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RESEARCH ARTICLE

INVESTIGATING THE PERFORMANCE OF A TRANSCRITICAL BOOSTER REFRIGERATION SYSTEM WITHCARBON DIOXIDE IN TROPICAL CLIMATES: THE CASE OF BENIN

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Abstract

This article presents a prospective study of low carbon footprint refrigeration systems in tropical climates. A transcritical carbon dioxide booster refrigeration system is simulated within the Engineering Equation Solver (EES) environment. The results are discussed under the weather conditions Cotonou, Benin. The performance of the transcritical booster cycle increases with low ambient temperatures; the Coefficient of Performance (COP) increased by 94.2% when the ambient temperature went from 40°C to 25°C). The gas cooler pressure and the medium pressure were ctoptimised and two correlations were developed to predict the optimal gas cooler pressure and intermediate pressure for which the COP value is maximum. It should be noted that these pressures are very sensitive to changes in ambient temperature and they vary proportionally with the temperature at the end of cooling and in a linear fashion.

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Introduction:-

Refrigerants, which enabled and supported the development of refrigeration and conditioning from the 1930s onwards, were synthetic products [1]. Chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) deplete the stratospheric ozone layer, their substitution by hydroflorocarbons (HFCs) is an effective approach to avoid further ozone depletion, but this generation of HFCs contributes to global warming [1]. Hydrocarbons (HC) are organic fluids with good thermodynamic properties, but are dangerous because of their flammability. Specialists in refrigeration have always refrained from using these fluids, although HC have recently reappeared in in cooling systems and insulating foams. Hydrofluoroolefins (HFOs) are new, environmentally friendly synthetic refrigerants that are becoming more widely available, but their slight flammability should be taken into account for applications with high refrigerant requirements. The environmental constraints (destruction of the ozone layer and global warming) of synthetic fluids have led to renewed interest in natural refrigerants. Among natural refrigerants, CO_2 is the most promising fluid one, due to its low global warming potential (GWP) and zero ozone depletion potential (ODP) [2]. Carbon dioxidehas become an increasingly relevant argument for reducing the carbon footprint. It does not need to be recovered or recycled compared to HFCs. Thus, the use of CO_2 isvery attractive where infrastructure is non-existent or too expensive, such as in Benin, a developing country. The high volumetric power generated by CO_2, due to thehigh working pressures required, allows the use of small components (small displacement for the compressor) and small diameter pipes [3]. This characteristic has made it possible to develop particularly compact small-diameter tube heat exchangers. This fluid can also be used in small installations such as automotive air conditioning systems. It has a limited pressure drop due to its low viscosity. CO_2 isnon-flammable, non-explosive and non-toxic in low concentrations. It has a low compression ratio, which then leads to an increase in the

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compressor's volumetric efficiency. This fluid operates in transcritical type cycles when the high-pressure heat rejection exchanger is cooled by external air hotter than 21°C. The listed properties make CO_2 an outstanding refrigerant in the refrigeration industry and a reliable solution to limit the use of polluting refrigerants [4]. $CO_2$is not used in West Africa. Thus the fundamental aim of this study is to investigate the thermodynamic performance of the transcritical CO_2 booster refrigeration system at two temperature levels in tropical climatic conditions.

Specificities of carbon dioxide as a refrigerant:

Thermo-physical properties of carbon dioxide (CO_2):

Carbon dioxide is a non-toxic natural refrigerant, inexpensive and environmentally friendly and can be used in heat pumps and refrigeration systems. The critical temperature of CO_2 is low (31°C), which can sometimes lead to transcritical regimes. In transcritical cycles, the perating high pressure is above the critical pressure (73.8 bar). There is therefore no phase change when the fluid cools down at high pressure. Contrary to the classic thermodynamic cycle, the heat transfer in the high pressure part is no longer condensation but gas cooling. The "condenser" of a refrigeration machine using a transcritical CO_2 cycle is then called a "gas cooler" [3]. The temperature during transcritical steam cooling varies greatly. The process performs well when combined with heat recovery to form a special Lorentz cycle [2]. Table 1 shows the thermophysical properties of $CO_2$and other refrigerants. It can be seen that at the same temperature, the pressure of CO_2 is high compared to other refrigerants. This means that the temperature variation (δT) associated with a pressure drop (δP) is small. Therefore, flows at higher mass velocities are possible, thus improving heat exchange (Table 1). In the liquid state, the density of $CO_2$is slightly higher than that of R717 and lower than that of other refrigerants when its dynamic viscosity is lower (Table 1). The thermal conductivity of CO_2 is higher than that of other fluids. The vapour surface tension must be taken into consideration when boiling refrigerants. Compared to other refrigerants, the surface tension of CO_2 is low. The low value of the surface tension promotes nucleate boiling, because the heat required for nucleation is lower. For this reason, heat transfer is improved in the evaporator with CO_2. This is due to the contribution of nucleation to the overall heat exchange is greater than that of convection [5]. These properties are simulated under EES at temperatures of 0 and 10°C.

