Research Journal of Engineering Sciences ISSN 2278 – 9472
Vol. **13(1),** 28-40, January **(2024)** *Res. J. Engineering Sci.*

Review Paper

Transcritical carbon dioxide refrigeration and air conditioning cycles and applications: State of the art

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Available online at: www.isca.in, www.isca.me
Received 28th July 2023, revised 24th October 2023, accepted 3rd December 2023

Abstract

The use of less polluting refrigerants is now a reality in the framework of research to lessen the environmental impact of refrigeration units. The drive to employ ecologically benign and safer-to-handle refrigerants has resulted in the recent increased usage of carbon dioxide in refrigeration cycles, which develops substantial benefits, particularly in Western countries. CO_2 systems are less efficient than conventional systems due to the transcritical nature of the CO_2 refrigeration cycle. As a result, numerous cycle adjustments can be included to increase system performance. This review article contains a database showcasing the state of the art in theoretical, practical, and technological developments made on transcritical CO_2 cycles, as well as their field of application. According to research, the biggest improvement in the transcritical cycle can be obtained by replacing the expansion device with a work-recovery expansion machine or by using numerous stages of compression. However, these are expensive upgrades in terms of buying price. As a result, recent research has mostly focused on ejector-driven transcritical cycles due to the large performance boost, lack of moving parts in the ejector, and low cost. Current developments and challenges in important application areas are summarised, and future research directions are highlighted.

Keywords: Refrigeration, Transcritical Cycle, Carbon Dioxide, Energy performance, Sustainable Development.

Introduction

Global environmental consciousness has lead to the hunt for refrigerants with the least amount of environmental impact. Hydrofluorocarbons (HFCs), which are commonly utilised refrigerants in the refrigeration industry, have negative environmental effects[1]. In fact, the majority of first-generation HFCs contribute to global warming. Hydrocarbons are highly efficient but very combustible fluids. On the other hand, the olefin family (HFO), which has minimal global warming potential, is dangerous because to its flammability. As a result, they are not permitted to be used in dry expansion setups for commercial or industrial refrigeration.

Carbon dioxide (CO_2 or R744), on the other hand, is a non-flammable, non-toxic, less expensive, and more ecologically friendly fluid that was one of the earliest refrigerants employed in refrigeration history[1]. CO_2 is known to all since it is present in the atmosphere at a volume concentration of 0.04%[2]. It was widely utilised around the turn of the twentieth century, particularly in naval refrigeration, where it was the dominating refrigerant from 1950 to 1960[3]. The high volumetric capacity of the refrigerant induces the use of small-dimensioned components (compressors and pipes)[4].

This property has also enabled the development of very compact tube heat exchangers with reduced pipe diameters. Carbon dioxide is also appropriate for low-capacity applications such as automobile air conditioning. Due to its low viscosity, it encounters low pressure drop. When a suitable refrigerant is paired with carbon dioxide in cascade refrigeration systems, which are generally highly compact and less demanding in terms of refrigerant quantity, low temperatures (-30°C to -100°C) can be attained. These fundamental advantages distinguish CO_2 as a refrigerant in the refrigeration industry and make it a solid choice for limiting the use of high GWP (Global Warming Potential) refrigerants[4].

A considerable pressure loss occurs in the expansion process in transcritical operating mode due to the large difference between high and low pressure during system operation. This has a negative impact on the efficiency of systems and is also the cause for the low performance of CO_2 transcritical cycles. Faced with the phasing out of a range of refrigerants already on the market, as well as the technological challenges experienced when installing CO_2 refrigeration systems, it is imperative to push additional research programmers on the use of CO_2 as a refrigerant.

Following that pattern, the article attempted to review existing research on the CO_2 refrigeration cycle, transcribe it, and inform about future research possibilities to investigate. It gave ideas for the enhancement of transcritical systems with the goal of generating more environmentally friendly refrigerating machines, resulting in lower greenhouse effect gas emissions, as well as a long-term reduction in energy consumption.

Transcritical refrigeration cycles

For negative refrigeration, the low pressure in transcritical cycles is around 40 bar, while the high pressure evolves about 100 bar. While heat transfer from the refrigerant to the outside is taking place, the high pressure is over its critical value of 73.8 bar (Figure-1). As a result, there is no longer any phase change. The gas cooler's outlet temperature remains unaffected by pressure within the supercritical zone. Several simulations, however, have shown that there is an optimum gas cooler pressure for a given end-of-cooling temperature[5]. Heat transmission in the gas cooler occurs in sensible heat rather than condensation.

Booster type transcritical refrigeration cycles: Figure-2 depicts the system diagram of a booster type transcritical refrigeration cycle. CO_2 released from the low-pressure compressor is directed into the suction of the high-pressure compressor (point 1). Following vapor-liquid separation in the intermediate tank (point 8), the saturated vapor undergoes expansion through an electronic valve (5 and 6) to interact with the separated liquid, facilitated by the sub-cooler. The sub-cooled liquid, upon exiting the sub-cooler (9), proceeds to the different evaporators (medium and low-temperature) through suitable expansion valves (nbr 10 and 11). The superheated vapor generated by the low-stage compressor combines with the vapor exiting the medium-temperature evaporator (point 12) and the vapor released from the sub-cooler (point 7). The mixed vapour flows towards the high stage compressor (point 1), is compressed and discharged then into the gas cooler (point 2). Such booster refrigeration systems are widely adopted for refrigeration in supermarkets, due to their negligible environmental impacts[7–9].

