Proceedings of the International Conference on Industrial & Mechanical Engineering and Operations Management
Dhaka, Bangladesh, December 26-27, 2020

A Review of Trans-critical CO_2 refrigeration cycle

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Abstract

Carbon dioxide (CO_2) is one of the natural refrigerants that can be used as working fluid in various refrigeration applications along with ammonia and hydrocarbons due to its eco-friendliness, higher volumetric capacity and good heat transfer properties, etc. In order to overcome inherently low efficiency in high temperature conditions and high operating pressure, particularly in transcritical cycles. A comprehensive overview of the advancements in modified technologies to overcome CO_2 refrigeration disadvantages is presented and the recent progress towards improving energy efficiency is summarized. Initially, the basic principles of the CO_2 cooling cycle and the important performance characteristics are discussed. A detailed study on the various modified technologies, as well as their basic operational, technical and performance characteristics are provided, which are followed by a summary of previous research.

Keywords
Natural refrigerant; CO_2; Energy efficiency; Transcritical cycle; Ejector, Global warming, Applications.

1. Introduction
As a safety part, carbon dioxide and green natural refrigerant have numerous good advantages, such as no toxicity, no inflammation, high volumetric ability, better compatibility with common lubricant and lower prices [1], [2]. The implementation of the Paris Agreement sparked a public speech focusing on atmospheric concentrations of greenhouse gases [3]. Almost three years later, the Intergovernmental Panel on Climate Change (IPCC) released an alarming report on the dangers of a rise in global warming above 1.5°C [4]. The study highlights the catastrophic effects of a rise of 2°C compared to an increase of 1.5°C. The deceptively small variation in between two temperature increases obscures the expected 10 cm rise in sea levels, the extreme reduction of Arctic sea ice and the depletion of 99 per cent of Earth's coral reefs. With carbon emissions rising, the IPCC warns that "rapid and far-reaching" action is needed to hold global warming at 1.5°C. Among the many global warming mitigation techniques, refrigerant management is inserted first in Hawken's book [5], in which a total of 80 techniques are identified and evaluated. Many of the hydrofluorocarbons (HFCs) used in the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry have a high global warming potential (GWP), for example, R134a has a GWP of 1340 over a 100-year period. In this sense, natural refrigerants have given rise to growing interest in the HVAC&R sector as the ultimate solution rather than finding new chemicals such as HFCs. Among many natural refrigerants, such as hydrocarbons, ammonia, air, water and CO_2. CO_2 is the only non-flammable, non-toxic, 0 ODP, 1 GWP refrigerant graded as A1 [6]. No other refrigerants may follow these characteristics at the same time. On this basis, CO2 can be the final solution in refrigeration systems without any environmental concern, which is a major problem for today's society. CO_2 is also cheap and exhibits higher latent heat, real heat, density and thermal conductivity and lower viscosity compared to HFCs. From the point of view of historical progress, CO_2 was first used as a refrigerant in the British patent granted to Alexander Twining in 1850[7]. Lowe designed the first CO_2 machine to generate artificial ice in the late 1860s [8]. Since then, CO_2 has still been on the market before synthetic refrigerants CFCs emerged vigorously in the 1930s with enhanced performance and lower prices [9]. In the late 1980s, ozone-depleting refrigerants, including CFCs, started to phase out. In 1990, the Norwegian professor Gustav Lorentzen issued a patent application for a transcritical CO_2 device for automotive air conditioning, which suggested the regeneration of CO_2 [10]. In 1993, Lorentzen and Pettersen published the experimental results of the first CO_2 prototype system [2]. The prototype system was further developed by Pettersen and showed comparable efficiency to that of R12 [11]. In 1999, the first transcritical CO_2 residential air conditioning was simulated and compared to

the R22 system in the German patent [12]. Since the beginning of the 21st century, researchers have never stopped researching transcritical CO_2 cycles in different applications.

Several review articles on the transcritical CO_2 cycle have been summarized and collected to the best of the authors' knowledge. Kim et al.[13] presented a study of transcritical CO_2 cycle technology in various applications for refrigeration, air-conditioning and heat pump applications, based on literature, from 1994 to 2004, in which fundamental issues of process and system design were figured out. In addition, four updated cycles, including internal heat exchange cycle, work recovery extension, two-stage cycle and flash gas bypass, have been implemented to increase system performance. Ma et al. [14] provided a detailed description of the transcritical CO_2 heat pump and cooling systems, including the properties of supercritical CO_2 and PAG lubricants, optimum friction, novel expander-based cycles. Pradeep [15] provided a historical view of the fundamentals and application of CO_2 in low-temperature cooling systems, especially in the food industry, suggesting that further fundamental research on unraveling the physics of CO_2 and CO_2 -oil condensation heat transfer extending to mini and micro heat exchangers is required. Maina [16] analyzed numerous CO_2 applications and situations in South Africa, and the authors assume CO_2 is the ultimate potential refrigerant. Qi [17] reviewed developments in the electric vehicle's CO_2 air-conditioning and heat pump system. Paride et al. [18] carried out an in-depth study of the CO_2 cooling plants for food retail applications from an energy, environmental and economic perspective. Brian et al. [19] also provided an overview of the transcritical carbon dioxide heat pump systems with numerical analysis, device elements, compression and expansion process configurations, and modifications. Luisa et al. [20] also investigated the heat transfer properties of supercritical CO_2, and Simarpreet et al. [21] reviewed various CO_2 transcritical job recovery expanders.

After all, there is limited systematic and detailed explanation of novel improvement technologies in core CO_2 refrigeration cycles which at the same time includes operating fundamentals, technical features and performance. In recent years, several new technologies have been creatively developed and significantly improved, although those innovations have not been summarized and evaluated in the reviews above. Empowered by this point, the present study aims to present a comprehensive state-of-the-art analysis on an update of information on various transcritical CO_2 cycle improvement technologies. This review begins with a brief summary of the underlying concepts and essential transcritical CO_2 cycle results. It is then accompanied by thorough discussions focused on its structure, functionality and results on various enhancement technologies. Ultimately we address our brief viewpoints on this field's prospects.

2. The fundamental Transcritical CO_2 refrigeration cycle

The Crucial CO_2 point is at 30.85°C and 73.53 bar. So many applications for air conditioning and refrigeration span the critical point in that the temperature for heat absorption is below the critical temperature and the temperature for heat rejection is above that. This suggests a transcritical period in which the evaporator acts as a common two-phase vapor liquid device but replaces the condenser with a supercritical heat-rejection system called a gas cooler [22]. Figure 1 shows the fundamental transcritical CO_2 cycle under which the 1g P-h diagram was drawn based on the following assumptions: no drop under pressure, 32°C gas cooler outlet temperature, 5°C evaporation temperature, 0 K superheat, isentropical compression phase and isenthalpic expansion. It can be shown that the high-side pressure is about 10 MPa, which is typically 5-6 times higher than traditional refrigerants, resulting in high costs and a hazard to reliability and safety of the systems [23].

Theoretically the transcritical CO_2 cycle is less effective under the same conditions compared to a traditional vapor compression cycle. Take for example the subcritical period R134a, Fig. 2 The contrast of the thermodynamic cycles for R134a and CO_2 is shown in the temperature-entropy diagram, in which the exergy losses in terms of areas are apparent [13]. At Figure 2, two processes are responsible for additional thermodynamic losses of the transcritical CO_2 cycle, assuming equivalent evaporating temperature, equivalent gas cooler outlet temperature with condensing temperature; i. e. throttling loss and loss of refuse gas. During the gas cooling process, the greater heat-rejection loss results from the much higher average CO2 temperature. Due to the greater pressure difference before and after CO_2 expansion unit, the throttling loss has more to do with the refrigerant properties, with temperatures calculated, larger entropy increase occurs during the throttling phase.

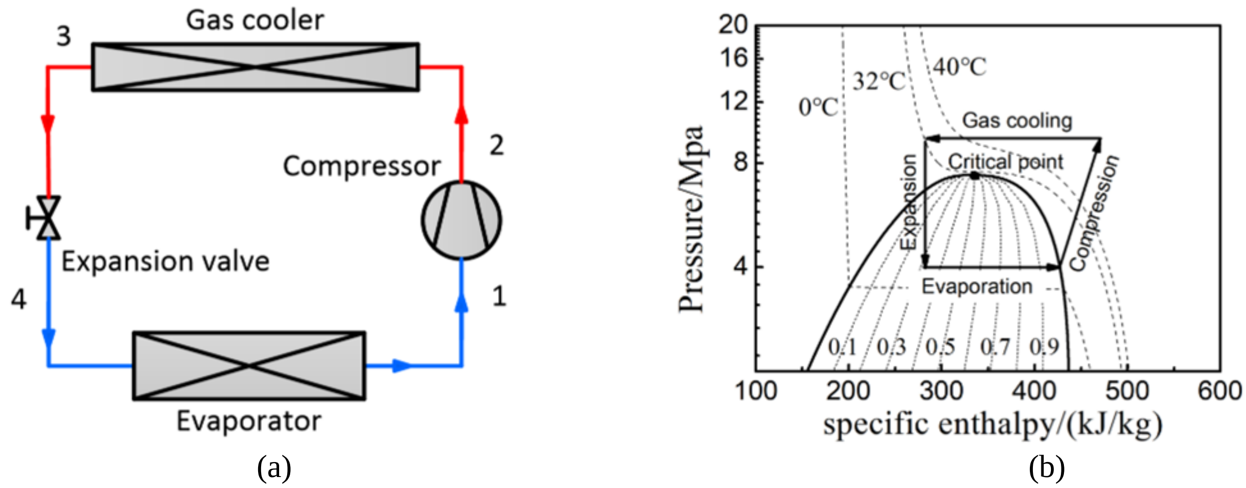


Figure 1. Fundamental transcritical CO_2 cycle (a), and 1g P–h diagram (b) [23].

However, the large pressure difference in transcritical CO_2 cycle generates a lower compression ratio, being around three in the cooling application while the conventional cycles operate at a ratio of eight[24]. The lower compression ratio tends to make a compressor more powerful. It should be noted that due to better CO_2 heat transfer characteristics, the temperature between the gas cooler outlet temperature and the temperature of the heat sink can be much lower than that between the condensing temperature and the temperature of the heat sink [25]. The transcritical CO_2 cycle, while given these advantages, exhibits poor efficiency performance at high ambient temperatures. Martin [26] and Yin [27] demonstrated that the COP (coefficient of performance) of a CO_2 air conditioning system is lower in the 10% usage conditions at high ambient temperature (above 30 ° C). Brown et al., evaluated the performance merits of CO_2 and R134a mobile air conditioning systems, their results from semi-theoretical models show that the COP of CO_2 was lower by 21% at 32.2°C and by 34% at 48.9°C. The COP disparity was even greater at high speeds and ambient temperatures [28]. Moreover, the CO_2 mobile air conditioning system achieved comparable cooling power compared to a traditional R134a mobile air conditioning system but had COP reductions of 26 per cent and 10 per cent at 27°C and 45°C outdoor conditions respectively[29].

