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Prospective study of single-stage carbon dioxide refrigeration cycles in a tropical climate.

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Abstract: This numerical investigation presents the comparative study of four refrigeration systems in the meteorologicalconditions of Cotonou (Benin). The systems were modelled and the physical phenomena were simulated under EES(Engineering Equation Solver). The results revealed that the refrigeration systems with expansion turbine present betterperformances under the considered climatic conditions compared to the standard cycles equipped with thermostaticexpansion devices. The use of an additional internal heat exchanger in the turbine system significantly reduces theperformance of the cycle by 85.6%. But the implantation of such heat exchanger in the basic transcritical cycle increases theperformance by 73%. Correlations were developed to predict the maximum performance of the refrigeration cycles. Thatparameter is very sensitive to the variation of the end-of-cooling temperature ranging from 35°C to 50°C. Moreover thetemperature obtained at the discharge of each refrigeration cycle varies according to the evaporation temperature. Someappropriate, experience approved operating ranges have been proposed to limit excessive overheating and to allow goodlubrication of the compressors. One solution for better use of the systems in tropical environments may be heat recovery toproduce domestic hot water or even steam for various applications as cogeneration.

Keywords: Refrigeration, transcritical cycle, carbon dioxide, tropical countries.

I. INTRODUCTION

Carbon dioxide (CO_2 or R744), is a natural refrigerant that is non-flammable, non-toxic, inexpensive and environmentally friendly, and was among the first fluids used in the history of refrigeration. Present in the atmosphere at a concentration of 0.04% by volume, CO_2 has been known since the beginning of humanity [1]. Its success was curtailed from the mid-20th century onwards by the development of synthetic refrigerants used in less robust and more efficient refrigeration systems [2]. As the technology of CO_2 refrigeration systems was no longer developed, the improvement and modernisation of existing systems was stopped as soon as new fluids with interesting high thermodynamic performances arrived. Although welcomed by the refrigeration industry, these fluids have undesirable effects on the environment [3]. Most of the first generation hydrofluorocarbons (HFCs) contribute to global warming. Hydrocarbons are highly flammable, high-performance refrigerants. The family of olefins (HFOs) which have low global warming powers are, however, dangerous because of their flammability [4]. The emergence of environmental impact regulations and global environmental awareness has led to a search for fluids that have the least possible impact on nature. CO_2, which had previously been abandoned, was proposed in 1990 by Professor Gustav Lorentzen as an alternative refrigerant [5]. It slowly came to the attention of the refrigeration industry again when the environmental debate started. Thus, several researches converged on this refrigerant [6]-[8]. Despite the high level of technology required for this natural fluid, it is very promising in refrigeration cycles, due to its low global warming potential: its GWP of 1 being the standard for measuring the greenhouse effect of all other refrigerants, as well as its zero ozone depletion potential (ODP: 0) [9]. CO_2 is also valued for its flame suppressant effect, hence its use as a fire extinguishing agent. There is little investigation in the literature on the use of carbon dioxide in West Africa and it does not appear to be used in West Africa as a refrigerant. Thus the basic aim of this study is to investigate the performance of eligible CO_2 single-stage refrigeration systems in tropical climatic regions.