Cascade refrigeration cycles: A cascade refrigeration system is made up of two refrigeration units, one at a lower temperature and one at a higher temperature. An internal cascade heat exchanger connects both units thermally. Carbon dioxide can be used for the low temperature stage. The internal heat exchanger serves as a condenser for the low temperature unit whereas an evaporator for the high temperature unit, increasing the refrigerating machine's performance[10]. Figure-3 depicts an NH_3/CO_2 cascade system in action. Each refrigeration system in this system comprises of a compressor, a condenser, an expander, and an evaporator. The condenser in this refrigeration system rejects a heat flux Q_h from the condenser at the condensing temperature T_h into its surroundings while the evaporator absorbs the cooling load Q_b from the cooling space at the evaporation temperature T_b. The heat absorbed by the evaporator at low temperature plus the energy corresponding to the work of the compressor of the same cell is equal to the heat absorbed by the evaporator of the high temperature cell. If T_{ecB} and T_{ecH} represent respectively the condensing and evaporating temperatures of the cascade condenser. The temperature difference $\Delta T = T_{ecB} - T_{ecH}$ across the cascade condenser is an important design parameter. Its order of magnitude of ΔT vary from 6°C to 10°C.

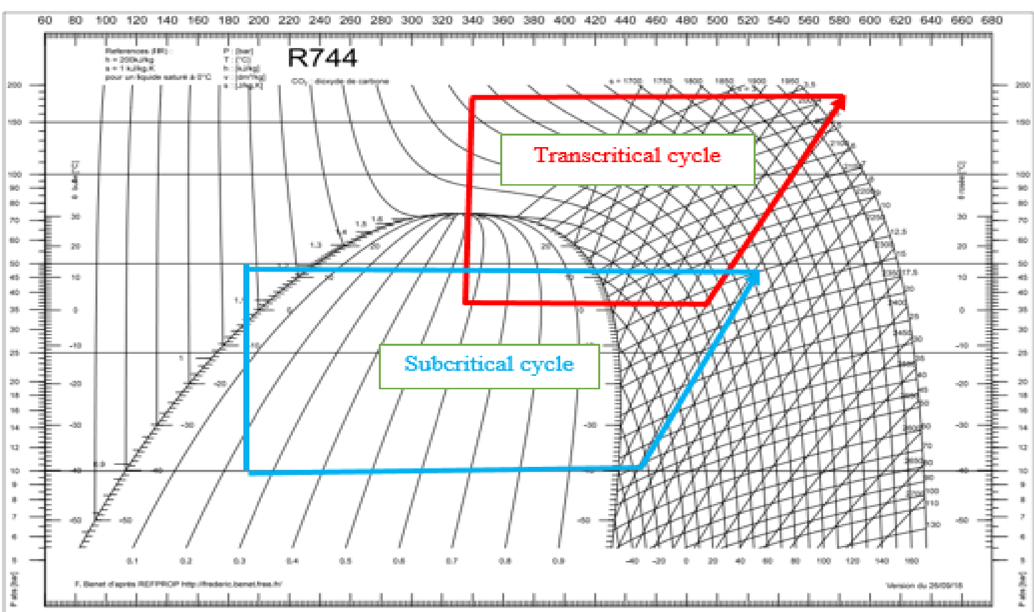

Figure-1: Carbon dioxide refrigeration cycle[6].

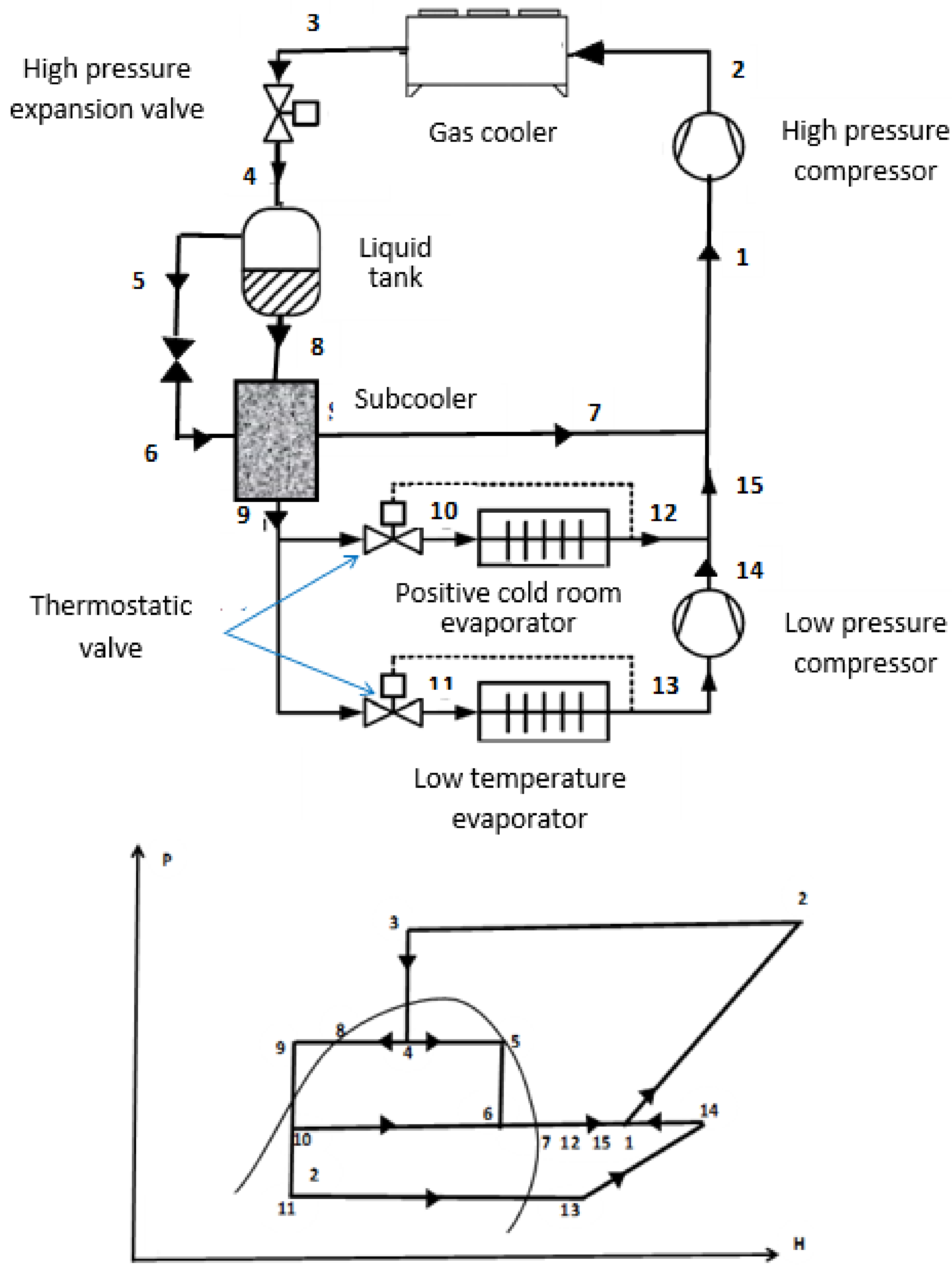

Figure-2: Booster type transcritical refrigeration cycles[6].

