO_2 and propane. The FHEX handles both phase separation and heat exchange processes. It was concluded that the cycle with FHEX perform better at all investigated conditions. They further reported that cycle performance is directly proportional to the CO_2 mass flow rate and inversely to the cooling water temperature. Mosaffa et al. (2016) conducted exergoeconomic and environmental analyses for two different NH_3/CO_2 cascaded refrigeration systems, the first one equipped with two flash tanks and the second with a flash intercooler. They reported almost identical performance for both the options. Sanchez et al. (2017) reported investigation on conversion of a direct to an R134a/CO_2 cascade refrigeration system based on energetic perspective. They reported variation of the energy consumption of the converted system to lie between -0.3% to 14 % as compared to direct expansion system for the range of parameters.

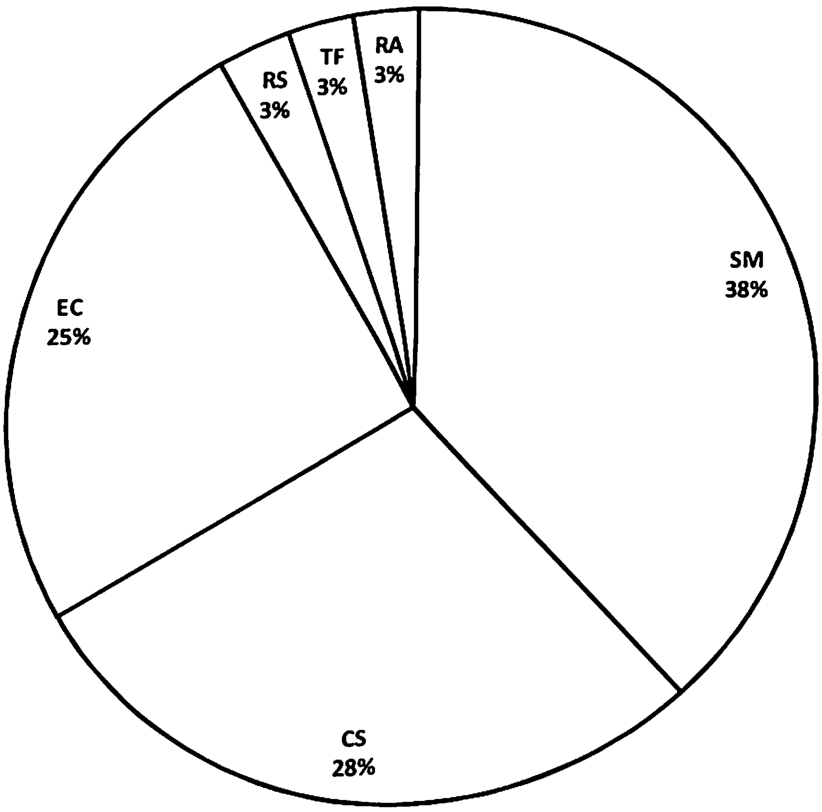
2.4 Application and commercialization

Synergetic effect of research output in trans-critical CO_2 system and demand from commercialization, is boosting each other and the same is evident from increasing number and variety of publications and large number of commercial CO_2 trans-critical installations in recent years. In 1996, at Bad Hersfield in Germany, the first CO_2 A/C system was installed in a bus owned by a public transport company. This research work was a novel initiative of Konvekta research and development center.