II. MATERIALS AND METHODS

A. Description of single-stage transcritical refrigeration cycles

Figure 1 shows four configurations of single-stage refrigeration systems operating with carbon dioxide. The basic transcritical cycle is shown in Figure 1 (Cycle 1). In transcritical cycles, the operating pressures of CO_2 are very high. The low pressure is around 25 bar depending on the operating temperature for single stage cycles and the high pressure is around 100 bar. During heat transfer from the fluid to the outside, the pressure is above the pressure of the critical point (73.8 bar). There is therefore no phase change at the high pressure heat release exchanger of the cycle. In contrast to the classical thermodynamic cycle, the heat transfer in the high-pressure part is no longer condensation but gas cooling. That's why that exchanger is not called "condenser" but "gas cooler" [5]. In the first cycle, the compressor draws in the cold refrigerant vapours (point 1) at low pressure and low temperature, compresses them and delivers them at high pressure and high temperature to the gas cooler (point 2). At point 3, the cooled gases are fed to the expansion valve. The latter drops the temperature and pressure in an isenthalpic process (point 4). The refrigerant vapours are then drawn off by the compressor and the cycle starts again. Cycle 2 is a modification of the first cycle by introducing an intermediate exchanger into the cycle. This exchanger is used to cool the fluid leaving the gas cooler (point 3) before it goes to the expansion stage (point 4), which decreases the value of the vapour content and therefore increases the refrigerating effect, and at the same time serves to superheat the fluid leaving the evaporator (point 6) before it enters the compressor to protect it (point 1). Cycle 3 is obtained by replacing the expansion valve (point 3- point 4) of cycle 1 with an expansion turbine. Its purpose is to increase the refrigerating effect on the one hand because the expansion process is no longer isenthalpic and on the other hand to reduce the energy consumed by the compressor by recovering the work generated by the turbine. Cycle 4 is obtained by inserting an internal heat exchanger in cycle 3.

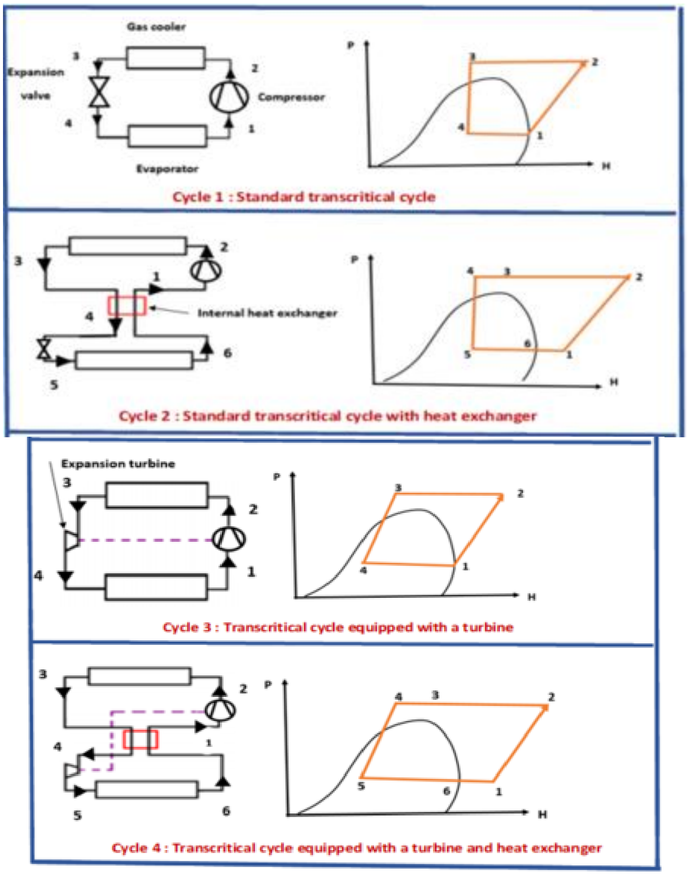


Figure 1: Single-stage transcritical refrigeration cycles using CO_2

B. Carbon dioxide applications

Heat Pumps

Heat pumps are installations containing a heat transfer fluid for seasonal comfort heat production, for year-round hot water production in residential areas, for heating swimming pools, for industrial heat production, etc. According to the fifteenth information note of the International Institute of Refrigeration, the use of carbon dioxide in heat pumps to produce water at 90°C can be a very interesting prospect. With the aim of reducing energy consumption and greenhouse gas emissions, the development of high-efficiency hot water heat pumps using natural refrigerants has recently received a lot of attention from manufacturers (Denso, Sanyo, Panasonic, Daikin, Hitachi, Mitsubishi Electric, Sanden, Corona, Toshiba Carrier and Matsushita Electric, etc.) who have manufactured many heat pumps using CO_2 as a refrigerant for the residential sector. This type of system has gradually gained popularity in Japan in recent years [10].