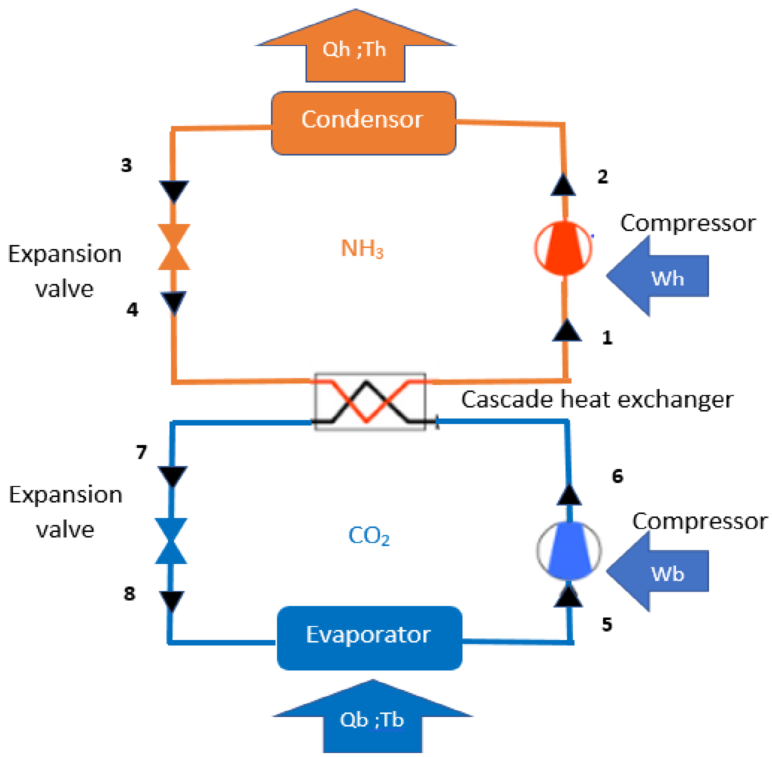

Figure-3: Cascade refrigeration system operating with NH_3/CO_2[6].

Ejector refrigeration cycles: Air conditioning and refrigeration systems can benefit from the usage of ejectors[11]. Figure-4 depicts the block diagrams of the conventional cycle and the ejector-equipped cycle. The isenthalpic process is illustrated in Figure 4 by the transformation between points 3 and 11, whereas the isentropic process is represented by the transformation between points 3 and 4. The refrigerant fluid specifies a cycle between points 8, 2b, 3, 11, and 8 for the conventional cycle. The ejector cycle, on the other hand, involves two fluid flows: the primary flow and the secondary flow. A compressor circulates the primary flow via the gas cooler, ejector, and separator (points 1, 2, 3, 4, 10, 5, and 1), while the secondary flow circulates in the expansion valve, the evaporator, the ejector and the separator (points 4 to 10 The constant-section mixing of the primary and secondary flows reaches the diffuser (points 10 and 5). In the ideal cycle, saturated vapour from the liquid-vapor separator enters the compressor and isentropically compressed to a high pressure while increasing in temperature. In the gas cooler, heat is rejected at constant pressure. The supercritical primary fluid is expanded (isentropic evolution) at the pressure of the mixture in the primary nozzle. The main fluid gains kinetic energy during expansion. The pressure at point 1 is greater than the suction pressure at point 8 in the typical cycle. This causes the compressor work of the cycle with ejector expansion to be less than that of the regular cycle. The motive flow, or primary flow, is accelerated and expanded through the motive nozzle from 3 to 4. Furthermore, as the fluid approaches the saturation curve, it partially vaporises and becomes two-phase. In actuality, however, the expansion process proceeds very quickly, making it practically impossible to preserve the two-phase mixture's hydrodynamic and thermodynamic equilibrium. As a result, meta stabilities (the feature of a seemingly stable condition that a disturbance can fast transform to an even more stable one) could induce delayed evaporation of the flux, influencing the ejector's performance. The high velocity two-phase flow offers momentum transfer and secondary fluid entrainment as it exits the evaporator. The extremely high velocity flow reduces pressure at point 4 and promotes secondary fluid flow from point 8 to point 9. The two flows are combined at the mixing chamber's input to form a single flow in the constant section area. The mixed flow is directed from point 10 to the diffuser through the secondary nozzle. The refrigerant decelerates in the diffuser, increasing its pressure (point 5). At point 5, the refrigerant flows to the separator. The vapour phase refrigerant in the separator runs as the primary flow via the compressor (points 1–2), while the liquid refrigerant from the separator flows through the expansion valve and into the evaporator (points 6, 7, and 8), as the secondary flow. The points A and B depicted on figure 4 represent the enthalpies at the pressure of the diffuser when the process is respectively isenthalpic and isentropic. Meanwhile, points 1 and 8 represent the points at the compressor suction and at the evaporator outlet, respectively. A high ejector compression ratio decreases the compressor compression ratio. Increasing the entrainment rate will reduce the mass flow rate of the compressor for a given refrigeration output. Ejector efficiency increases as drive rate and/or compression rate increases.