Further, in the year 1997, two new buses, one equipped with CO_2 A/C system and another with R-134a A/C system were introduced in the German transport services. Both the buses were identical with respect to their weight and overall size. By the end of one year, performance of both were compared. Trans-critical cycle with EEV was found equally effective with the R-134a cycle for the application. They also reported that incorporation of IHX in the system improves cooling capacity. Further, they elaborated some steps involved in the design of the compressor (Kohler et al., 1998). McEnaney et al. (1998) reported a comparative study on CO_2 mobile A/C systems against conventional R134a systems based on experiments and concluded that, at outdoor ambient temperatures below 40°C, the performance of CO_2 system exceeds that of the baseline R-134a system. Mu et al. (2003) investigated trans-critical CO_2 air conditioning system. The authors claimed construction and testing of the first CO_2 system prototype for domestic application that also attained design capacity during performance test. Sarkar et al. (2004) based on their steady state analysis reported importance of optimum gas cooler pressure to achieve maximum COP for simultaneous cooling and heating applications. Jakobsen et al. (2004) tested a reversible CO_2 residential air conditioning system. They compared the performance of CO_2 based system with a R410A unit and reported comparable performance for ambient temperature in the range 27.8°C and 35°C. Memory et al. (2005) developed a High Mobility Multipurpose Wheeled Vehicle (HMMWV) for military purposes and investigated its cooling functionality using CO_2 as refrigerant. The system performance was compared with base line R134a system and it was concluded that CO_2 is a favorable alternative for the cooling application. Automotive CO_2 air conditioner system was investigated by Liu et al. (2005). They studied system performance and concluded that the same is sensitive to refrigerant charge. The need of an appropriate high side pressure controller was also stressed by them. A paper highlighting parametric effect on the performance of trans-critical CO_2 refrigeration system was reported for mobile air conditioning application by Kim et al., (2009a). They developed an algorithm for computing the optimum high pressure to achieve the maximum COP. In the same year, (Kim et al., 2009b) evaluated the performance of trans-critical CO_2 refrigeration system employed for stack cooling system for mobile air conditioning applications in fuel cell vehicles. Both heat releasing effectiveness and mutual interference, were analyzed and the same was compared with conventional cooling systems with and without involvement of cabin cooling. They reported 36% increment in heat release, after employing trans-critical CO_2 refrigeration system. Furthermore, they also suggested a control logic for stack cooling and cabin heat control during peak load. Kim et al., (2010) proposed a new design for stack cooling system. The experimentation was carried out under various realistic operating conditions with respect to stack cooling, cabin cooling and combined cooling operations. The intention was to support the re-design of stack cooling system and thereby to enhance the stack power generation and efficiency. Cecchinato et al. (2010) investigated trans-critical CO_2 chiller for commercial refrigeration. Based on the study, the authors suggested modifications in system architecture and control logics to overcome the high pressure associated with carbon dioxide as refrigerant. They also suggested control logic to optimally operate the cycle in sub-critical or trans-critical mode according to the ambient conditions. Cecchinato and Corradi (2011) developed and tested a single door CO_2 based bottle cooler and compared the same with systems operated with R401A and R134a as refrigerant. Comparable performance was reported with CO_2 system for application at medium to low gas cooler inlet temperature. An experimental setup for investigating trans-critical CO_2 system for household refrigeration was fabricated by Boccardi et al. (2013). The system was equipped with two hermetic reciprocating compressors, a finned coil type heat exchanger and a double expansion system. The authors also reported a plant performance map as a function of ambient and evaporator (cabinet) temperature. Suamir and Tassou (2013) evaluated a tri-generation system, using CO_2 as refrigerant for tri-generation, a co-generation concept for efficient utilization of source (solar energy), where simultaneous cooling, heating and electricity generation are explored. They demonstrated that significant energy savings can be achieved using such hybrid system. Various alternative configurations were explored by them and ultimately a system utilizing cooling produced from sorption system to condense CO_2 was recommended. Elbel et al. (2014) reported development of CO_2 based beverage display coolers and compared the same with performance of R134a based systems. They successfully implemented and validated the CO_2 based technology and showed that there is scope of improvement of the overall system with further improvement in individual component's performance.

Tremendous growth in commercialization is observed in recent years and this in-turn has brought funding for application focused research. Various industrial houses and entrepreneurs have started business in the area of manufacturing, supplying, construction and commissioning of such systems. A website, R-744.com maintains an up-to-date list, which at present shows more than

350 companies that are involved in business in allied areas worldwide. Percentage wise distribution of the industries in the various sub-areas of the business are shown in Fig. 2.2, as of June 2018. We observe that booster system and cascade system-based CO_2 refrigeration cycles have found better commercial acceptance. Two major cycle modification strategies namely ejector expansion and parallel compression are at present being incorporated into commercial systems. To comprehend the extent of commercialization, as per the information to date, more than seven thousand trans-critical CO_2 refrigeration systems are already in operation worldwide. Majority of such systems are in colder countries in Europe, followed by America and Asia. Some developing countries like Brazil, Argentina, South Africa and India are also looking forward to installation of trans-critical CO_2 based refrigeration systems.