Table 1:- Thermophysical properties of CO_2 and other refrigerants.

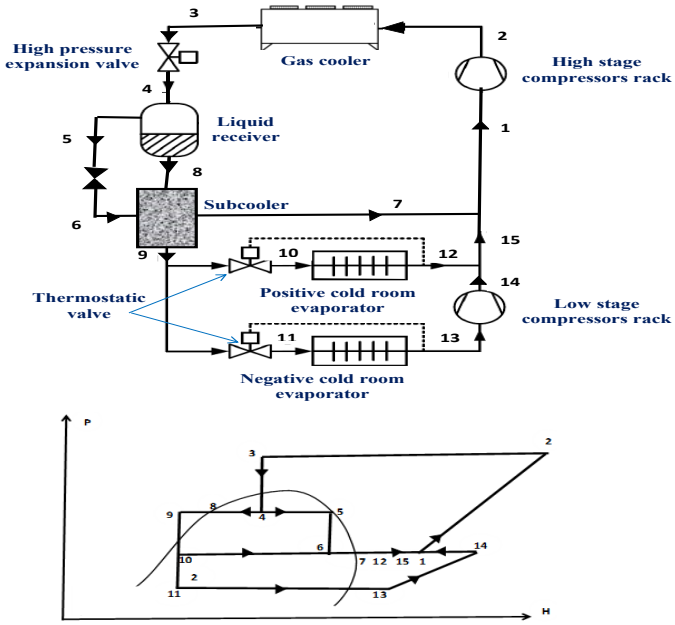
Refrigerants	R744 (CO_2)		R134a		R404A		R717 (NH_3)		R1234yf	
GWP [1]	1		1300		3780		0		4	
Saturation state temperature(°C)	0	10	0	10	0	10	0	10	0	10
Pressure (MPa)	3.485	4.502	0.293	0.415	0.610	0.827	0.429	0.615	0.315	0.437
Density at the bubble point (kg/m3)	927.4	861.1	1295	1261	1150	1110	638.6	624.6	1177	1144
Density at the dew point (kg/m3)	97.65	135.2	14.44	20.24	30.48	41.69	3.458	4.87	17.69	24.33
Enthalpy at the bubble point (kJ/kg/K)	2.542	2.998	1.341	1.37	1.39	1.44	4.617	4.676	1.263	1.294
Enthalpy at the dew point (kJ/kg/K)	1.865	2.558	0.8973	0.9456	0.9968	1.072	2.68	2.842	0.9364	0.9721
Dynamic viscosity in saturated liquid state (µPa. s)	99.39	82.55	265.5	234.1	176.6	154.4	170.1	153	219.7	193.7
Dynamic viscosity at the dew point (µPa. s)	14.79	16.06	10.92	11.32	11.46	12	9.056	9.364	10.02	11.48
Thermal conductivity in saturated liquid state (W/m/K)	0.1087	0.0955	0.0946	0.0902	0.0766	0.0733	0.5592	0.529	0.0750	0.0713
Thermal conductivity in saturated vapour	0.0188	0.0234	0.0121	0.0130	0.0125	0.0136	0.0234	0.0244	0.0091	0.0098

state (W/m/K)										
Surface tension (mN/m)	4.344	2.674	11.56	10.14	6.847	5.679	25.46	23.13	9.323	8.001

Transcritical booster cycles:-

In conventional booster refrigeration systems (two-stage systems), carbon dioxide is cooled but not condensed at the outlet of the gas cooler, since it is at temperatures above the critical temperature. The schematic diagram of the system is shown in Figure 1. The CO_2 discharged from the low pressure compressor is routed through an intercooler to the suction side of the high pressure compressor (point 1). After the intercooler has separated the vapour from the liquid (point 8), this separated vapour is cooled through an electronic valve (between point 5 and point 6) to exchange heat with the separated liquid through the subcooler. The sub-cooled liquid at the outlet of the subcooler (point 9) enters the different evaporators (medium and low temperature) through different expansion valves (points 10 and 11). The vapour leaving the low-temperature evaporator is pressurised by the low-pressure compressor (item 14). This vapour is mixed with the vapour leaving the medium temperature evaporator (item 12) and the vapour from the subcooler (item 7). The mixed vapour is sucked in (item 1), compressed and discharged by the high-pressure compressor into the gas cooler (item 2). The refrigeration booster system is a promising candidate for the integration of refrigeration systems in supermarkets because of its negligible environmental impact. Ge and Tassou have developed a model for the thermodynamic analysis of a $_{CO2}$ booster refrigeration system which shows that the optimal refrigerant pressure at the chiller only varies with the ambient air temperature and the efficiency of the suction line heat exchanger. This pressure is independent of the medium and low pressures as well as the superheat at the medium and low temperature evaporators [6]. By using mechanical subcooling, the energy consumption of the system is reduced according to the results obtained by Gullo and his collaborators. The total equivalent warming impact (TEWI) is reduced by at least 9.6% compared to cascade systems [7]. Farsi et al. proposed a transcritical CO_2 refrigeration system combined with a desalination booster system. The authors reported that the combined system saves 37.8% and 29.1% respectively of the total annual energy consumption costs in Iran and Toronto [8].