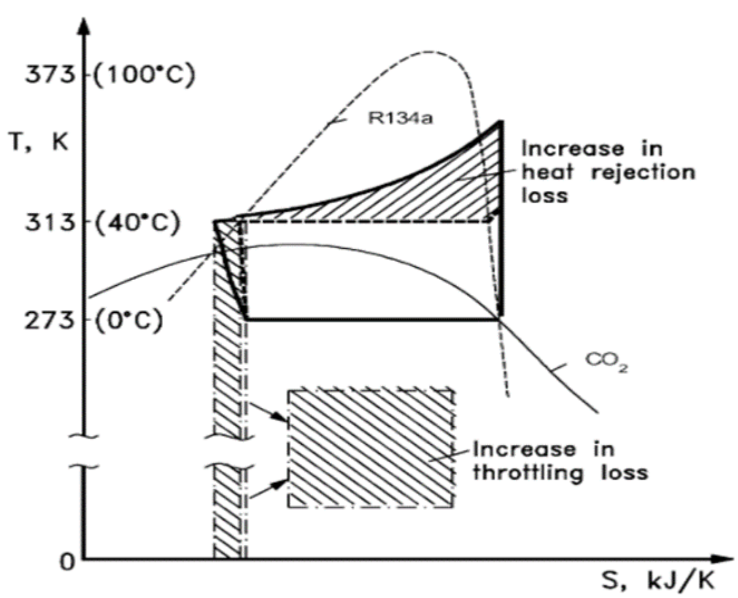


Figure 2. Comparison of thermodynamic cycles for R-134a and CO_2 in temperature–entropy diagrams [13].

The foregoing analysis of the literature illustrates the key challenges of the transcritical CO_2 refrigeration cycle, i.e. high operating pressure and poor energy consumption under high ambient conditions. Hence researchers have dedicated themselves to solving them through different technologies.

3. Modified Transcritical CO_2 cycle

Comparing the exergy losses between the subcritical and the transcritical cycles shows how penalized a transcritical CO_2 cycle is. Although the theoretical and experimental findings suggested that the efficiency of the CO_2 fundamental cycle system will be lower, a large number of improved technologies are promising to make the actual efficiency of the transcritical CO_2 cycle equal to or even greater than traditional subcritical cycles. Figure. 3 Portrays the outline of all the technologies for development that will be built in the following subsections [23].

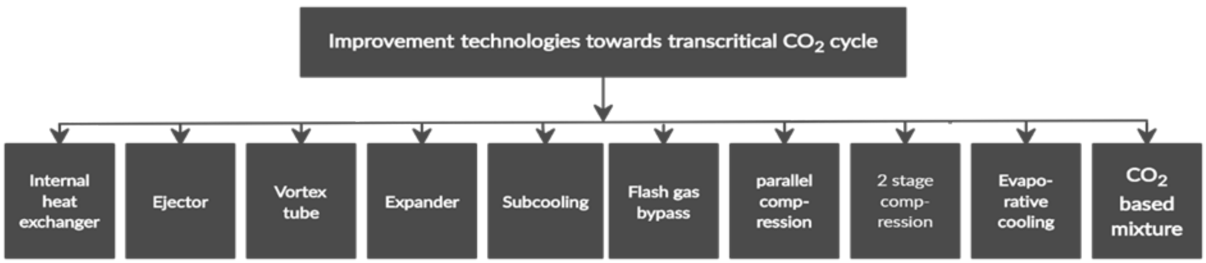


Figure 3. Outline of the review for improvement technologies towards transcritical CO_2 cycle.

3.1. Internal heat exchanger cycle

The traditional refrigeration system P-h diagram with classical IHX (interior heat exchanger) is shown in Fig. 4 It was shown that the IHX can both boost or reduce system efficiency due to the trade-off between increased capacity and discharge temperature depending on the working fluid and operating conditions[30]–[32]. Aprea et al. [33] proposed a simplified criterion for determining the potential thermodynamic benefit of implementing an IHX. The criterion has been described as the following inequality based on the state point in Figure 4:

$$CpT_1 > (h1\text{-}h4) \qquad (1)$$

Where Cp (kJ/kg K) is the constant pressure specific heat in the suction condition. The use of an IHX turns out to be beneficial when checking this inequality [34]. In the case of CO_2, the device configuration and 1gP-h diagram of the transcritical CO_2 cycle with and without IHX is shown in Figure 5, in which the low-pressure side of the IHX process is similar to that of the traditional refrigeration cycle. Theoretically, the criterion discussed in Ref. [33] is applicable to CO_2. Thus, one can easily determine that the use of IHX in summer conditions with high ambient temperatures is useful for transcritical CO_2 use, while the internal heat exchanger does not enhance the efficiency of the subcritical CO_2 cycle[33], [35].

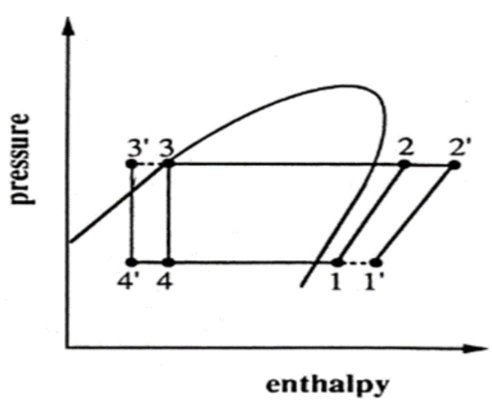
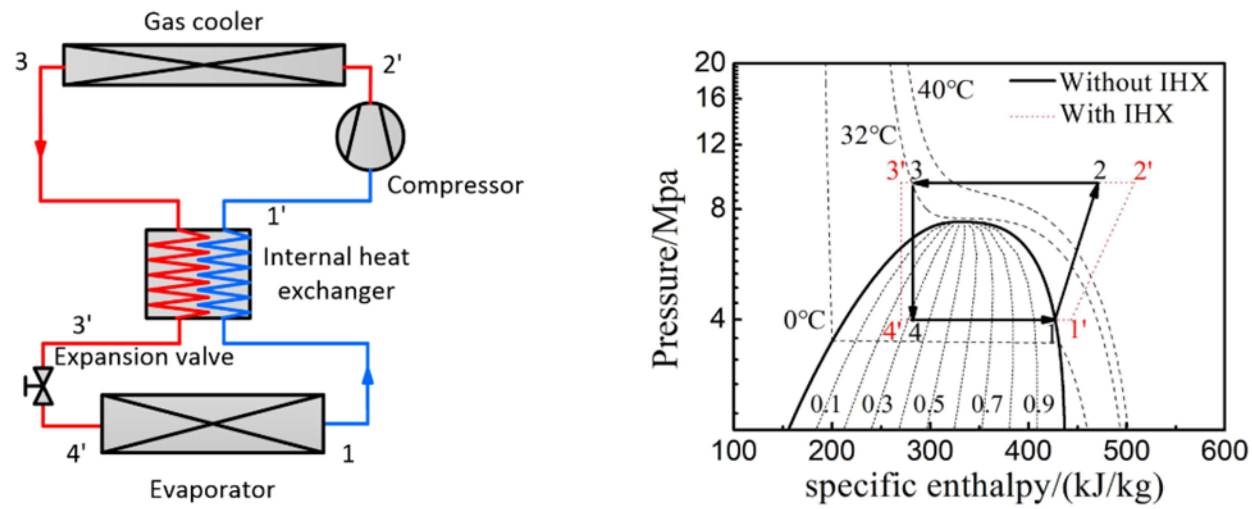


Figure 4. P-h diagram of the conventional refrigeration cycle with (1-1’-2’-3-3’-4’-1) and without IHX (1-2-3-4-1) [33].

In fact, the advantages of using IHX for the transcritical CO_2 cycle are very significant, and this has been validated in various applications from different aspects by various research. Both power and COP could be increased by up to 25 per cent for mobile air conditioning. At high air temperatures the gas cooler detects greater influence. In detail, IHX counterflow is better than parallel if the temperature of the compressor is below its limit. Larger IHX is advantageous for increasing power and COP and reducing the optimum pressure where the full COP value is obtained, but the limited size exists to prevent the temperature of the compressor discharge from reaching its design limit[36], [37].



(a) (b)

Figure 5. A sketch of the CO_2 refrigeration cycle with IHX (a), and 1gP–h diagram of transcritical CO_2 cycle with and without IHX (b) [23].

For commercial refrigeration, Joaquim et al. [38] examined the effect of IHX through computational simulation and experimental validation on a single stage CO_2 transcritical cycle, finding that the addition of an IHX significantly increases the COP. The COP of a cycle with an IHX of 2 m duration increases about 23% and 35 % respectively under ambient temperature of 35 and 43. When the air temperature rises COP increases further. In household applications, a 10 % CO_2 increase was obtained in a transcritical CO_2 cycle operating as a classic "split system" to cool air [39].

Purohit et al. [40] examined the effects of IHX on device efficiency from an energetic and exergetic perspective for a CO_2 transcritical cycle for chiller application. IHX has the least contribution to exergy loss in the scheme, and the highest COP enhancement and IHX exergetic performance is 5.71% and 5.05% respectively. The IHX cycle, however, also results in a maximum temperature rise of 24 ° C in the compressor discharge.

With regard to the efficacy of IHX, as there can be phase changes in IHX and the specific heat changes dramatically in the region close to the critical pressure, the standard definition used to define the efficacy of heat exchange is inadequate, a realistic term for IHX was derived from Chen et al. [41] based on the difference in enthalpy, an IHX with high efficacy is an IHX with high efficacy. In another analysis, however, system COP was found to be inversely proportional to the thermal efficiency of IHX [42]. Normally, the classical position of an IHX in the above studies is all at the exit of the gas cooler, however different positions even different results can have for IHX configurations.

As shown in the figure. 6, Sanchez et al. [43] compared the energetic behavior of a transcritical CO_2 refrigerating plant operating with several IHX configurations: exit gas cooler (classical position), exit liquid receiver, and simultaneously in both positions. While a general improvement in COP and cooling capability was observed regardless of the location of the IHX, with two IHX at the same time a maximum increase of 13% was obtained at COP.