Automotive air conditioning

In automotive air conditioning, there is a need for lightweight and ultra-compact systems [11]. The use of carbon dioxide in automotive air conditioning is currently booming. Driven by the desire to use environmentally friendly technology, this application in vehicles is very recent and significant benefits are noted [12]. Micheletto and Rosso presented a study on low-temperature bodies using a two-stage transcritical cycle running on R744 [13]. The different steps that allowed to reach much higher performances (i.e. 31.5% increase in COP) than those reached with common systems running on R507A are detailed. Experimental studies by KrUSe and colleagues have shown that the coefficient of performance (COP) of a prototype automotive air conditioning system using CO_2 compares favourably with a conventional system using R12. This performance would result in a lower Total Equivalent Warming Impact (TEWI) [14].

Commercial and industrial refrigeration

Following the work of Lorentzen and Pettersen on the use of carbon dioxide as a refrigerant in automotive air conditioning systems, which showed that such systems can compare favourably with traditional systems using R12 or R134a in terms of capacity, energy, installation cost, weight and size, further papers from industry have been investigated [11]. Several authors have presented various air conditioning and refrigeration systems using carbon dioxide as a refrigerant in supermarkets, buses, etc. [15]-[21]. The design and experimental analysis of a transcritical carbon dioxide chiller for commercial refrigeration has been presented by Cecchinato and co-workers to increase the performance of refrigeration machines operating with transcritical CO_2 [22].

Cogeneration

To underline the success of heat recovery technology, new cogeneration systems producing heat and/or refrigeration are proposed, analysed and optimised. These systems use carbon dioxide as the working fluid. It is a combination of the Brayton compression cycle (elementary gas turbine cycle) and the transcritical carbon dioxide refrigeration cycle with an expansion valve [23]. Optimisation is performed for the system when it cogenerates energy and refrigeration or produces only refrigeration. The lowest evaporating temperature is achieved when the system is optimised for maximum exergy efficiency. Lig and Wang described the advantage of the combined system over the conventional system [24]. At evaporation temperatures of 273.15 K and 253.15 K respectively, they showed that the exergy efficiency of the combined system is 2.45% and 5.87% higher than that of the separate system. In order to balance the contradiction between investment and system performance, the multi-objective optimisation is performed with exergy efficiency (to be maximised) and the annual cost per heat consumption (to be minimised).

C. Modelling assumptions for transcritical refrigeration cycles

In order to analyse the performance of the refrigeration systems' operating cycle, the following assumptions were made. These refrigeration requirements and assumptions correspond to those of a shopping centre in Benin:

The end of cooling temperature at the gas cooler: $35°C \leq T_{fr} \leq 50°C$.

Evaporation temperature: $-20°C \leq T_{ev} \leq 10°C$.

The high pressure (discharge pressure): $75 \text{ bar} \leq P_h \leq 140 \text{ bar}$.

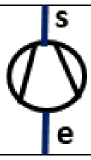
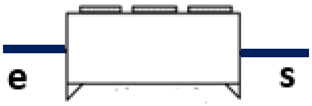
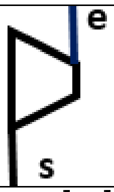
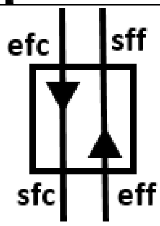
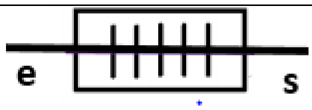
Isentropic efficiency of the turbine: $\eta_{tr}=0.7$.

The thermal efficiency of the internal exchanger: $\xi=0.8$.