Figure 1:- Transcritical CO_2 booster refrigeration systems.



Applications of equipments with CO_2:

Heat pumps:

Heat pumps are defined as systems containing a heat transfer fluid for seasonal comfort heat production, for the production of hot water in residential areas throughout the year, for swimming pool heating, for the production of industrial heat, 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. A comparison between CO_2 and R134a has been made using air-to-water heat pumps used in domestic hot water heating applications [9].

The results show that $_{CO2}$ can replace synthetic refrigerants provided that the system is designed to take advantage of its benefits.

Commercial, industrial and automotive refrigeration:
In automotive air-conditioning, lightweight, ultra-compact systems are needed [10]. Driven by the desire to use environmentally friendly technology, this application in vehicles is very recent and significant advantages are noted [11]. Gbenagnon et al. recently evaluated the use of natural refrigerants and their mixtures for vehicle air conditioning. It was pointed out that the critical point of the mixture obtained from CO_2 and R600a is close to that of R134a. [12]. The application of CO_2 as a refrigerant in air conditioning systems and supermarkets shows that CO_2 systems can be favourably compared to traditional systems using R12 or R134a in terms of power, energy, installation cost, weight and dimensions [10]. The design and experimental analysis of a transcritical carbon dioxide chiller for commercial refrigeration was presented by Cecchinato et al. to increase the performance of refrigeration machines operating with transcritical CO_2 [13].

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 [14]**.** Optimisation is carried out for the system when it cogenerates energy and refrigeration or produces refrigeration only. The lowest evaporation temperature is reached when the system is optimised for maximum exergy efficiency. B. Li et al. have described the advantage of the combined system over the conventional system [15]. At evaporation temperatures of 273.15 K and 253.15 K respectively, they showed that the combined system has an efficiency of 2.45% and 5.87% higher than the separate system.

Modeling the transcritical cycle booster:
Refrigeration cycle modeling assumptions:
In order to analyse the performance of the operating cycle of booster and cascade refrigeration systems, the following hypotheses were made. These refrigeration needs and hypotheses correspond to those of a shopping centre in Benin.

1. Medium temperature cooling capacity (QMT) :130 kW
2. Low-temperature cooling capacity (QLT) :30 kW
3. Evaporating temperature average temperature ($_{TMT}$) :-10 °C
4. Low temperature evaporation temperature ($_{TLT}$) :-35°C
5. Pressure at the outlet of the gas cooler and the ambient temperature (Tamb) are selected : 75bar ;140 bar] with 25°C ≤ Tamb ≤ 40°C [16];
6. The efficiency of the subcooler is 0.7
7. The isentropic efficiency of compressors is modelled according to the analysis of [16].
8. The superheat at the evaporator is 10°C.

Simulation models:
The thermodynamic models applied to the transcritical booster cycle are summarised in Table 2. For each component of the system, the mass and energy balance equations were applied. Each process was mathematically represented and integrated into EES (Engineering Equation Solver). W_{MT} and W_{LT} refer respectively to the different works supplied to the high and low pressure compressors,
$\dot{m}_i$: the mass flow rate of the refrigerant at the different points of the refrigeration system,
h_i : thecorresponding mass enthalpies at the different points of the cycle,
P_r : thecompression ratio,
η_{LP}: theisentropic efficiency of the low temperature compressor,
η_{HP} : the isentropic efficiency of the high temperature compressor,
P_{gc} : the pressure at the outlet of the gas cooler,
T_{amb}: the ambient temperature,
T_{gc}: the temperature at the outlet of the gas cooler,
P_{int}: the intermediate pressure,
$\dot{m}_{MT}$: the mass flow rate of the refrigerant in the medium temperature evaporator,
$\dot{m}_{LT}$: the massflow rate of the refrigerant in thelow temperature evaporator
and $\dot{m}_{gc}$: the mass flow rate of the refrigerant in the gas cooler.

Table 2:- Thermodynamic models applied to the transcritical booster type refrigeration cycle.