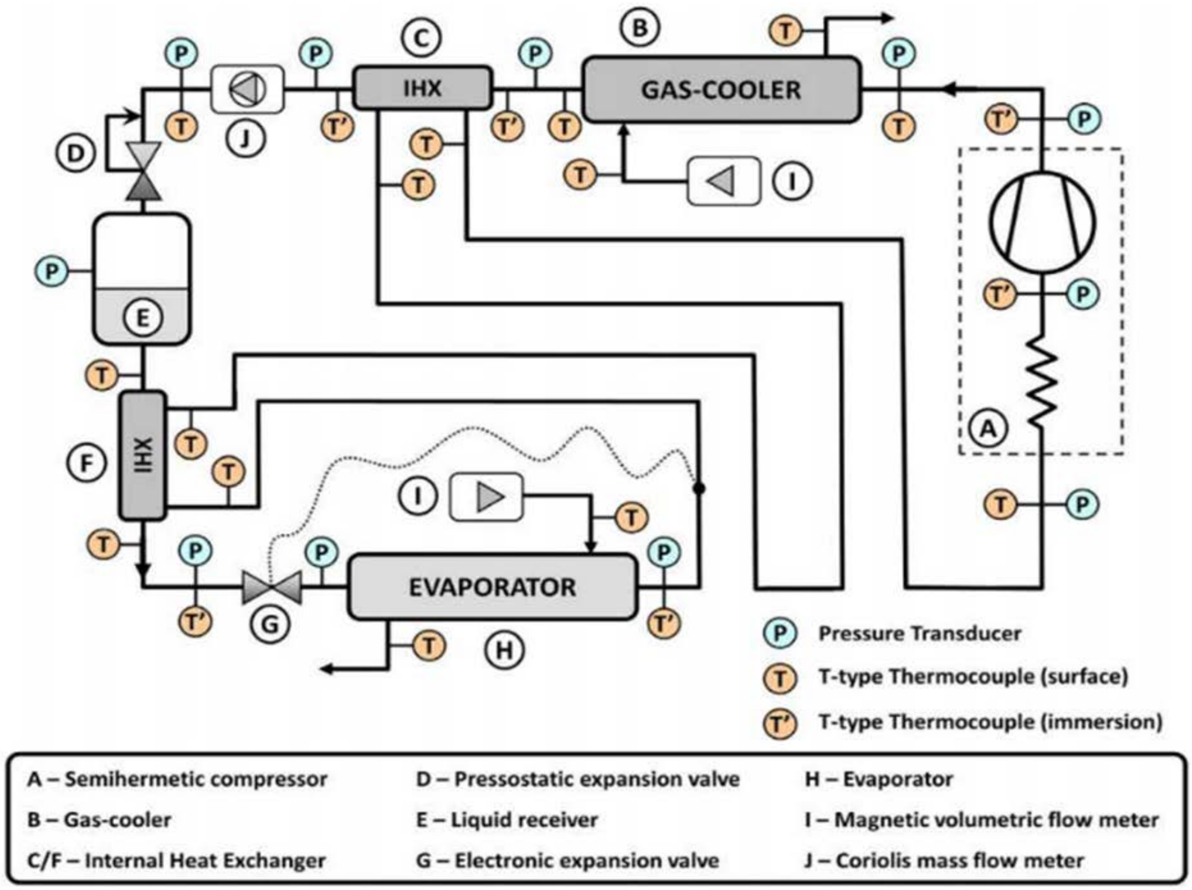
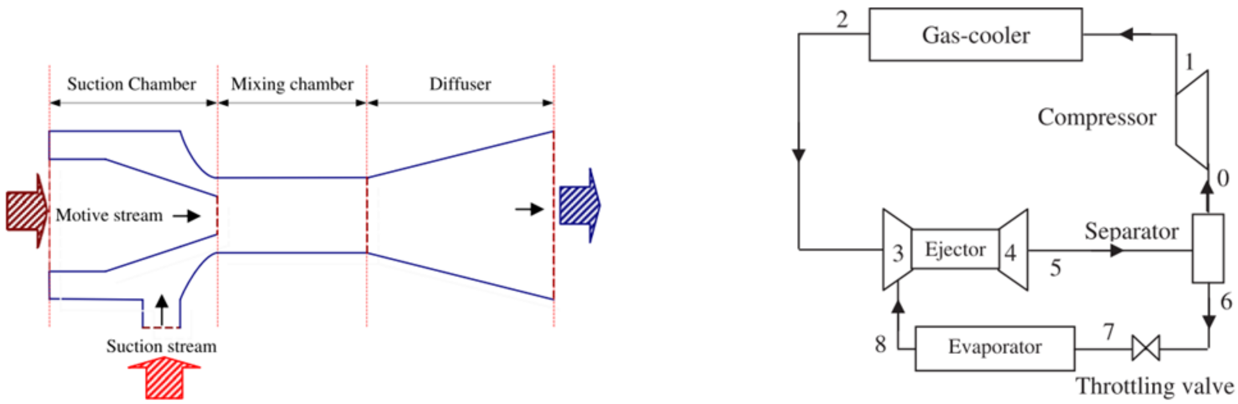


Figure 6. Experimental framework model to investigate the effect of IHX positions on system performance [43].

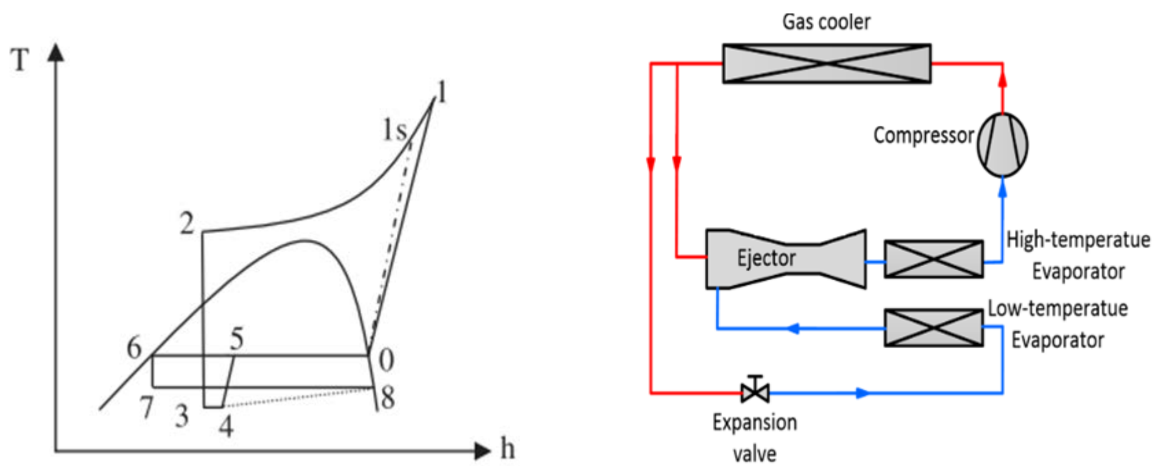
3.2. Ejector expansion cycle

The transcritical carbon dioxide cooling system with ejector consists of a compressor, a gas cooler, an ejector, a separator, a throttling valve and an evaporator. The structural description of the ejector is shown in Figure 7(a). The ejector is composed of a suction chamber, a mixing chamber and a diffuser. The motive flow from the gas cooler increases in the motive nozzle, and the friction is minimized. The enthalpy of the motive stream drops from h_2 to h_3 and the velocity increases to Umo. The suction stream also extends in the suction nozzle, and the Usu velocity increases. In the mixing chamber, the two streams combine and form one current. The mixed stream further increases its pressure in the diffuser by converting kinetic energy to internal energy. The purpose of the ejector is to recover some kinetic energy from the throttling phase and increase the inlet pressure of the compressor. The schematic view of the ejector cycle system is shown in Figure 7(b) and the corresponding T_{eh} (temperature-enthalpy) diagram is shown in Figure 8. Compared to the ejector cycle, the throttling cycle is without ejector, as shown in Figure 8, for one unit of refrigerant saturated mass refrigerant (0), detach from the gas liquid separator and join the compressor to be compressed. High temperature and pressure refrigerant (1) from the compressor is cooled to point 2 by the gas cooler under the same pressure. The supercritical refrigerant reaches the nozzle and expands into a two-phase jet (3) by reducing the pressure and the speed of the extended portion of the ejector. At the same time, the m unit of the refrigerant mass (8) of the evaporator is also energized and extended in the nozzle. The additional pressure of the two streams is the same. Motive and suction streams are combined under continuous pressure in the mixing area of the ejector. The unit of mixed refrigerant mass (4) then reaches the diffuse portion of the ejector where its kinetic energy is converted into a pressure boost (5). The efficiency (x) of the mixed refrigerant at the outlet of the ejector is assumed to be $1/(1+\mu)$ to maintain constant mass flow of the compressor. The unit of saturated liquid refrigerant mass (6) from the separator is throttled into a low-pressure two-phase mixture (7) and evaporated into the evaporator. And one unit of saturated refrigerant mass (0) of the separator reaches the compressor. Compared to the ejector cycle, one unit of refrigerant mass the throttling valve cycle continuously flows through all its components [44].



(a) (b)

Figure 7. (a)The structural architecture of the ejector; (b) Sketch of the Transcritical CO_2 cycle with the ejector [44].



(a) (b)

Figure 8. (a)T- h diagram of the CO_2 transcritical cycle with ejector [44] ;(b) New ejector cycle[45]

The mechanics of the ejector, the operating characteristics, the design effects of the ejector, the simulation and the basic control issues have been discussed. Among them, the most extensive analysis of more than 300 papers was given by Besagni et al. in 2016[46], and an overview of the ejector refrigeration systems that have been built over the last nearly 30 years has been categorized and shown as Figure 9, in which the transcritical CO_2 ejector cooling system was partly addressed in the 8th module.

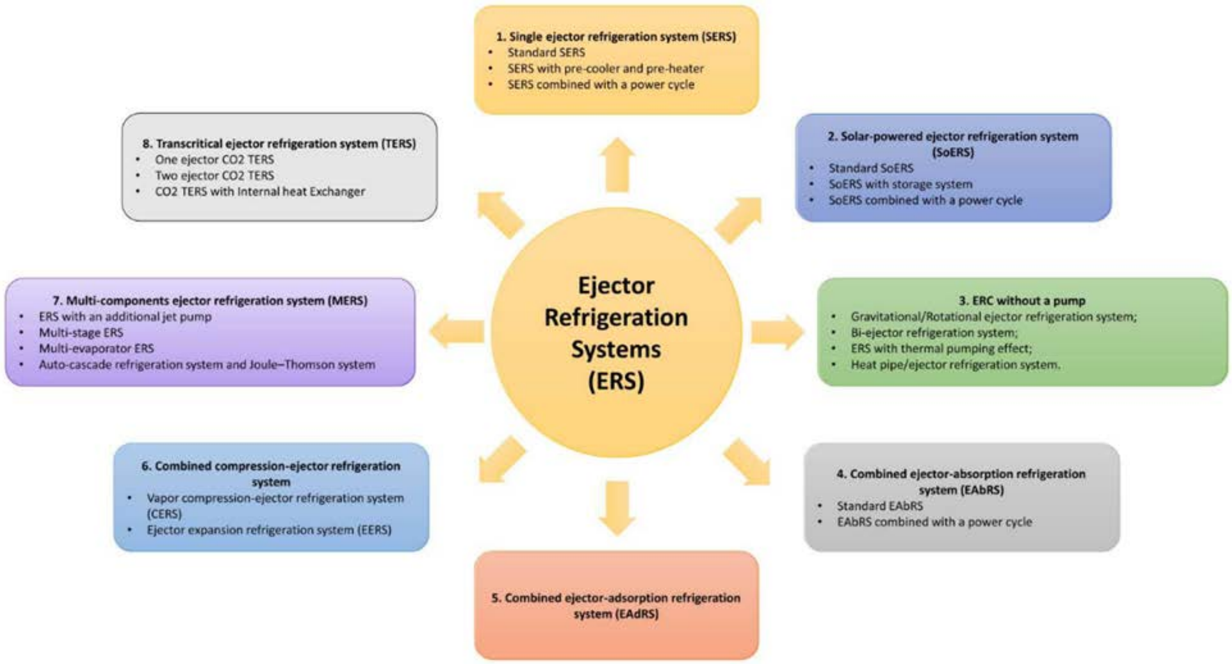
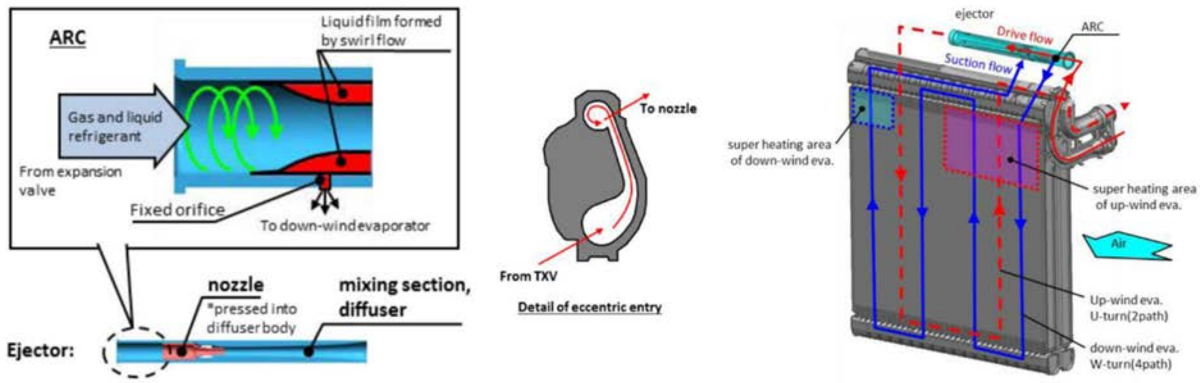
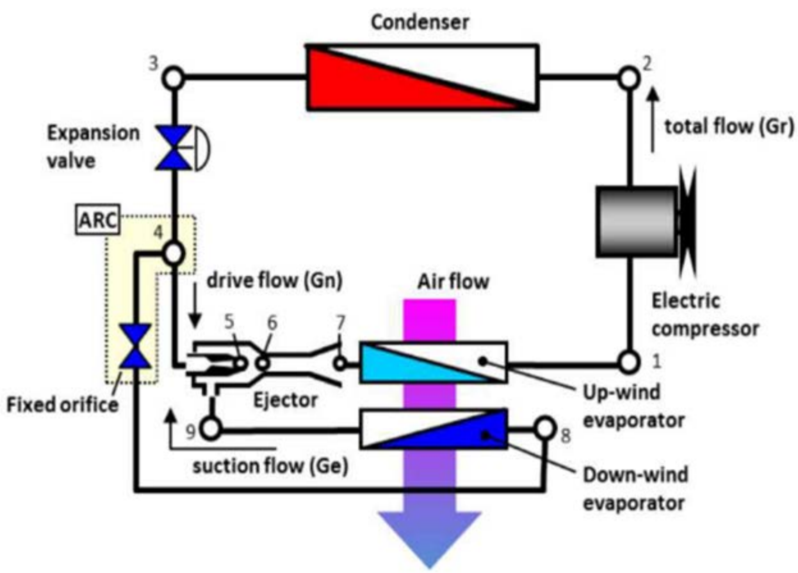