D. Simulation models

The thermodynamic models applied to the refrigeration cycles are summarised in Table 1. For each component of the system, the mass and energy balance equations are applied. Each process is represented mathematically and implemented in EES (Engineering Equation Solver). In Table 1, $\dot{Q}_{ev}$ denotes the cooling capacity, W_{tot}: the work supplied by the compressor by unit time, W_{tr}: the work supplied by the turbine, $\dot{m}$: mass flow rate of carbon dioxide, h_e and h_s: enthalpy of the refrigerant at the inlet and outlet of the system components, η_{cp}: compressor efficiency, η_{tr}: isentropic efficiency of the turbine, p_e and p_s: pressure of the refrigerant at the inlet and outlet of the compressor, T_{efc}: temperature at the inlet of the hot fluid, T_{sfc}: temperature at the outlet of the hot fluid, T_{eff}: temperature at the inlet of the cold fluid and T_{sff}: temperature at the outlet of the cold fluid, e and s: inlet and outlet of each system component.

Table 1: Energy relations of the components of transcritical cycles.

Components	Schemes	Energetic properties
Compressor (cp)		$\eta_{cp} = 1{,}003 - 0{,}121(\frac{P_s}{P_e})$ [25] $\eta_{cp} = \frac{hsis - he}{hs - he}$, efficiency $W_{cp} = \dot{m}\,(h_s - h_e)$, compression power
Gas cooler		$\dot{m}_e = \dot{m}_s$, mass flow
Expansion valve		$h_s = h_e$, enthalpy
Expansion turbine (tr)		$\eta_{tr} = \frac{hs - he}{hsis - he}$, efficiency
Internal heat exchanger		$\dot{m}(h_{sff} - h_{eff}) = \dot{m}\,(h_{efc} - h_{sfc})$ $\xi = \frac{Tsff - Teff}{Tefc - Teff}$, efficiency efc: hot fluid inlet, sfc: hot fluid outlet eff: cold fluid inlet, sff: cold fluid outlet
Evaporator (ev)		$\dot{Q}_{ev} = \dot{m}\,(h_s - h_e)$
Coefficient of performance: $COP = \frac{\dot{Q}ev}{Wtot}$. For cycles with turbine: $W_{tot} = W_{cp} - W_{tr}$		

III. RESULTS AND DISCUSSION

A. Temperature in southern Benin (Cotonou)

Statistically, over the course of the year, the temperature generally varies between 24 °C and 32 °C and rarely exceeds this. Figure 2 shows the average annual temperature in southern Benin. These data were collected using Meteonorm 7 software.

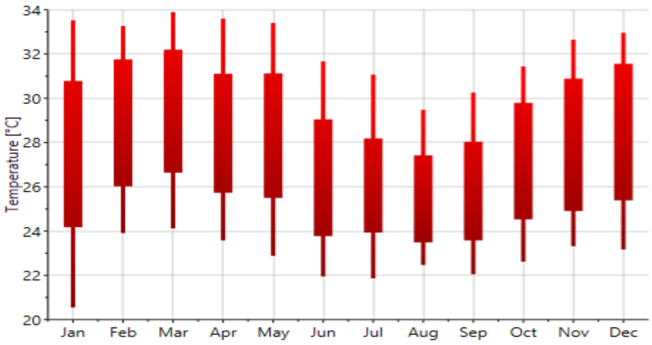


Figure 2: Average annual temperature in southern Benin (Cotonou)

B. Validation model

The coefficient of performance is used to evaluate the efficiency of refrigeration systems. The COP depends on several parameters such as the evaporation temperature, the condensation temperature, the refrigeration capacity, the compressor power input etc. The results of the model developed in this work are compared with those of Beheta et al. [26]. The low pressure and high pressure are kept fixed (40 bar and 100 bar). Figure 3 shows the variation of the COP as a function of the temperature at the outlet of the gas cooler. The correlation is almost linear. The highest COP value was 3.915 for a temperature of 35°C. The COP increases as this temperature decreases. The COP becomes lower than 1 when the temperature exceeds 50°C. The COP value is 0.55 for a temperature of 55°C. The relative errors are small and vary from 0.15 to 12.67% (Table 2). The differences between the two models may be due to differences in modelling assumptions. In view of the small deviations, the model developed in this paper can be considered as validated. However, it should be noted that the validation is only presented for the basic transcritical cycle (cycle 1).