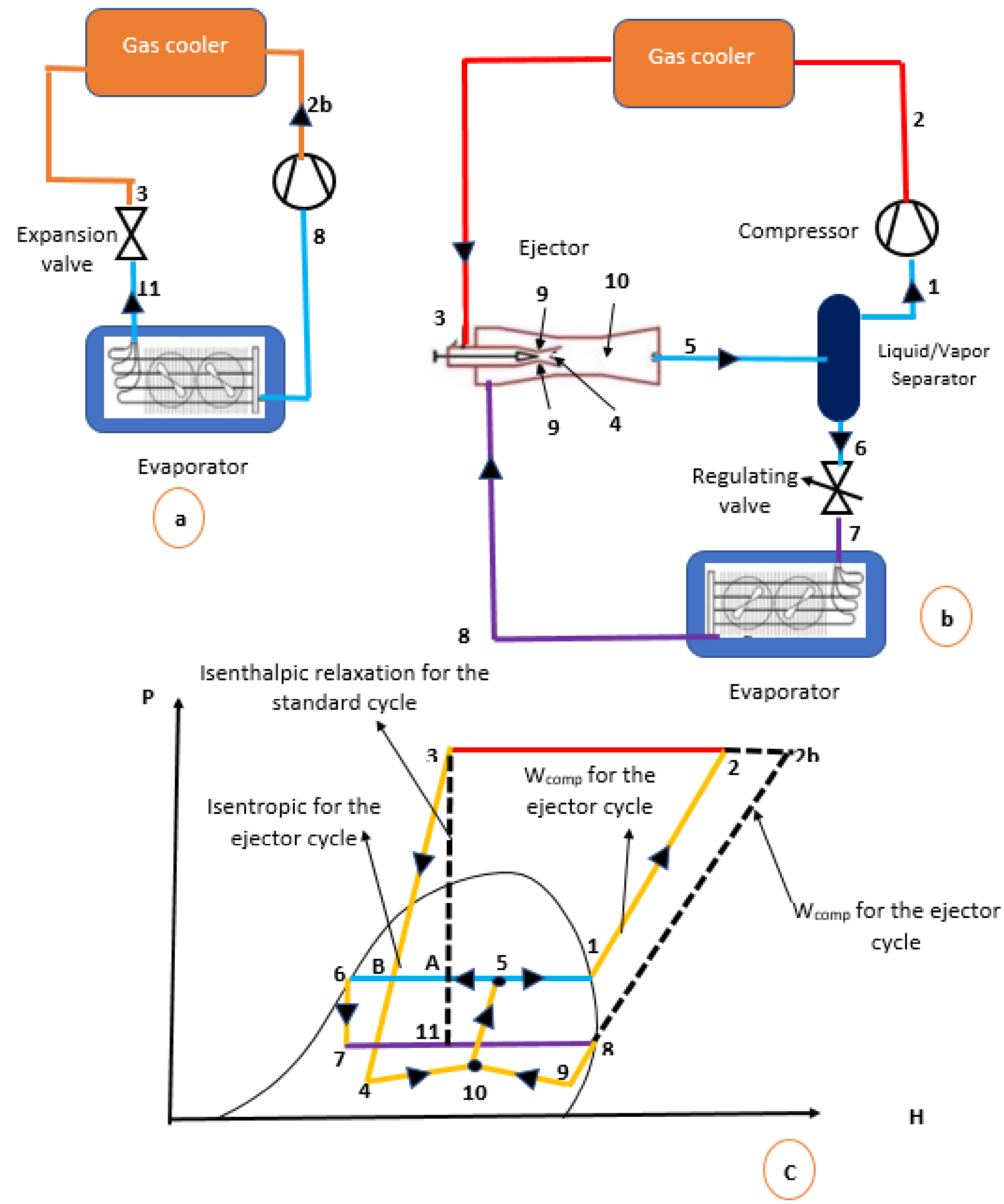

Figure-4: CO_2 transcritical refrigeration cycle (a); Two-phase ejector cycle (b); pressure - specific enthalpy diagram of the two cycles (c)[6].

Applications of carbon dioxide

Heat pumps: Heat pumps are devices that incorporate a heat transfer medium and are employed for generating seasonal comfort heat, supplying residential hot water throughout the year, heating swimming pools, generating industrial heat, and more. With the goal of reducing energy consumption and greenhouse gas emissions, the development of high-efficiency hot water heat who have made a lot of heat pumps for the residential sector using CO_2 as a refrigerant. In recent years, this type of system has progressively gained popularity in Japan [12].

Air conditioning systems in automobiles: In vehicle air conditioning, the demand is for not heavy and highly compact systems[13]. Carbon dioxide (CO_2) is increasingly being used in automotive air conditioning.

This application has gained the vehicle air conditioning market due to the need to adopt ecologically friendly technology, and significant improvements in terms of performance and CO_2 non-flammability have been recognised[14].

Commercial and industrial refrigeration: After the research conducted by Lorentzen and Pettersen[13], which explored the application of carbon dioxide (CO_2) as a refrigerant in vehicle air conditioning systems, showing that these systems can surpass conventional ones using R12 or R134a in terms of power, energy efficiency, installation cost, weight, and dimensions, other industry communications also emerged. Some writers have described air conditioning and refrigeration systems in supermarkets and buses that employ carbon dioxide as a refrigerant[14,15]. Cecchinato et al.[16] investigated transcritical carbon dioxide water coolers for commercial refrigeration with the goal of improving performances. The cycle's ideal high pressure was maintained in transcritical mode by using a flash reservoir and two electronic valves. When the gas cooler incoming air permits for subcritical working conditions, valve management improves refrigeration machine efficiency. The results of a chiller simulation model were validated using experimental data. A measuring trial was conducted, testing the chiller at outside temperatures ranging from 18 to 35°C, with the unit's energy efficiency varying from 3.1 to 2.0 [16].