Figure 9. Overview of ejector refrigeration systems [46].

In addition, Lawrence and Elbel [47] argued that this ejector cycle has more substantial functional advantages over the regular ejector cycle for those working fluids that give less recovery capacity, such as R134a. Based on this method, DENSO [48] proposed its new generation ejector for the mobile air conditioning system. Figure 10 (a) shows the working theory and its application with upwind and downwind evaporators. It's in Figure 10 (a), The Active Flow Ratio Control (ARC) system is used to achieve an optimum refrigerant flow ratio through the ejector by separating the two-phase refrigerant flow into the gas and liquid with centrifugal force generated by the eccentric input. Then the liquid-rich refrigerant flows through the fixed orifice into the downwind evaporator, the remaining gas-liquid two-phase refrigerant flows to the side of the nozzle and then reaches the upwind evaporator. Test results show that the latest generation's ejector performance is 3 times higher than the previous ejectors. At the same time, in Fig, the device with this new ejector. Figure 10(b), has achieved approximately 20% savings in power relative to a traditional air conditioning system without an ejector. It also achieved 10% savings in power compared to the previous ejector system [48].



(a)



(b)

Figure10. (a) New generation swirl flow ejector and its integration with up-wind and down-wind evaporator by DENSO, (b) the system with new generation ejector [48].

In addition to investigating the efficiency of the system, visualization is often an efficient tool for better understanding internal fluid flow phenomena and enhancing the geometry of the CO_2 ejector and model validation. Recently, with the advancement of high-pressure visualization technology, complex flow dynamics and phase shift processes within the CO_2 ejector have been found in several studies using direct photography or internal physical variable measurement methods. Zhu et al. [78] represented the supersonic two-phase expansion flow after the nozzle in the suction chamber and the mixing chamber of the transcritical CO_2 ejector using a direct photography process. The results indicated that the liquid fraction in the active flow in the suction chamber increases with an increase in the active flow and suction pressures. The mixing of the active and suction streams in the mixing chamber is very fast. The expansion angle of the active flow at the exit of the nozzle decreases with rising suction flow pressures [49]. The input ratio is inversely proportional to the expansion angle. As regards the CO_2 phase change phenomenon in the nozzle, Li et al.[79] investigated the transformation of the phase change position in the primary converging-diverging nozzle using the direct photography method as shown in Figure 11. Visualization images showed that the phase change might start after or before the throat, depending on the operating conditions. The direction of the phase shift shifted upwards when the inlet pressure and temperature of the primary flow decreased simultaneously [50].

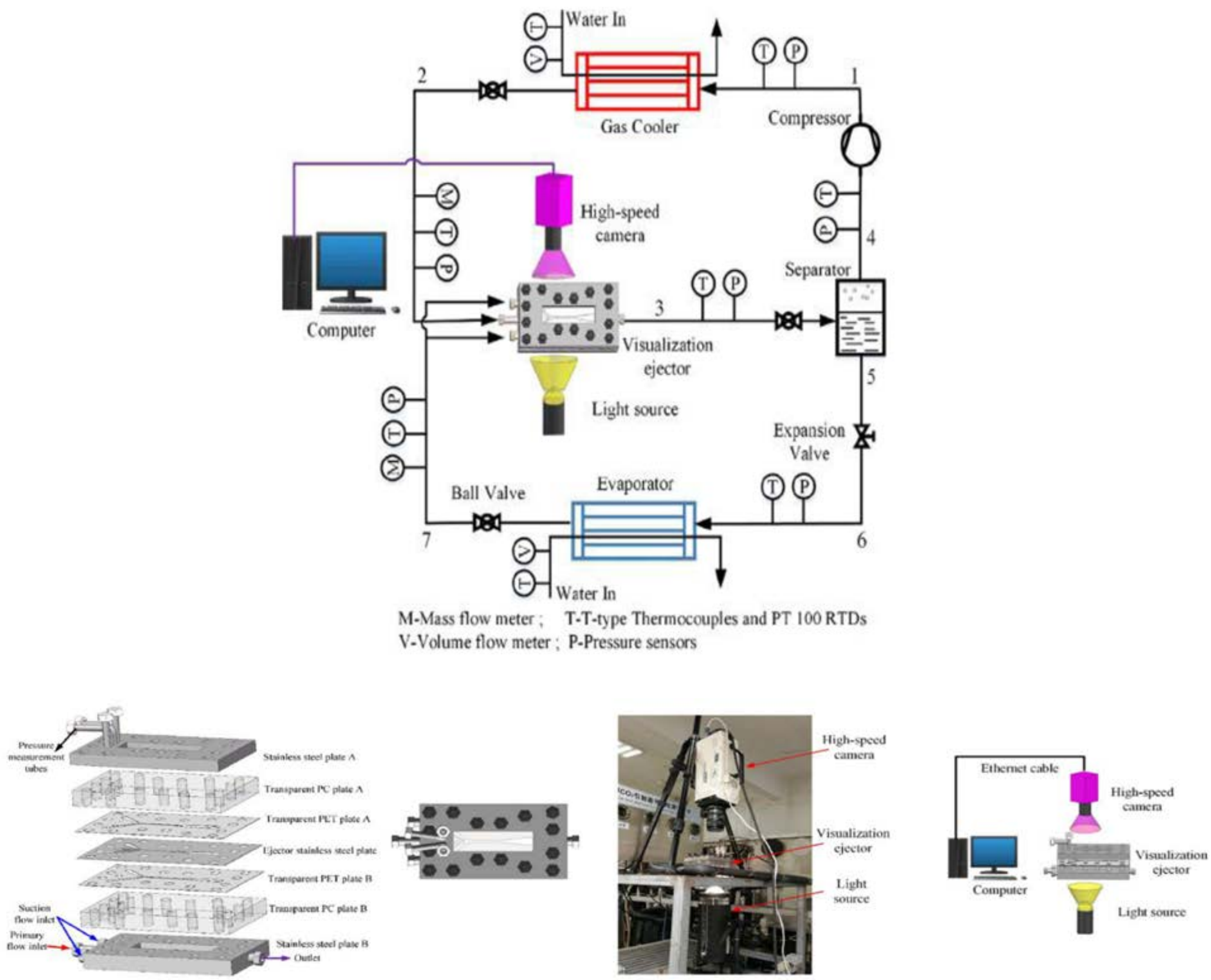


Figure 11. Schematic diagram of the visualization ejector and photography test platform [50].

After all, there is still no valid theory in the literature on supersonic multi-phase flow with phase transition, resulting in the fact that the analysis of the flow theory in the supersonic ejector is not definitive, which further leads to a significant error between the estimation of the CFD model and the experiments. What's more, the current shock wave theory is based on single phase flow, the shock wave pattern inside the expansion section of the nozzle, and the initial position of the choked flow for the actual fluid, which may be clarified by more simulation methods in the future, has not been well understood.

3.3. Vortex tube

The vortex tube will split the inlet gas into two consecutive streams with differing temperatures, as shown in Figure 12, one of the segregated streams is at a higher altitude, while another stream is at a lower temperature than that of the added fluid. Detailed knowledge on the vortex tunnel, including geometry design, experiments, simulation, and so on, has been reviewed by Thakare et al. [51]. It was revealed by Ranque [52] in 1933 and then further examined by Hilsch in 1947[53]. The vortex tube is also recognized as the Ranque – Hilsch vortex tunnel. Based on the Ranque – Hilsch impact, smart folks can easily conclude that they are capable of operating as an expansion system in the refrigeration cycle. In 1997, Keller [54] suggested a steam-compression refrigeration cycle with a vortex tube, as shown in Figure 13. Its fundamental theory is as follows. The compressed high-temperature refrigerant is cooled in the gas cooler and flows into the intermediate cooler, where the coolant is further cooled by the steam stream from the separator. At the same time, the steam stream is heated in the intermediate cooler and then enters the vortex tunnel, because of the Ranque – Hilsch effect, the steam stream is divided into two streams, i.e. cold gas and hot gas. The hot fraction must be cooled in a furnace and then combined with the cold gas. The liquid is then combined with the evaporator vapor and the blended three streams flow into the compressor to be compressed. Keller theoretically evaluated the cycle efficiency using R22, R134a and CO_2 as working fluids. The results showed that the COP was increased by 5 %, 10 % and 15%, respectively, compared to the traditional simple steam compression cycle.

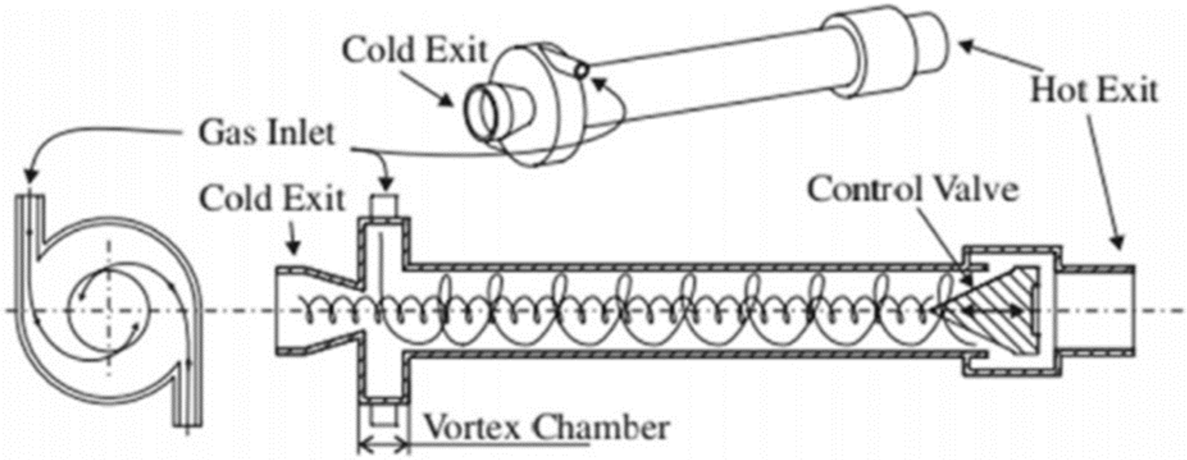


Figure 12. Schematic of a vortex tube [55].