Table 2: Comparison of the model results with data from Beheta et al. [26].

Gas cooler outlet temperature (°C)	COP (this study)	COP [26]	Relative Error (%)
35	3.915	3.944	0.73
40	3.252	3.247	0.15
45	2.243	2.167	3.50
50	1.226	1.148	6.79
55	0.551	0.489	12.67

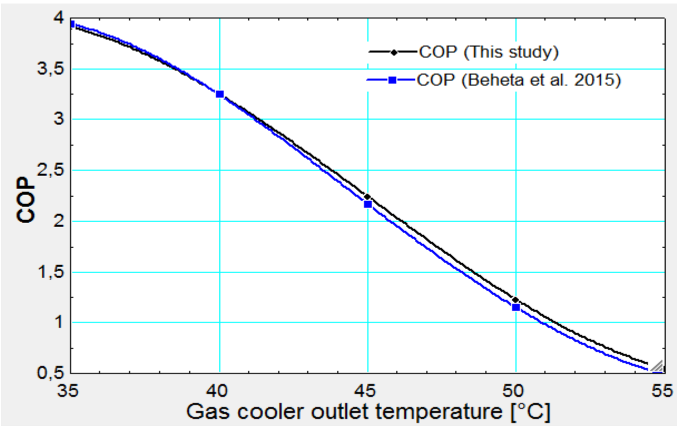


Figure 3: COP variation as a function of gas cooler outlet temperature.

C. Variation of single-stage cycle performance as a function of high pressure

With the evaporation temperature and the gas end-cooling temperature held fixed at 0°C and 35°C, Figure 4 shows the variation in performance of the single-stage cycles (cycle 1, cycle 2, cycle 3 and cycle 4) as a function of high pressure ranging from 75 bar to 130 bar. For all the single-stage cycles studied the performance coefficients increase rapidly for pressures between 80 and 95 bar before increasing for higher pressures. At higher pressures, the isotherms approach the vertical and a further increase in discharge pressure will have almost no effect on the refrigeration output, although it will result in an increase in power input.

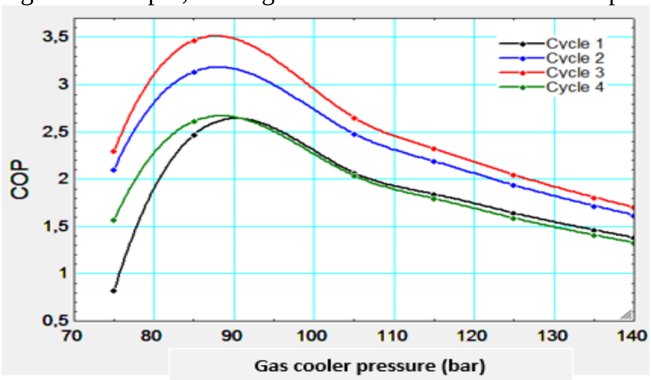


Figure 4: Variation of COP with gas cooler pressure.

Polynomial regression analysis was used to develop correlations to write the maximum coefficient of performance (COP_{max}) as a function of the end of cooling temperature (T_{fr}). Table 3 shows the coefficients of the correlations developed for each cycle. To assess the quality of the regression analysis the coefficient of determination R^2 was used. The coefficients of determination of the four correlations are the same (99.99%). These correlations are valid for end-of-cooling temperatures of 35 to 50°C.

$COP_{max} = a + b.T_{fr} + c.T_{fr}^2 + d.T_{fr}^3 + e.T_{fr}^4 + f.T_{fr}^5 + g.T_{fr}^6$

With temperatures in °C and pressures in bar.