Cogeneration: Novel cogeneration systems are being developed, analysed, and optimised to showcase the effectiveness of heat recovery technology, generating both heat and/or refrigeration. The operating fluid in these systems is carbon dioxide. It is a hybrid of the compression Brayton cycle and the transcritical carbon dioxide refrigeration cycle[17]. When a system cogenerates power and refrigeration or produces solely refrigeration, it is optimised. It should be observed that when the evaporation temperature falls, the system's exergy efficiency improves. Certain authors have delineated the benefits of the integrated system compared to the conventional approach[18]. They discovered that the combined system's exergy efficiency is 2.45% higher than that of the individual system at evaporation temperatures of 273.15 K and 253.15 K, respectively. To address the conflict between investment and system performance, multi-objective optimisation with exergy efficiency (to be maximised) and annual cost per heat consumption (to be minimised) is undertaken.

Studies on carbon dioxide refrigeration cycles

Effect of the intermediate exchanger in a transcritical cycle: Kim et al.[19] investigated the impact of an intermediate exchanger on the COP of a heat pump for heating water. They validated a simulation model using experimental data from their research. The sizing of the intermediate exchanger was then optimised using this model. The model indicated that elongating the interstage exchanger enhances the system's COP moderately (by 4%). Additionally, the researchers found that elongating the interstage exchanger results in a reduction in the optimal gas cooler pressure. These results corroborate the findings of Chen and Gu, who found that an increase in intercooler efficiency leads to a reduction in optimal pressure and an increase in COP[20]. Upon integrating an intermediate exchanger into a fundamental transcritical system, Robinson and Groll's simulation demonstrated a notable 7% enhancement in COP[21].

The relationship between performance and secondary fluid temperature: According to the studies of Laipradit and others, secondary fluid temperature has a substantial effect on system performance in carbon dioxide refrigeration systems. The results of their heat pump investigation show that the coefficient of performance (COP) improves with decreasing water temperature at the gas cooler inlet[22]. The reduction in water temperature at the inlet corresponds to a drop in refrigerant (CO_2) temperature at the outlet of the gas cooler. This leads to an increased temperature differential between the refrigerant's inlet and outlet at the gas cooler stage, thus accounting for the observed rise in COP. To confirm the effect of the temperature differential between the intake and exit of the CO_2 gas cooler. Cecchinato et al. conducted a theoretical and experimental investigation of a heat pump. According to the findings, the COP of the heat pump utilised to heat the water increased greatly as the water temperature at the gas cooler outlet decreased[23].

Gas cooler: Because of the importance of heat transfer mechanisms in a carbon dioxide (CO_2) refrigeration system, the design of a gas cooler requires special consideration[1,24]. To improve the performance of CO_2 refrigeration machines, the design and experimental study of a carbon dioxide transcritical chiller for commercial refrigeration has been explored[16]. Fronk and Garimella highlighted the significance of the ratio of CO_2 transfer coefficients and secondary fluid in 2011. Air has a lower transfer coefficient than CO_2, whereas water has a greater transfer coefficient than CO_2[25]. The overall heat transfer coefficient in a water-cooled heat exchanger is more sensitive to the CO_2 transfer coefficient, whereas the overall heat transfer coefficient in an air-cooled heat exchanger is more susceptible to the air side heat transfer coefficient. Enhancing the performance of the gas cooler occurs when minimizing the temperature difference between CO_2 and the secondary fluid at all points within the gas cooler. Sarkar et al. delved into the irreversibility in the gas cooler and determined that temperature discrepancies between the refrigerant and the secondary fluid contribute to approximately 90% of heat exchanger losses[26]. Conversely, the losses attributed to pressure drop across the gas cooler were relatively insignificant. The minimal difference (dtmin) in a counter-current gas cooler represents the distinction between the CO_2 exit temperature and the inlet temperature of the secondary fluid. Fronk and Garimella have shown that as dtmin falls, so does the optimum pressure of the gas cooler[25]. As a result, the compressor's workload is reduced. Losses generated by temperature differences can be mitigated by expanding the heat transfer area, although this method has a limit to its efficiency. Sarkar et al.[27] performed a theoretical optimisation of the geometry of a concentric tube type heat exchanger. In 2015, Dai and his colleagues developed a model to evaluate the gas cooler's performance under different oil concentrations[28].

Evaporator: In 2007, Yun et al.[29] employed a numerical model to conduct a comparison between a microchannel evaporator designed for air conditioning and a conventional evaporator. The validation of the model was based on results from a system operating with R134 a and insights from Beaver et al.[30]. In contrast to a conventional heat exchanger, the micro-channel evaporator exhibited a higher heat transfer capacity (33%). Similarly, Kim and Bullard formulated a model to analyze the performance of a microchannel evaporator. Their model incorporated established correlations for pressure drop and heat transfer coefficients. This particular model proves useful in the design of compact microchannel evaporators [31]. Jin et al. created a new correlation-based model that was designed exclusively for CO_2. The model has been experimentally tested and is applicable for high or even superheated steam systems[31]. Because of its high operating pressure and high vapour density, CO_2 can then be employed in micro-channel evaporators to decrease the problem of phase misdistribution.

Expansion valve: Capillary expansion valve: Agrawal and Bhattacharyya undertook a series of numerical studies focused on the flow characteristics of carbon dioxide (CO_2) within adiabatic capillary tubes, as well as an evaluation of the performance of transcritical systems including the capillary expander[32]. In this study, the capillary expander was utilized within a system engineered for concurrent heating and cooling purposes.

The system's coefficient of performance (COP) was juxtaposed with a system explored by Sarkar et al.[33], which incorporated an adjustable regulator. The length of the capillary has a major impact on the pressure of the gas cooler and the system's COP. A system has been equipped with an optimum capillary and tested for several gas cooler outlet temperatures. The COPs with the expansion valve and the capillary tube were almost the same. However, the refrigerating capacity of the capillary expansion system exhibited a marginal improvement.

Adiabatic expansion: The operation of CO_2 refrigeration systems is influenced by adiabatic expansion. Madsen and colleagues investigated the COP of a transcritical CO_2 cycle with capillary expansion. A capillary tube was compared to a fixed-pressure and adjustable expansion valve [34]. An adiabatic expansion model was used in the simulation. The capillary tube system had a greater COP than the fixed-pressure regulator, but the adjustable regulator performed the best.