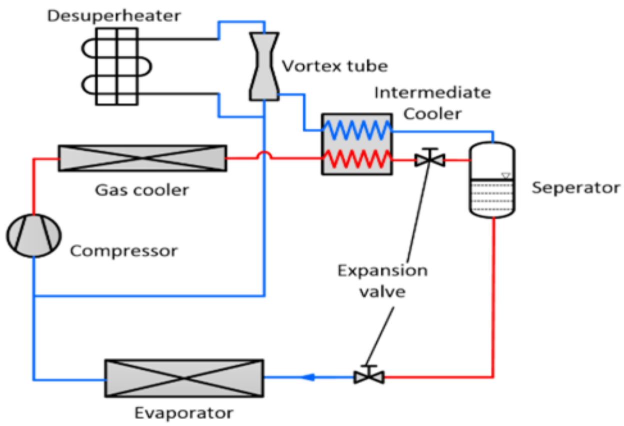
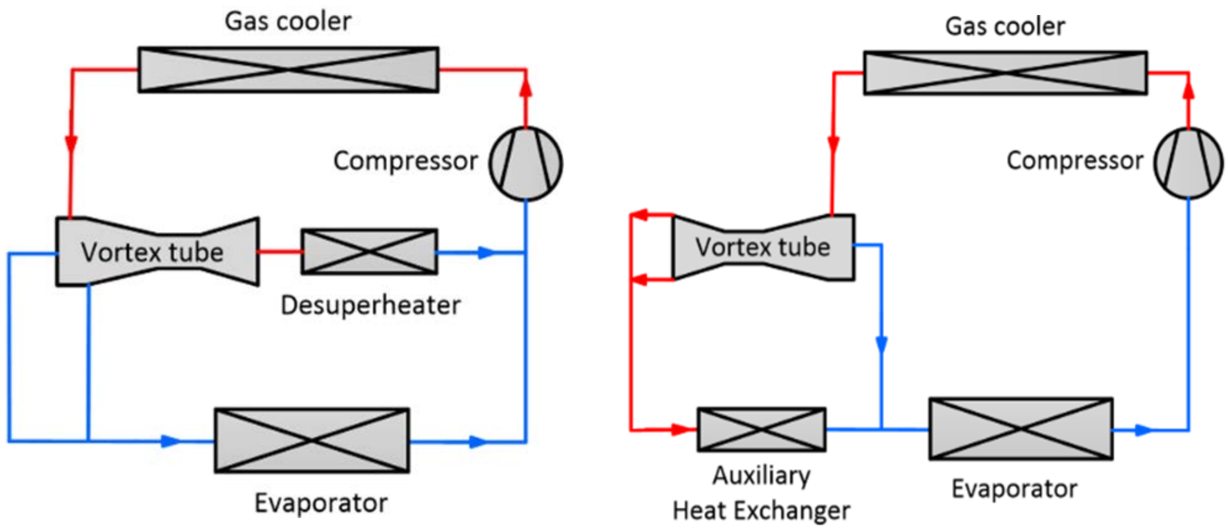


Figure 13. Schematic of the vortex tube expansion cycle proposed by Keller [54].

Keller's cycle might be a little difficult, but in 1999, Maurer suggested a transcritical cooling cycle with a vortex tube, as shown in Figure 14(a) [56]. There are no throttling valves in this system, and more significantly, there is a two-phase flow in the vortex tube for this system. The cooled gas from the gas cooler flows into the vortex tube and is separated into three low-pressure components: superheated hot vapor, saturated liquid and saturated cold gas. The saturated liquid and the vapor are then combined and poured into the evaporator. The superheated gas is cooled in the desuperheater and combined with the vapor from the evaporator to be compressed.

Likewise, Figure 14(b) shows a further modification of the transcritical CO_2 vortex tube cooling system proposed by Li et al. [57]. In this scenario, the cooled liquid stream leaves the cold side of the vortex, whereas the superheated steam stream leaves the hot side of the vortex. The superheated hot vapor reaches the auxiliary heat exchanger for cooling and then blends with the saturated liquid. Assuming 100 % gas-liquid separation efficiency, this vortex tube system could provide up to 37 % increase in cycle efficiency, and 50 per cent separation efficiency could provide around 20 % increase in cycle efficiency compared to the traditional basic system [57].

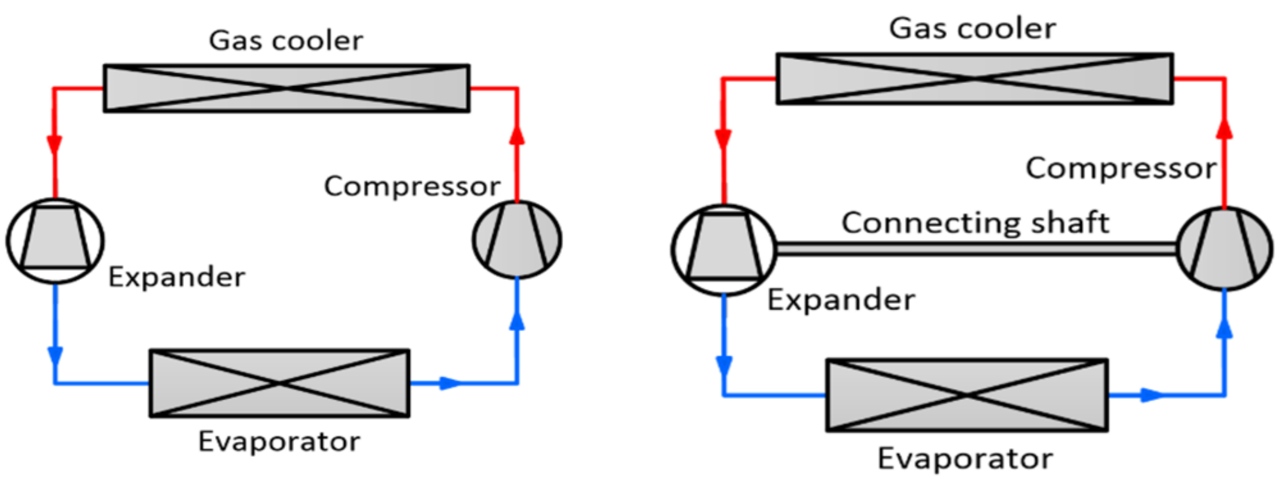


(a) (b)

Figure 14. Vortex tube transcritical CO_2 refrigeration system proposed by: (a) Maurer [56], (b) Li et al. [57].

3.4. Expander

Some other beneficial extension of the work protective measure for the transcritical CO_2 cycle is the expander, as the throttling loss is much higher than that of standard working fluid due to its physical properties. The potential for improvement of the system's energy efficiency depends mainly on the isentropic performance of the expander, which is usually lower than that of the compressor, resulting from the different phases of the fluid during compression and expansion, the single phase during compression and the two phase during expansion, as the two phase flow is more prone to friction. In this case, the transcritical cycle behaves better than traditional cycles, because the expansion process in the transcritical cycle contains a large part of a single phase, the fluid above the critical pressure is dense gas and only the final expansion includes two-phase fluid, and throughout the two-phase process the densities of CO_2 liquid and vapor are not as dissimilar as in traditional cy-cycling. All that means keeping the isentropic efficiency of the CO2 expansion process high enough, unlike what happens when trying to use expanders with common refrigerants [58]. The configuration of the expander system is shown in Figure 15, where the expander replaces the expansion valve, the expander can act either independently or coaxially with the compressor, approximately 37% of the compressor function.



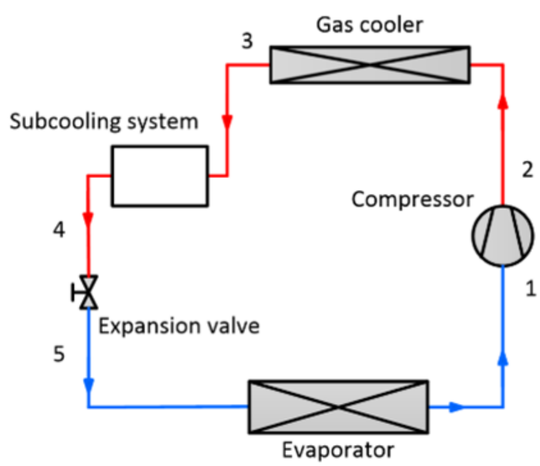
(a) (b)

Figure 15. Schematic of the CO_2 refrigeration system with expander working independently (a), or coaxially with compressor (b).

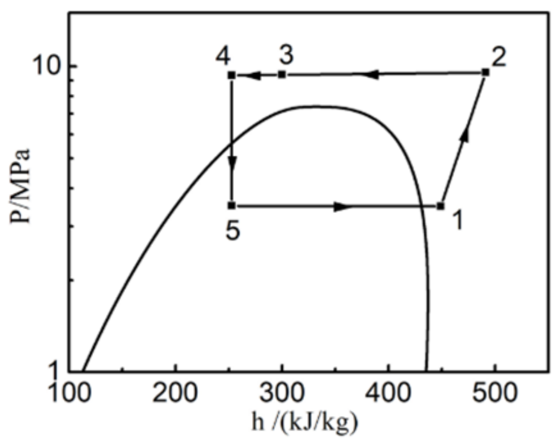
can be recovered by the CO_2 expander and this number increases with the inlet temperature of the expander [14].Tian et al.[59] theoretically measure the transcritical CO_2 cycle with the expander and 6-10 % higher recorded. Yang et al. [60] proposed both the first and second rule of thermodynamic analysis of the usage of the expander, claimed that the expansion system prevented a 50 % reduction in the exergic loss in the expansion phase and resulted in a 30% improvements in overall system of exergic performance, resulting in a 33% increase in the COP. Ma et al. [14] and Simarpreet et al. [21] have reviewed the major element of the expansion technologies.

3.5. Subcooling

In the standard CO_2 system, the supercritical CO_2 flow from the gas cooler outlet flows directly into the expansion tank to be throttled. Theoretically, the temperature of the gas cooler outlet for CO_2 is regulated by the ambient temperature, and there is an approach temperature that varies with the output of the gas cooler. After throttling, the evaporator inlet output (vapor fraction) cannot be further reduced due to a limited gas cooler outlet temperature and thus the basic cooling capacity is restricted respectively. In order to overcome this constraint, the researchers introduced sub-cooling technologies at the exit of the gas cooler and demonstrated performance improvement despite the higher-consumption work in the sub-cooling system. The combination of the sub-cooling and the main CO_2 cycle is shown in Figure 16. The sub-cooling system is used to further cool the CO_2 from the gas cooler, which is a 3-4 phase in Figure 16(a) therefore the efficiency of the evaporator inlet after throttling is reduced as in point 5 of Figure 16(b) and the special cooling power increases. Two sub-cooling technologies have been widely studied for this cycle: Dedicated Mechanical Subcooling (DMS) and Thermoelectric Subcooling (TES). Compared to the basic system, the total system COPs were validated in order to increase significantly for the two sub-cooling systems, considering the minimal extra labor expended in the sub-cooling system[61], [62].