SM=System manufacturer, CS= Component supplier, EC= Engineering & contracting, RS=Refrigerant supplier, TF=Training & formation, RA= Research & academia

Fig. 2.2 Percentage wise distribution of the industries in various business areas.

2.5. State of art of CO_2 refrigeration system in supermarkets

Supermarkets have massive refrigeration, air conditioning (A/C) and heating load that consume approximately 50% of the supermarket energy consumption (IIR, 2015) and significantly contributes towards carbon emissions. Moreover, supermarkets employ large tonnage of HFC refrigerants having high Global Warming Potential (GWP) leading to substantial environmental worries. Various environmental regulations are being enforced to eliminate wholly or partially the use of HFCs. Complying to the changing regulations is a persisted challenge for supermarkets. A possible shifting over to all-natural refrigerant can potentially provide a permanent solution to the problem. CO_2 has been employed in indirect, cascade and booster configurations for supermarket refrigeration. CO_2 refrigeration system with integrated A/C and heating loads are also finding acceptance worldwide. More than 11000 installations (Shecco, 2016) are already made worldwide, mostly the booster configuration, in cold climate, owing to their cost comparison advantage with respect to conventional systems (Zeiger et al., 2016) and their capability to recover heat (Sawalha, 2013).

2.5.1 CO_2 booster system

CO_2 booster systems simultaneously satisfies medium temperature (MT) as well as low temperature (LT) loads for example chilled food display cabinets and frozen food storage respectively. Booster systems are equipped with low and high stage compressors. The low stage compressor operates sub-critical while the high stage compressor operates either in sub-critical or in trans-critical mode depending upon the ambient temperature.

Performance of CO_2 booster configuration in a cold climate is found to be better compared to conventional HFC based direct expansion (DX) systems (Fricke et al., 2016). Prominent modifications of CO_2 booster configuration in supermarket include parallel compressor dedicated mechanical sub-cooler and multi-ejector. Gullo et al. (2016a) suggested a combination of parallel compression with dedicated mechanical sub-cooler as efficient alternative in terms of energy consumption, especially for warm climates. Hafner et al. (2016) reported reduction in energy consumption up to 30% for multi ejector configuration over the solution employing parallel compression based on the field measurements of a refrigeration plant located in Spiazzo (North of Italy). Schonenberger (2016) compared and reported a maximum of 20% energy savings for CO_2 refrigeration plant assisted with multi ejector unit compared to that using parallel compression for two Swiss stores. Tsamos et al. (2017) showed that the solution with parallel compression consumes 5% less electricity than the basic CO_2 booster configuration in the city of Athens.

2.5.2 CO_2 indirect and cascade system

In the indirect and cascade solutions, CO_2 operates at sub-critical running modes irrespective of outdoor temperature, removing the technical and economic challenges of high pressure operation and have advantages over "all CO_2" system, especially in warm climatic conditions (Sharma et al., 2014a; Llopis et al., 2015b). The adoption of indirect configuration allows simplified maintenance compared to DX system (Wang 2010), besides reduction in refrigerant charge down to 95% (Hesse 1996). However, two main disadvantages associated with the secondary loop systems are the need for a pump and the presence of an additional heat exchanger. Adoption of CO_2 as a secondary working fluid offers reduction in pump work as well as pipe size and improvement heat transfer characteristics (Inlow 1996a). Inlow and Groll (1996b) reported that CO_2 as secondary fluid can counterbalance the drop in COP caused by the additional heat transfer level and the pumping power for secondary loop systems.