Non-adiabatic capillary expansion: During expansion of CO_2 in a non-adiabatic capillary tube, the refrigerant rejects heat to the environment. This heat is useful for superheating steam.

By welding a segment of the capillary to the refrigerant line between the evaporator outlet and the compressor inlet, the capillary tube can be utilised as an internal heat exchanger. This mechanism supplies superheated vapour to the compressor akin to the function of an intermediate exchanger. Chen and Gu investigated the heat exchange and flow characteristics of a capillary tube within a transcritical cycle, employing a non-adiabatic flow model[20]. They determined that at a given diameter and internal pressure, the cooling capacity of the system diminishes as capillary length grows.

Expansion turbines: The goal of using an expansion turbine in a transcritical cycle is to increase the cooling effect on one hand because the expansion process is no longer isenthalpic, and on the other hand to reduce the energy consumed by the compressor through the recovery of the turbine's work[8]. Chegnimonhan and his colleagues tested the performance of four refrigeration systems in the Cotonou (Benin) weather conditions. The results revealed that, when compared to the other cycles tested, refrigeration systems using expansion turbines operate well in the climatic circumstances considered. The inclusion of a heat exchanger in the turbine system lowered cycle performance by 85.6%[8]. Yang and colleagues demonstrated that utilising an expander instead of a conventional throttle device reduces the gas cooler's optimal pressure and increases the COP in cooling mode by 33% [35]. Huff and Radermacher's theoretical research allowed them to compare the performance of a CO_2 system with a turbine and an R22 system with a throttle valve. It was found that the transcritical system outperforms the R22 system at low ambient temperatures. Experimental and theoretical study has been conducted on a wide range of turbines, including scroll, piston, rolling piston, screw, and vane turbines[36].

Ejector expansion valve: In the field of CO_2 refrigeration, ejector expansion valves have also been researched. The employment of an ejector as an expansion element decreases expansion losses while increasing compressor inlet pressure and thereby lowering compressor effort[37]. Because of the ejector cycle's superior performance compared to the basic cycle and its lower cost compared to the turbine cycle, research has focused on this component. Sarkar examined the implementation of an ejector within a transcritical cycle for the purpose of integrated heating and cooling[38].

The author modelled two systems: an ejector cycle and Li and Groll's modified ejector cycle[39]. Except at low evaporator temperatures, where the modified cycle performed somewhat worse, the difference in performance between the two cycles was minimal.

The outcomes demonstrate that both ejector cycles lowered the optimal discharge pressure in comparison to the expansion valve cycle. As per the authors' analysis, both ejector cycles exhibited higher COP values than the conventional cycle, with the turbine system yielding the most exceptional performance. Elbel and Hrnjak conducted a study on a prototype of a transcritical system incorporating an ejector, and the conclusions were employed to validate a mathematical model. The implementation of the ejector led to a 7% increase in COP[40].The table below summarises the most recent studies on the transcritical CO_2 ejector.

Table-1: Latest advanced research on the transcritical CO_2 ejector.

Type of cycles	Study method	Conclusion	References
CO_2 transcritical cycle	Advancedexergo-economic analysis	Environmental impact and exergy cost are low	41
CO_2 refrigeration cycle two-stagecompression transcritical with one ejector and two evaporators	Theoretical analysis	In contrast to the conventional system, the new system exhibited significant enhancements in COP and exergy efficiency, registering improvements of 19.6% and 15.9% respectively. Notably, the high-temperature compressor discharge temperature experienced a reduction of 10.5°C.	42
Hybrid transcritical vapor compression systemCO_2 and ejector refrigeration	Theoretical analysis	At the evaporation temperature of 12.5°C, the R32-CO2 hybrid system showcased remarkable advancements, with nearly a 50% increase in refrigeration capacity and a 45% improvement in COP.	43
Two-stage CO_2 refrigeration cycle equipped with an ejector	Thermoeconomic analysis	The systemis efficient and compact. The average annual COP increases.	44
CO_2 refrigeration system transcritical equipped with an ejector	Exergy analysis and experimental study	Ejector provides exergy efficiency maximum of 23%.	45
Transcritical CO_2 ejector refrigeration system equipped with a thermoelectric subcooler	Exergy analysis	The compressor efficiency climbs from 0.5 to 0.9, while the preventable endogenous exergy destruction of the relevant system components rises from 93.61% to 82.33%.	46
CO_2refrigeration cycle transcritical with two evaporators and two ejectors	Energy and exergy analysis	Across all provided conditions, both COP and exergy efficiency witness improvements ranging from 15.9% to 27.1% and from 15.5% to 27.5%, respectively.	47
Two-stage CO_2 refrigeration cycle with ejector	Thermodynamic analysis	The COP increases up to 13% under typical ejector working conditions.	48
The CO_2 cycle modified transcritical with ejector	Thermodynamic analysis	During operation, the ejector and compressor incur the highest energy destruction. However, the ejector contributes to decreasing the energy destruction rate of the entire cycle. Simultaneously, the ejector constitutes the primary origin of entropy production.	49

The compression: Intermediate cooling: In a transcritical CO_2 system, intermediate cooling (Figure-5) between two stages increases system performance. Cecchinato et al[50] investigated the performance of a two-stage system with intercooling in order to increase the performance of a CO_2 refrigeration system. The two-stage cycle enhanced cooling performance by 9% when compared to a single-stage compression system. Because of the greater temperatures in the transcritical cycle, the CO_2 can reject heat to the surrounding air via a heat exchanger for intercooling, according to the authors. Cavallini et al. investigated the performance of a two-stage experimental system at various intercooler temperatures. Under optimal discharge pressures, the performance achieved was 2.1 for temperatures of 20.5°C and 21.5°C, and 1.8 for a temperature of 33°C. Additionally, a simulation model was constructed to investigate three distinct modifications to the two-stage system. The inclusion of an intermediate heat exchanger resulted in a performance enhancement of 7.6%. The dual trigger increased performance by 21.1%. The combination of the two improvements improves performance by 24% above the baseline system performance. Another method of intermediate cooling is to use a flash tank (Figure-6), which is one of the various ways of cooling the refrigerant between compression stages by mixing the expanded vapour in the tank and lowering the intermediate temperature of CO_2. Unlike other intercooling approaches, Agrawal and Bhattacharyya discovered that two-stage compression with intercooling in a flash tank reduced COP compared to an analogue single-stage system[51]. This is mainly due to the fact that splitting the compression ratio reduces the energy consumption of the compressors. This is mostly because splitting the compression ratio minimises compressor energy usage.