(a)



(b)

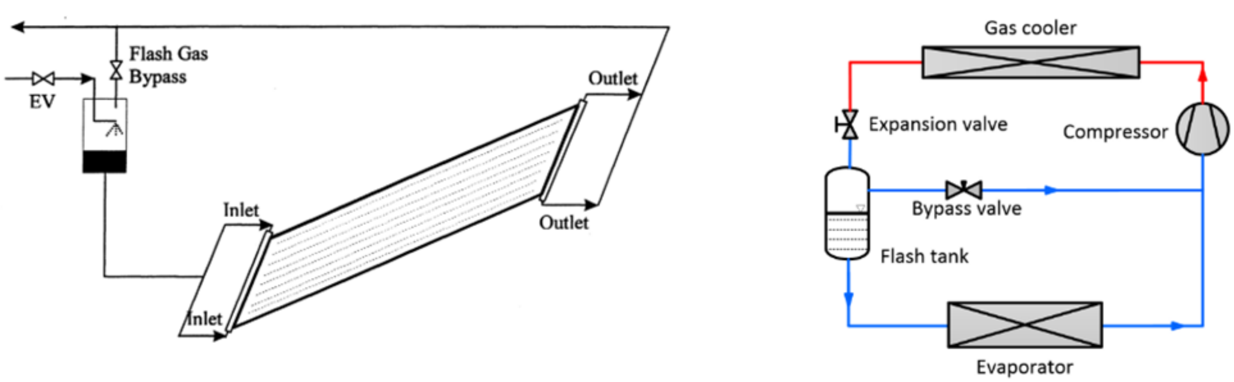
Figure 16. Schematic of the CO_2 subcooling refrigeration system (a), and its 1gP-h diagram (b).

The sub-cooling system is used for the DMS process, in Figure 16 (a) usually uses a subcritical steam compression system, which eliminates heat to the same heat sink as the main CO_2 cycle. Theoretically, Llopis et al. [63] investigated the effects of subcooling degrees offered for supermarket application by the mechanical subcooling system with various refrigerants at a refrigeration facility. It should have been reported that the overall performance of COP enhancements are identical for different refrigerants used in the subcooling cycle, such as R290, R1270, R1234yf, R161, R152a and R134a. Subsequently, they conducted an experimental analysis on the single-stage CO_2 transcritical cooling plant with R1234yf DMS system, evaluating the cooling power and COP improvements with maximum capacity increases of 55.7 % and 30.3 % at COP [64]. For the TES process ,The subcooling system is in Figure 16(a) usually use a Peltier-based thermoelectric module, where a DC current is applied to both semiconductors, the temperature difference between the two may be produced, then the CO_2 may be subcooled. This definition was first introduced by Schoenfield et al [96, 97] at the base CO_2 transcritical refrigeration facility.

3.6. Flash gas bypass

First, the idea of flash gas bypass was placed forward to address the challenge of the issue of two-phase flow delivery in evaporators, from headers to tubes [65], [66] ,which is an inherent problem for almost all refrigerants. Many devices have used small or micro-channel tubes in evaporators to handle the extremely high pressure for the CO_2 application, and the distribution issue is becoming much more severe [13]. In addition, in a traditional direct expansion vapor compression system (DX), which is not unique to refrigerants, the refrigerant state at the outlet of the expansion device is in a two-phase situation, resulting in the vapor state entering the evaporator without having a significant cooling impact. One potential solution to solving the above issues simultaneously is the introduction of a Flash Gas Bypass (FGB), the principle behind this technique is to bypass the vapor state flow preventing entry into the evaporator and being sucked explicitly by the compressor itself. Figure 17 demonstrates the FGB solution schematic and its incorporation into the CO_2 scheme. The gas cooler's supercritical CO_2 is thrown into two-phase condition and reaches the flash tank, where phase separation occurs. The separate vapor stream then flows into the bypass circuit and is directly pumped by the compressor, while the separate liquid flow with a content equal to zero feeds the inlet header of the evaporator. Note that the bypass valve in Figure 17 (b) plays an extremely important role in regulating the evaporator outlet conditions, because closing the bypass valve makes more refrigerant via the evaporator, thus minimizing overheat at the exit of the evaporator [65]. Tuo and Hrnjak assessed the FGB scenario in an R134a mobile air conditioning system and showed that the COP can be substantially improved at the same cooling power by 37 % -55 % over the DX system, they speculated that this is due to improved refrigerant distribution and reduced refrigerant pressure drop in the evaporator. Figure 18 displays their acquired surface temperature of the evaporator by infrared imaging showing the distribution of refrigerant among the microchannel tubes.

It is apparent that in FGB mode, a more consistent profile is obtained because only liquid refrigerant reaches the inlet header, effectively removing the consistency misdistribution [67]. In addition, the flash gas bypass device control strategy and dynamic behavior during startups and transients were clearly described and experimentally investigated by the same research group [68]. It is regrettable that no such CO_2 condition regulation studies have been performed. For the CO_2 FGB system, numerous studies have based experimental or thermodynamic methods on system efficiency.



(a) (b)

Figure 17. Evaporator with flash gas bypass configuration [65] (a), and its integration in the CO_2 system (b).

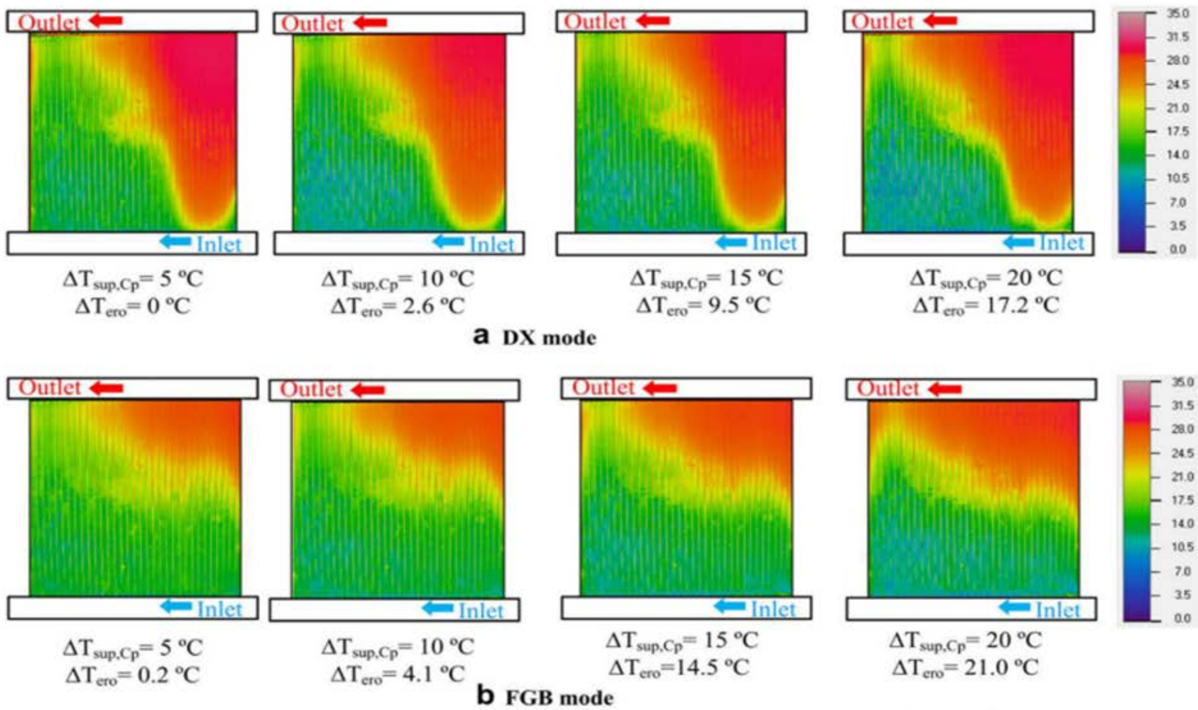


Figure 18. Evaporator surface infrared images in FGB and DX modes [67].

3.7. Parallel compression

The parallel compression configuration is identical to the FGB system was employed above, the distinction is based on the compression process, where an external auxiliary compressor is used to compress the flash gas to gas cooler pressure rather than using the same compressor that primarily compresses the evaporator vapour. This concept's main purpose is to reduce the throttling losses [69]. Figure 19. Demonstrates the parallel compression method seen, apart from using two separate compressors, the dotted line means that the parallel compression can also be done by a coupling compressor, i.e. a dual single shaft compressor, a T-shaped shaft and parallel shafts, respectively[70], [71]. Another way to compress the flash gas directly to the gas cooler pressure is to conduct the gas through a vent into the main compressor's compression chamber [72]. In both situations, the compression power required is reduced and, in addition, the compressor provides improved efficiency at a lower compression ratio, resulting in cooling systems using parallel compression achieving at least the same efficiency level or even more effective than flash gas bypass cooling systems [73]. Comparing the transcritical CO_2 system with parallel compression with the other two systems, Sarkar and Agrawal also used the flash gas bypass system. They found the parallel compression cycle not only increases the optimum COP for cooling but also decreases the optimum pressure for discharge. The parallel compression system is particularly efficient for lower temperature applications, achieving a COP improvement of 47.3 % over the basic system for the selected ranges of operating conditions, the ideal intermediary pressure is stated to differ only marginally [74]. Bell [75] conducted a theoretical and experimental study on this system, taking into account the different gas cooler outlet temperatures, suction superheat, medium vapor pressure and compressor swept volume ratio, it is concluded that CO_2 parallel compression is more beneficial in terms of COP and capacity under the same conditions than hydrocarbon. The compressor swept volume ratio needs to be changed to optimize a wide range of gas-cooler outlet temperature. Da Ros[76] explored the optimization of parallel compression, achieved through tuning of the compressor pressure discharge and displacement ratio, providing COP values and discharge pressure of both compressors under a few operational conditions.

He et al. [77] compared CO_2 parallel compression with the basic cycle, the efficacy of parallel compression is most pronounced in low evaporation temperature and high ambient conditions, with COP increases by up to 21 % and discharge pressure decreases by 5.3 bar across the considered parametric range, while COP improvements are usually below 10 % when the system operates in subcritical conditions.