Table 3: Coefficients of the developed correlations.

Coefficients	Cycle 1	Cycle 2	Cycle 3	Cycle 4
a	$-3.03897161.10^{4}$	$-2.36958204.\ 10^{4}$	$-2.49216389.\ 10^{04}$	$-2.07404866.\ 10^{4}$
b	$4.25881481.10^{3}$	$3.32182559.\ 10^{3}$	$3.48077785.\ 10^{03}$	$2.90038352.\ 10^{3}$
c	$-2.47668496.\ 10^{2}$	$-1.93229088.\ 10^{2}$	$-2.01769309.10^{02}$	$-1.68320011.\ 10^{-2}$
d	7.65339933	5.97255494	6.21614386	5.19117824
e	$-1.32581510.\ 10^{-1}$	$-1.03486207.\ 10^{-1}$	$-1.07377820.\ 10^{-1}$	$-8.97617358.\ 10^{-2}$
f	$1.22102139.\ 10^{-3}$	$9.53252279.\ 10^{-4}$	$9.86270793.\ 10^{-4}$	$8.25232285.\ 10^{-4}$
g	$-4.67131844.\ 10^{-6}$	$-3.64755420.\ 10^{-6}$	$-3.76378167.\ 10^{-6}$	$-3.15196950.\ 10^{-6}$

D. Variation of the discharge temperature of refrigeration systems

Figure 5 shows the effect of evaporating temperature on the discharge temperature of refrigeration systems. As the gas cooler pressure is fixed (85 bar), all discharge temperatures are high compared to discharge temperatures of standard refrigeration systems. Discharge temperatures are typically limited to values below 100°C to avoid problems of poor lubrication and excessive material overheating. Thus, it can be deduced that under the climatic conditions of this study, cycles 1 and 3 are technologically operational only for evaporation temperatures of -10°C or higher. The range is reduced to values above 5°C for cycles 2 and 4. However, the high discharge temperatures observed indicate great potential for energy recovery for domestic hot water production or even cogeneration. This makes it possible to extend the operating ranges together.

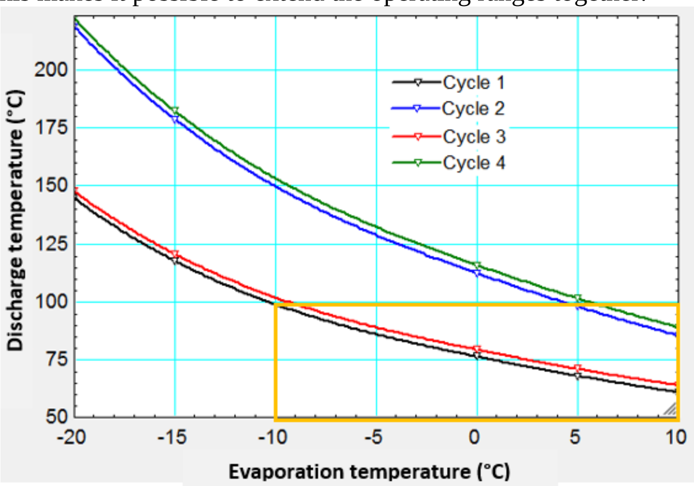


Figure 5: Variation of the discharge temperature of the studied refrigeration systems.

IV. CONCLUSION

The CO_2 transcritical refrigeration cycle models were simulated thermodynamically. The refrigeration cycle with expansion turbine had the highest COP (3.47). The latter depended mainly on the gas cooler outlet temperature and the evaporator temperature. To compute the maximum performance coefficient, correlations were developed for the studied cycles. The maximum performance is sensitive to the variation of the end-of-cooling temperature between 35 and 50°C. The temperatures obtained at the discharge of each cycle are quite high compared to other conventional refrigeration cycles. Heat recovery must be considered to reduce high discharge temperatures for applications in warm tropical areas.

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