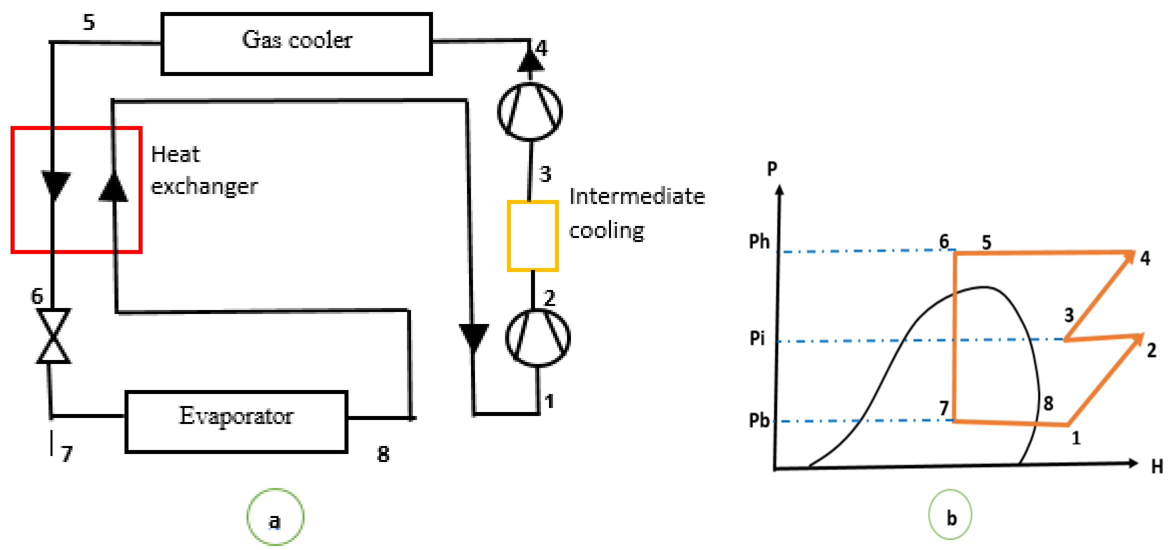

Figure-5: Two-stage system with intercooling.

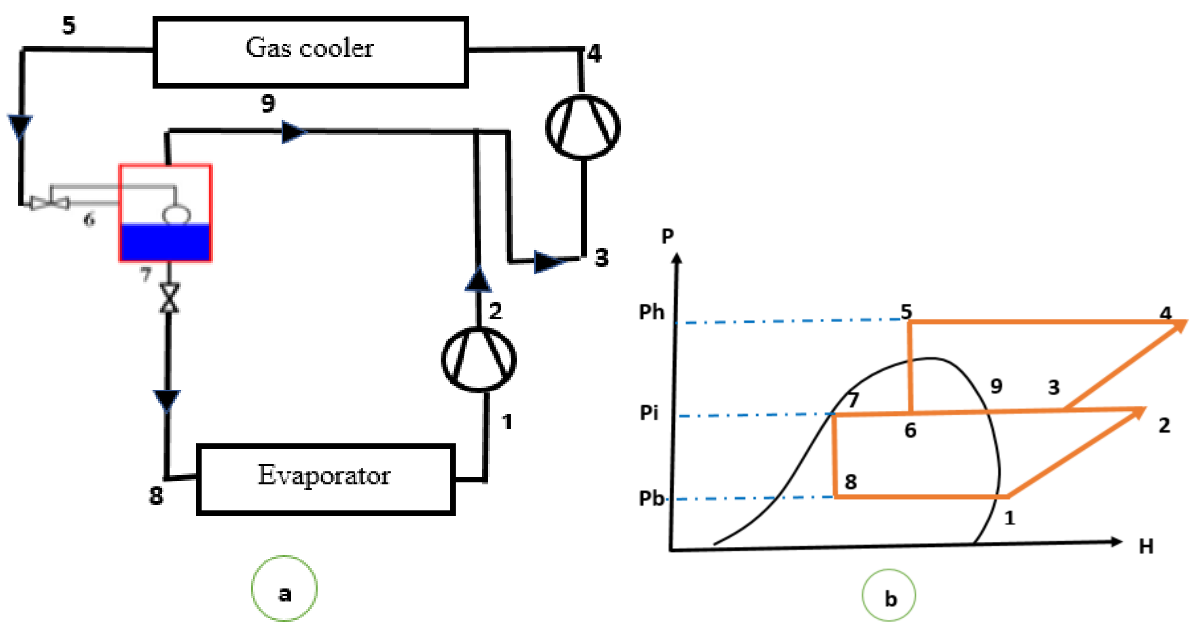

Figure-6: Schematic diagram of flash tank system.