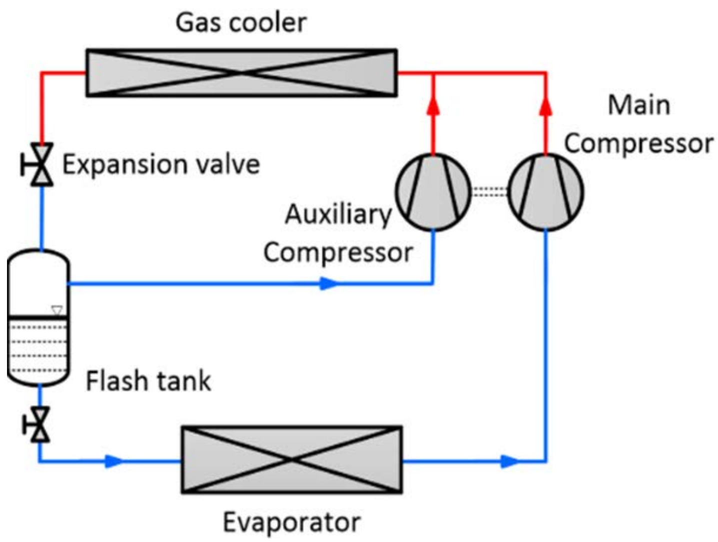


Figure 19. Schematic of the parallel compression system

3.8. Two-stage compression

In the simple CO_2 method, which is a single-stage compression cycle, there is a slight loss of energy during the actual compression process, usually the isentropic efficiency of the compressor is a strong compression ratio property, which can be written as [78]:

$$\eta = 1.003 + 0.121\ \varepsilon \tag{2}$$

This equation was obtained by fitting the experimental data of the CO_2 compressor, where ε stands for the compression ratio described by the discharge and suction pressure ratio. It is easy to assume that the reduction of the ε will improve the efficiency of the compressor. Historically staggered compression is an efficient way to do this by decreasing the compression ratio for each stage. Multi-stage compression with more than two stages is, with course, more effective in minimizing exergetic losses, but in actual use, multi-stage compression systems are seldom used due to the complexity and implementation costs. In addition, if the compression ratio is extremely low, there will also be a decrease in isentropic efficiency [58]. As a consequence, a two-stage compression method is commonly studied for study and application. Kim et al. [13] published a thorough analysis of two-stage, literature-based, transcritical CO_2 systems prior to 2004. However, to date, there seems to be no ideal intercooling process with different situations between the two compressions. Figure 20 introduces four standard two-stage CO_2 cooling systems with various intercooling scenarios, in which Figure 20(a) is the simplest intercooler system and uses one expansion valve and two-stage compressors. The discharge pressure of the first-stage compressor is cooled by the external fluid, typically the same for gas cooling, resulting in a substantial increase in energy efficiency and a decrease in compression function for both compressors and a significant reduction in the discharge temperature of the upper compressor. For this method, Srinivasan [79] noticed that equivalent discharge temperatures at the end of each stage are an acceptable criterion for a significant reduction in maximum cycle temperatures. Defining the inter-stage pressure index as n [79]:

$$n = \frac{ln\,P\,i}{ln(p1s\ p2h)} \tag{3}$$

When Pi is the intermediary pressure, P_{1s} is the suction pressure of the very first stage compressor and P_{2h} is the discharge pressure of the second stage compressor, it studied the effects of evaporation temperatures on the optimum intermediate pressure and established that more compression needs to be performed in the lower stage as the evaporator temperature decreases with rising inter-stage pre-pressure. Figure 20(b) is a double throttling and double compression cycle with a flash tank, the supercritical gas stream is throttled to the intermediary vapor-liquid mixture and flows into the flash tank in which the two phases are separated, then the vapor is pumped into the second compressor. The liquid phase is throttled again and reaches the evaporator, the vapor flow from the first compressor is combined with the vapor phase from the flash tank [58].

Cho et al. [80] studied experimentally this method, which varied the amount of refrigerant charge, compressor frequency and expansion valve openings, and compared the findings with the method. Figure20 (a) The COP increases with the openings of the first and second stage expansion valves resulting from the reduced compression ratio. After all, some loss of system cooling power is incurred as the mass flow rate via the evaporator decreased when the opening of the first stage expansion valve was enhanced. The optimum control of both expansion valves openings is therefore needed in this system. Aprea and Maiorino [81] performed experimental research on this system to enhance heat-rejection pressures varying with ambient temperatures. They proposed a simplified model to predict the optimum heat-rejection pressure confirmed by experimental results. The correction developed to predict optimum pressure is written as [81]:

$$P_{opt} = \frac{2.7572 + 0.1304T_e - 3.072K/C}{1 + 0.0538T_e + 0.1606K/C} T_{out,gc} - \frac{8.7946 + 0.02605T_e - 105.48K/C}{1 + 0.05163T_e + 0.2212K/C} - 0.003T_{out,gc} + 0.174 \tag{4}$$

Where P_{opt} is the optimum pressure, T_e is the temperature of evaporation, $T_{gc,out,}$ is the temperature of the outlet cooler steam, K and C. Figure 20 (c) indicates a flash intercooling system similar to the flash gas injection method in Figure 20 (b) The distinction is that the compressed vapor from the very first-stage compressor reaches the flash intercooler and is de-heated by evaporation of liquid CO_2 at an intermediate temperature.

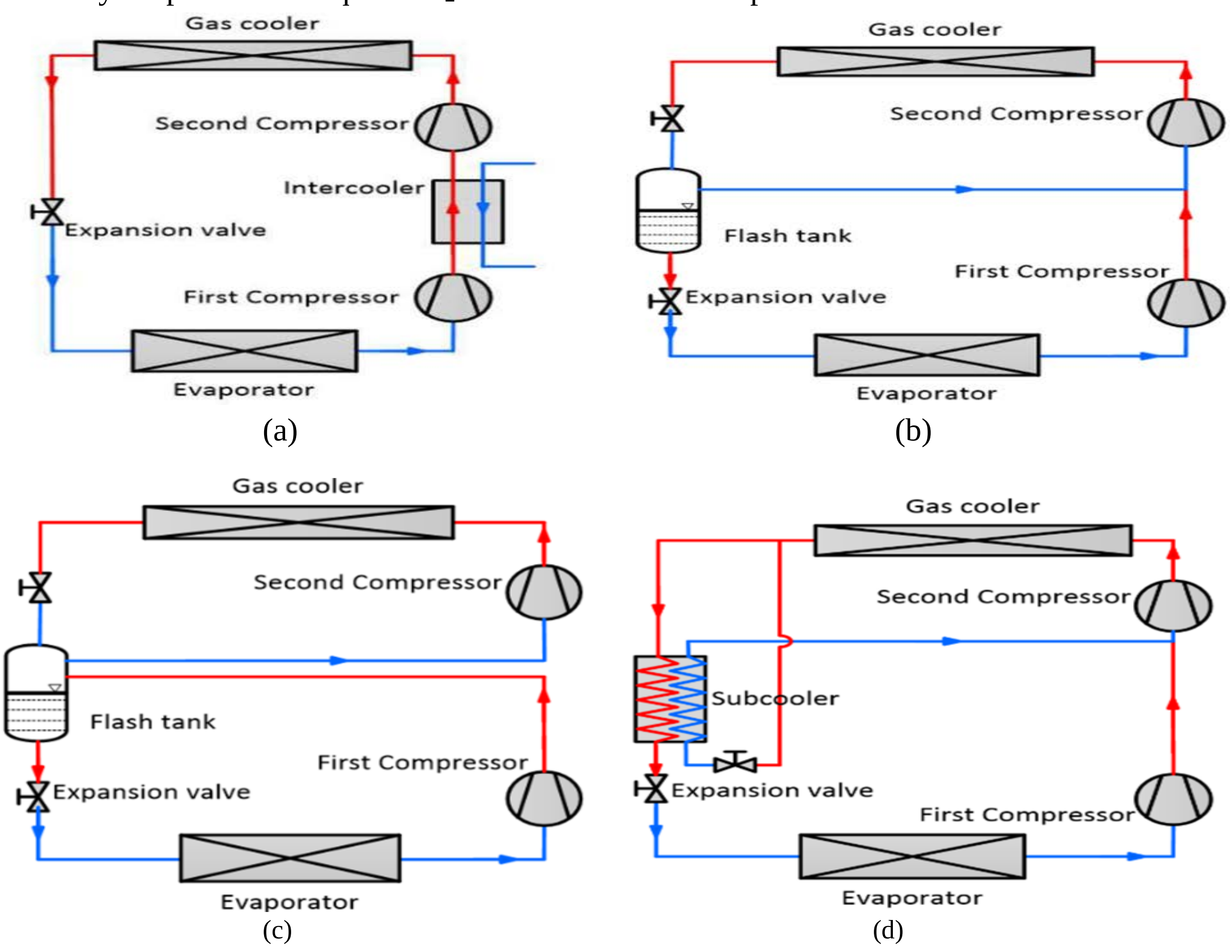


Figure 20. Schematic diagram of CO_2 two-stage compression systems using different intercooling scenarios: (a) intercooler; (b) flash gas injection; (c) flash intercooling; (d) subcooler.

This results in a double impact because, on the one hand, this method raises the CO_2 vapor flow rate to be pumped by the second-stage compressor while, on the other hand, the suction temperature for the second-stage compressor is greatly reduced compared to the flash gas injection system[4]. Zhang et al. [82] measured the efficiency of this system by thermodynamic analysis compared to the system. Figure 20 (a) (a). In Figure 20(d) structure replaces the flash tank with a heat exchanger called a subcooler used to cool the steam stream before entering the expansion valve. Compared to the devices using the flash tank. Figure 20 (b) and (c) the elimination of the flash tank leads to a cost savings, but the temperature of the fluid entering the expansion valve before the evaporator is marginally higher since there is an approach temperature between the hot and cold fluid in the subcooler.

The cycles with the subcooler and the flash tank behave very similarly, and the COP is almost the same in conventional subcritical applications, except for CO_2 transcritical cycles, the isobaric curves in the dense gas or liquid area do not strictly conform to the lower limit curve in the T-s diagram, so the split cycle is usually not penalized as opposed to the open flash tank cycle [58].

3.9. Evaporative precooling

As stated in section 2, one of the key challenges for the basic CO_2 system is the intrinsic degradation of efficiency in application of high ambient temperature. In other words, the CO_2 gas cooler prefers low-temperature inlet cooling medium, in this context, the method of evaporative cooling to precool the air before reaching the gas cooler could be a promising technology to enhance the energy efficiency of the basic CO_2 system.

Conventionally, spraying water on the condenser is a well-known refrigeration solution which faces peak temperatures [83]. Even if it exhibits good efficiency, it needs a huge amount of water and leads to disaster of scaling and corrosion as water contact with fins, unless a high cost surface treatment is administered. Considering the broad variation for most regions of the world between dry-bulb temperature (DBT) and wet-bulb temperature (WBT) in summer, evaporative cooling is increasingly desirable nowadays for enhancing energy efficiency through both indirect and direct methods. For his principle of adiabatic saturation and the traditional psychrometric chart [84], the origins of evaporative cooling can be traced to Dr. Willis H. Carrier.

Figure 21 displays two potential configurations where 100% (a) or just 33% (b) of the gas-cooler inlet air is precooled. The two solutions were analyzed by Girotto et al. [85], [86] and found that the COP increase was 17 % for the 30% precooling solution and 27 % for the 100 % precooling.

By replacing the 100 % precooling solution with the 30% precooling solution, water consumption is decreased by 70 %. Their results are summarized in Table 1. Therefore, the best choice should be chosen with regard to the actual environment of the region where the gas cooler is to be built and the availability of water should be considered [58].

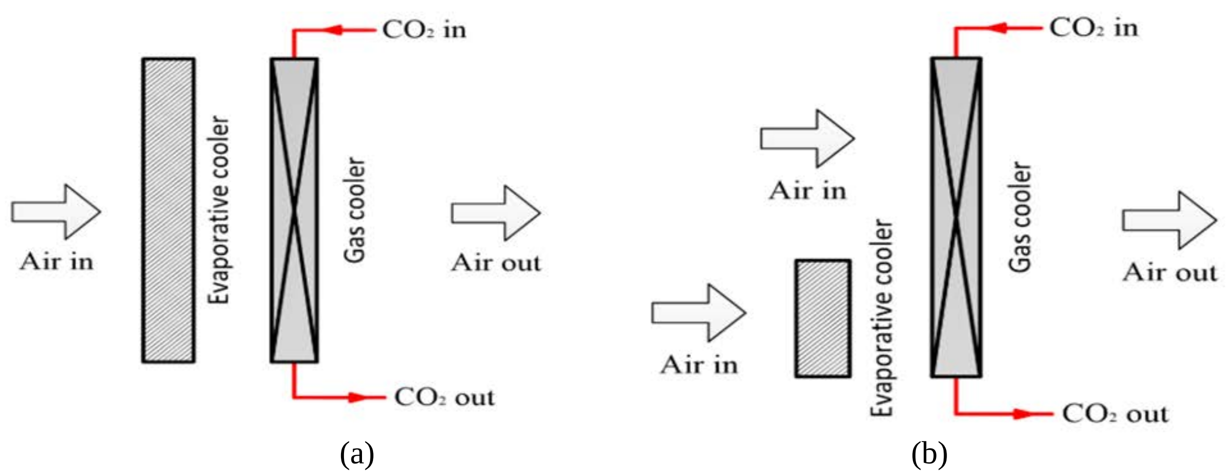


Figure 21. Application of evaporative cooler in the basic CO_2 system with 100% air precooled (a), and 30% air precooled (b).