2.5.3 Integrated CO_2 system

Integrated "all-in-one" configurations are reported to satisfy mostly or full air-conditioning (A/C) and heating demands for a supermarket rather than fulfilling only the refrigeration loads (Karampour and Sawalha, 2015). Such configurations are already installed in more than ten supermarkets in Sweden, which firstly, eliminates the need of alternative working fluid and secondly, reduces both direct and indirect emissions (Sawalha et al., 2017). Such system also gainfully recovers the heat, otherwise liberated into atmosphere in the gas cooler. Further, the "all in one" configurations are claimed to be capable of service without interruption and provide better controls among the various components installed in a supermarket. Reinholdt and Madsen (2010) concluded that integration of heat recovery is an efficient modification for booster configurations which can lead up to 25% improvement in COP. Cecchinato et al. (2012) claimed that CO_2 booster system integrated with heating and air-conditioning result into comparable or even less energy consumption than the reference R404A system. Nordtvedt and Hafner (2012) reported work on developing CO_2 air conditioning systems for large scale plants like supermarket application. Funder-Kriestensen et al. (2013) presented a case study of a Danish supermarket employing CO_2 trans-critical booster system with heat recovery. They reported 20% saving in running cost compared with the gas heating system. Karampour and Sawalha (2014) reported comparable seasonal performance factor for CO_2 booster system integrated with heat recovery with conventional heat pumps. Sharma et al., (2014b) studied integrated waste heat recovery in the various CO_2 trans-critical supermarket refrigeration systems. They conducted bin analysis for US climatic zones and concluded that heat recovery could nearly satisfy the desiccant regeneration and water heating demands. Giretto (2016) investigated integrated CO_2 system which provides direct space heating and cooling for a small supermarket in Italy and found 15–20% energy saving and considerable cost reduction. Karampour and Sawalha (2017) presented comprehensive analysis on integrated CO_2 supermarket refrigeration solutions. They concluded that an integrated CO_2 booster configuration can satisfy whole refrigeration and A/C demands efficiently, for cold and mild climates. Polzot et al. (2017) reported a maximum of 6.5% savings in annual electricity consumption for a CO_2 booster system integrated with heat recovery for space heating in North Italian climate compared to baseline system comprising of R134a/CO2 cascade system for refrigeration and R410A heat pump. Gullo and Hafner (2017) showed that integrated multi-ejector based R744 arrangements depicted 3.1% to 22% in AC mode and 57.8% to 100.1% in heating mode greater total COP than separated HFC technologies over the evaluated range of ambient temperatures.

2.6 Gap areas in existing research

From literature survey, it is concluded that the following significant issues need to be addressed which will also be focus of this research work.

a) Published literature on experimental studies on CO_2 trans-critical refrigeration system (Table 2.1) is very limited when ambient temperature crosses 40°C. Further, only a limited number of studies have focused on effect of gas cooler face velocity on COP, as well as sensitivity of the same.

b) Modelling of CO_2 refrigeration system as well as predictions of optimum operating conditions are limited to physics-based models.

c) Simultaneous optimization of performance of CO_2 trans-critical refrigeration system in terms of gas cooler face velocity and high side pressure is absent.

d) In context of refrigeration system for supermarket application, the improvement in performance of CO_2 booster cycle due to adoption of various modifications like parallel compression, flooded evaporator, multi-jet ejector and work recovery expander are evaluated mainly based on energetic perspective, economic analysis is scarce.

e) Reported literature on supermarket refrigeration is limited to the warm climatic conditions where the frequency of ambient temperature over shooting 25°C during a year is not significant, and is therefore cannot be reflected for most of the cities in India in summer or for Gulf regions.

Table 2.1 Summarized studies reported on CO_2 trans-critical system with IHX

Author	T_e (°C)	T_{amb} (°C)	p_{gc} (MPa)	$V_{gc,fv}$ ($m \cdot s^{-1}$)
Boewe et al., (2001)	26.7$	32.2 – 43.3	8 – 14	0.378*
Liu et al., (2005)	27$	27.5 – 45	6 – 12	2.5 – 6
Cho et al., (2007)	27$	35	7.5 – 10.5	–
Cabello et al, (2008)	-18.1 – -0.9	31.1 – 40.4	7.6 – 10.3	0 – 4*
Aprea et al., (2008)	5	25 – 40	5.5 – 12	–
Rigola et al., (2010)	-5	30 – 50#	8.5 – 12	0 – 2**
Tao et al., (2010a)	19 – 29$	32 – 37#	9	0.4 – 2.4
Sanchez et al., (2010)	-17 – 0	23 – 29#	7.4 – 10.5	0 – 4*
Torella et al., (2011)	-15 – -5	31 – 34#	7.4 – 10.5	0 – 4*
Aprea et al., (2013)	0 – 5	25 – 40	8 – 10.1	–
Sanchez et al., (2014)	5 – 15$	25 – 35#	7.4 – 10.5	1*
Kim et al., (2017)	27$	29 – 35#	7.5 – 10	0.140**

f) *$Evaporator air/water inlet, #gas cooler air/water inlet, **Mass flow rate ($kg \cdot s^{-1}$), *Volume flow rate ($m^3 \cdot h^{-1}$)*