Introduction of an economizer: The economizer is a sort of subcooler that evaporates a portion of the refrigerant, often 10-20%, at a greater evaporation temperature than the main evaporator while drastically subcooling the balance of the refrigerant flow[52]. The installation of an economizer port is known to considerably boost the capacity of screw compressor-based low temperature refrigeration installations. The use of two-stage cycles allows for the use of more particular and regulated subcooling systems, such as the economizer cycle or the two-stage cycle with subcooler, as shown in Figure 7. The refrigerant leaving the gas cooler is split into two streams in this design. The auxiliary flow is throttled to intermediate pressure and evaporated within the economizer or subcooler, allowing the primary refrigerant vapour to be subcooled. According to Torrella et al.[53], the COP increase that the savings allow is dependent on the thermal efficiency of the subcooler or closed flash tank separator, and the improvement for 100% efficiency reaches the performance of the open flash separator at intermediate pressure. By adding an economizer to an experimental two-stage system, Cho et al. increased performance by 16.5%[54]. Cecchinato et al. theoretically analysed a similar system, resulting in a 16.8 to 28.7% performance gain over a single-stage system[50]. Sarkar and Agrawal[55] investigated a parallel compression system with an economizer. When compared to the standard transcritical system, the performance is improved by 47.3%.

Several research on an experimental system with an economizer and a single-stage compressor have been conducted[56,57]. Single-stage compression cycles with and without an economizer were compared. Economizer cycles improve heating capacity under all working conditions, but only at low compressor speeds and low evaporation temperatures. The installation of an economizer port is known to considerably boost the capacity of screw compressor-based low temperature installations. Cecchinato et al.[50] investigated a transcritical system that differed from earlier investigations. The system is divided into two loops, each with its own compressor. Only at low intermediate pressures did the cooling COP outperform the more complex two-stage cycles.

Conclusion

This article examines the most recent advancements in CO_2 refrigeration system technology (transcritical and subcritical) as well as the various applications of CO_2 refrigeration machines. CO_2 performs admirably as a natural refrigerant. Because carbon dioxide has a low critical temperature (31°C), it operates on a transcritical cycle, evaporating in the subcritical region and rejecting heat above the critical point into a gas cooler rather than a condenser when the high-pressure heat exchanger is air-cooled at temperatures above 21°C. When compared to typical cycles, this design has poor performance. However, transcritical cycle performance can be enhanced by combining cold production with heat energy recovery at the compressor output for hot water needs, as is common in the food industry where cold and hot needs are frequently simultaneous. A cascade system of two natural refrigerants (NH_3 / CO_2) is an advantageous method for obtaining low temperatures ranging from -30°C to -100°C. When working temperatures are very low, this technology not only ensures higher environmental safety and good product packaging with investment and competitive operation. Cascades also have the advantage of being small and sturdy systems in general.

Perspectives in this area include: i. Because CO_2 refrigeration systems are complex, experimental research with integrated mechanical subcooling systems and economizer cycles is required, as existing research has not yet reached the limits of improvement. Furthermore, dedicated subcooling systems should be investigated experimentally, where the employment of refrigerant mixtures in the auxiliary refrigeration cycle may increase performance even further. Heat recovery systems that are integrated into the refrigeration cycle, as well as those that use phase change materials, should be considered. The ejector is a hot research topic these days, but there are many issues, such as its geometry and operating conditions. In addition, simulation software is necessary to visualize the internal flow of two-phase flows, in particular when there seems to be a lack of investigation of the physical phenomena linked to the presence of the nozzle at the inlet of the refrigerant fluid; ii. The combination of CO_2 refrigeration cycles with heat recovery systems for subcooling such as absorption refrigeration systems or adsorption systems is almost non-existent. Published theoretical work indicates that the combination of these systems would be very positive, however, experimental evaluation of its combination is a mechanical challenge. iii. To optimise the required cooling power, the evaporation phenomenon should be considered. Heat transfer should be given special consideration in order to improve heat transfer correlations tailored to carbon dioxide. iv. The survey carried out on the refrigeration systems installed in Benin reveals that the transcritical CO_2 systems are not yet introduced in Benin and do not seem to be in neighboring countries. Moreover, no studies appear on these transcritical systems in the literature in African contexts. In a future perspective, this study deserves to be pursued by developing experimental prototypes operating in the demanding climatic conditions of West Africa. The analysis of the results will enable the design of the CO_2 refrigeration installations adapted to tropical regions.

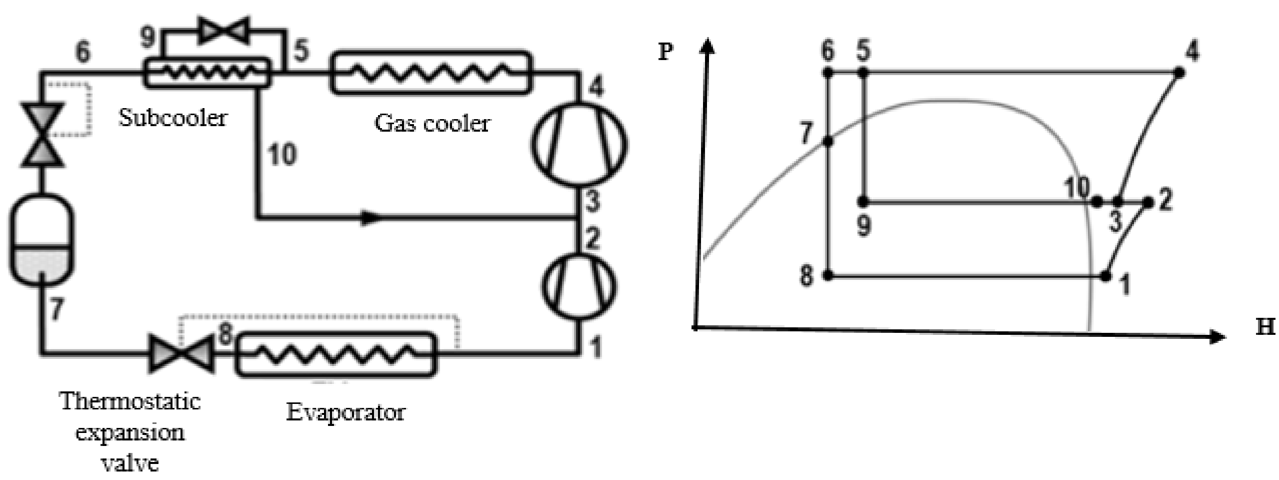

Figure-7: The economizer cycle[52].

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