Table 1: COP and water consumption of two different pre-cooling solutions, for 40°C and 50% relative humidity external conditions [86].

Solutions	Air temperature after evaporative cooling [°C]	CO_2 temperature out of the gas cooler [°C]	Optimal high pressure [bar]	COP [-]	COP (with fan power input) [-]	Water consumption [$(m^3/h)/kW$]
No precooling	40.0	41.5	116	1.38	1.38	-
100% precooling	32.2	34.7	100	1.75	1.71	0.000926
30% precooling	32.2	35.6	114	1.6	1.59	0.000291

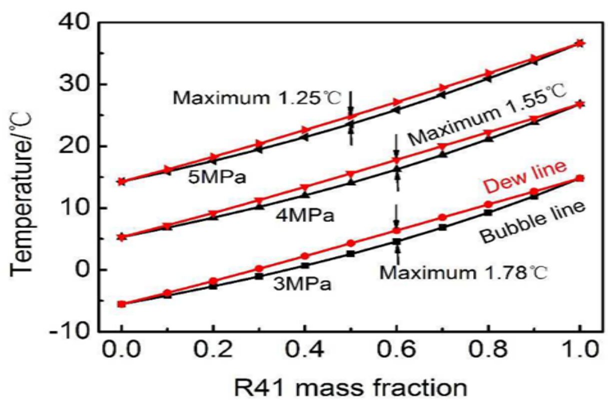
3.10. CO_2-based mixture

One of the challenges of all the above-mentioned technologies to enhance the basic system is to increase the complexity of the system, all of which require additional equipment to achieve the goal, which undoubtedly requires additional costs and increased installation space, thus reducing economic efficiency. In such cases, certain CO_2-based mixtures like refrigerant blends and CO2-absorption cofluids, that are all CO_2-based, have become a promising drop-in technology without any modification to the basic device setup.

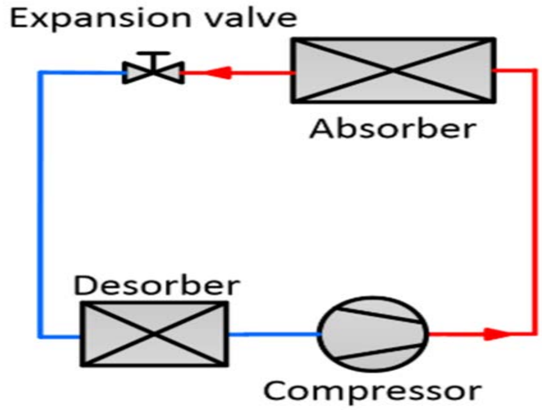
3.10.1. CO_2-based refrigerant mixtures

Since the two unavoidable disruption created by using CO_2 as a refrigerant in the basic system, as protection issues arising from high pressure and efficiency of operation, one can easily assume that adding another coolant to CO_2 will increase system performance and reduce operating pressure, of course, the adding refrigerant must meet the conditions of low operating pressure, high efficiency and low GWP. The temperature slip during the evaporation or condensation process allows the refrigerant mixtures to be categorized into zeotropes, Based on the difference in normal boiling temperature for each refrigerant, near-azeotropes and aceotrops. Take the CO_2 / R41 mixture for example, in Figure 22(a), the temperature and composition variations of CO_2 / R41 mixtures at fixed pressures represent the bubble line and the dew line, and the two lines split the diagram into three regions: the subcooled liquid, the superheated vapor and the two-phase regions. The temperature at the dew point of the mixed refrigerant with some R41 mass fraction is marginally higher than the temperature at the bubble point.

A temperature slip occurs during the evaporation or condensation process, simply because of this element. The mean temperature drifts for fixed pressures of 3, 4 , and 5 MPa are 1.75 °C, 1.55 °C, and 1.25 °C , respectively, which are significantly higher than azeotropic refrigerants without even a temperature slip, but considerably lower than those of zeotropic mixtures[87].



(a)



(b)

Figure 22. (a)Temperature-composition diagram of CO_2/R41 blends at fixed pressure [88]. (b) Schematic diagram of the CO_2 absorption-compression refrigeration system.

It is worth further studying the heat transfer properties in the heat exchangers and optimizing the compressor for the system using CO_2-based mixtures, as the drop-in system has some potential for improvement with respect to heat transfer and compressor efficiency [88].

3.10.2. CO_2-absorption cofluids

The transcritical operating theory works a far higher operational pressure in the simple transcritical CO_2 system than traditional refrigerants, leading to much higher costs and poor system efficiency. Prior study has attempted to remove this issue and at the same time increase system performance by replacing the evaporation and gas cooling with absorption and desorption from an absorbing fluid. Thus the cofluid of CO_2-absorption circulates in the CO_2 absorption-compression cooling method shown in Figure 22(b), this cycle can be considered as an extreme case of a zeotropic refrigerant cycle due to an entirely non-volatile absorption of fluid rather than a refrigerant having a different normal boiling temperature to CO_2. At Figure 22(b), the evaporator and the gas cooler are substituted by the absorber and the desorber; the two-phase flow of the CO_2-absorbing cofluid is compressed in a compressor known as the "wet compression." The mixture is cooled by ambient in the absorber, and CO_2 from vapor is absorbed into the cofluid.

During the throttling phase the mixture then expands, and CO_2 begins to desorb. The heat source heats up the extended low-temperature mixture in the desorber, and extracts more CO_2 into the vapor process. Finally, the cofluid, which absorbs two phases of CO_2, flows into the compressor to begin another new cycle. The discharge pressure is greatly reduced to no more than 35 bar due to the absorbing fluid absorbing CO_2 in the absorber, so that the current components for conventional-refrigerant based vapor compression systems can be directly incorporated.

4. A summary of preceding research

According to the above-mentioned key findings related to each technique, the energy efficiency enhancements of the updated technologies evaluated above over the basic transcritical period of CO_2 are summarized in Figure 23.Also in table 2 shown properties of several refrigerants also.

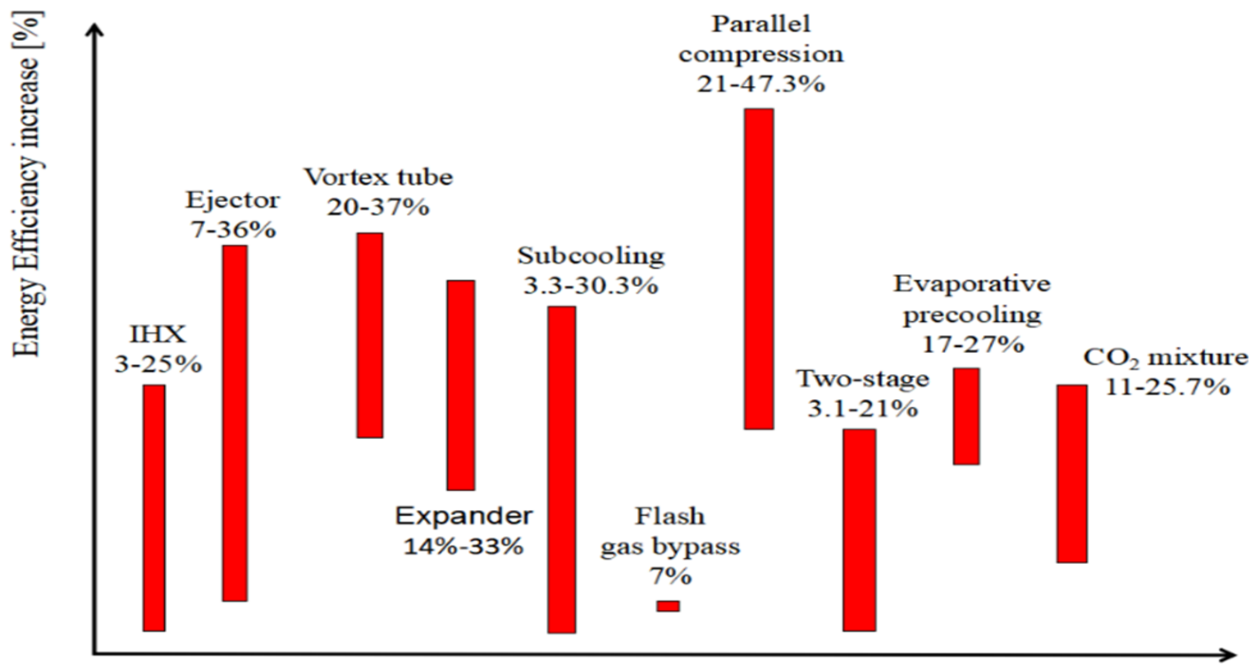


Figure 23. Summary of the energy efficiency improvements for the modified technologies [23].

Table 2: Properties of several common refrigerants [58].

Refrigerant	Evaporator temperature = -30 °C			Evaporator temperature = 0 °C		
	Saturation pressure (bar)	Liquid thermal conductivity ($Wm^{-1}K^{-1}$)	Vapour density (kg/m^3)	Saturation pressure (bar)	Liquid thermal conductivity ($Wm^{-1}K^{-1}$)	Vapour density (kg/m^3)
R22	1.64	0.1084	7.38	4.98	0.0947	21.23
R407C	1.62	0.1187	7.21	5.13	0.102	21.88
R134a	0.84	0.1058	4.43	2.93	0.092	14.43
R410a	2.79	0.1293	10.57	7.99	0.1099	30.63
R404a	2.05	0.0862	10.69	6.05	0.074	30.72
R744	14.28	0.1469	37.1	34.85	0.1104	97.65

5. Conclusion

From its historical context, CO_2 as a refrigerant has been studied for the unique properties that influence its performance in the refrigeration industry. As a consequence of its superior properties, especially with regard to refrigeration, I suspect that R744 will be the superior refrigerant in many applications of refrigeration technology in the future. This analysis summarizes recent developments in updated technologies for the transcritical CO_2 refrigeration cycle over the last two decades, which have been shown to be promising for the enhancing the energy performance and operating pressure. This research is aimed to help determine the appropriate solution for transcritical CO_2 refrigeration while assessing various implementations. The IHX studies are the most detailed, there is no uncertainty that they would be used in transcritical CO_2 cooling; numerous disadvantages for ejectors need to be addressed, such as high cost and considerable complexity; complicated internal flow conditions may require more visualization on the basis of advanced techniques as well as the CFD model. Primary experimental validation should be performed on vortex tubes, thermoelectric subcooling and CO_2-absorption liquids to evaluate their effect on CO_2 cooling output and energy efficiency. More practical investigations are necessary for the mass and heat transfer and behavior of hardware components, as well as a broader range of environmental conditions for and integrated system. It should be stated that this analysis only includes updated systems using the above-mentioned single technology, whereas the effects of the possible development of two or more technologies combined concurrently in one system are not included, even though this is most frequently used at the level of practical implementation.

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