Components	Mass and energy balance
Gas cooler	$T_{gc} = T_{amb} + 3°C$ for $T_{amb} > 25°C$[17] $P_{gc} = \exp(4.361 - 0.020.T_{amb} + 0.0007.T_{amb}^2)$ $25°C \leq T_{amb} \leq 40°C$ and $75bar \leq P_{gc} \leq 140$ bar [16]; $\dot{m}_{gc} = \frac{\dot{m}_{MT} + \dot{m}_{LT}}{1-x4}$
Intermediate container	$P_{int} = 162.57 - 3.66.T_{amb} + 1.25.P_{gc} + 0.042.T_{amb}.P_{gc}$ [16]
Subcooler	$h_9 = h_8 - \frac{x4(h_7 - h_6)}{1-x4}$; $\dot{m}_7 = x_4.\ \dot{m}_{gc}$; $E = \frac{T_8 - T_9}{T_8 - T_7}$
High temperature Compressor	$\eta_{HP} = \exp[-0.397 + 0.0307.P_r - 0.0092.(P_r)^2]$, with $2.22 \leq P_r \leq 3.71$ [16] $h_2 = h_1 + \frac{h_{2is} - h_1}{\eta_{HP}}$, $W_{HP} = \dot{m}_{gc}(h_2 - h_1)$
Low temperature Compressor	$\eta_{LP} = \exp[-0.584 + 0.1902.P_r - 0.0449.(P_r)^2]$ with $1.58 \leq P_r \leq 2.93$ [16] $h_{14} = h_{13} + \frac{h_{14is} - h_{13}}{\eta_{LP}}$; $W_{LP} = \dot{m}_{LT}(h_{14} - h_{13})$
Expansion valves	$h_{in} = h_{ou}t$
Medium temperature evaporator	$Q_{MT} = \dot{m}_{MT}(h_{12} - h_{10})$
Low temperature Evaporator	$Q_{LT} = \dot{m}_{MT}(h_{13} - h_{11})$
Coefficient of performance	$COP = \frac{Q_{MT} + Q_{LT}}{W_{HP} + W_{LP}}$

Results and Discussion:-

Analysis of the properties of supercritical carbon dioxide:

A supercritical fluid is when a fluid is heated above its critical temperature and being compressed above its critical pressure. The physical properties of a supercritical fluid (density, viscosity, diffusivity) are intermediate between those of liquids and gases. Supercritical carbon dioxide has a high density and its viscosity is similar to that of the gas but much lower than the viscosity of the liquid. It undergoes very large variations without phase change [2]. It has good flow and heat transfer characteristics (Figure 2). The thermal mass capacity of CO_2 changes from typical gas values at high temperatures to typical liquid values at low temperatures. While the specific heat of CO_2 is infinite at the critical point, it decreases at higher pressures and peaks at higher temperatures. Under a given supercritical pressure (Figure 3), the density decreases with increasing temperature.At a constant temperature, the density increases with increasing pressure. When CO_2 is close to the critical point, its density is very sensitive to changes in pressure and temperature, i.e. a small change in pressure or temperature can cause a dramatic change in density. It can be seen that at a given pressure, the density decreases with an increase in temperature when the enthalpy increases at the same pressure. The variations in mass volume and thermal conductivity along the 75 to 160 bar isobars within a temperature range of 10 to 150 °C are shown in Figures 4 and 5 respectively. Figure 4 shows that the physical properties of CO_2 in thesupercritical state undergo a sudden change as it approaches the critical point. The greatest change occurs exactly at the critical point. The points, other than the critical point, where the properties change are called pseudo-critical points. Here, the specific heat has a peak for a given pressure (Figure 2). These points correspond to the inflection points of the isotherms as shown in Figure 6. The red curve starting from the critical point represents the evolution of the pseudo-critical temperature with pressure. These temperatures represent the place of these rapid changes in properties with temperature during isobaric evolution. For a given pressure, the pseudo-critical temperature is the temperature for which the mass thermal capacity is maximum. The particularity of the super-critical region, apart from the strong variations of the properties, is that the temperature is no longer coupled with the pressure. The high pressure of the trans-critical cycle is no longer imposed and a search for its optimal value is necessary.

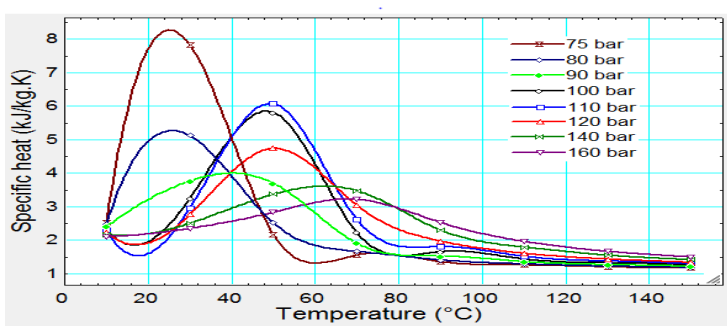


Figure 2:- Mass heat as a function of temperature.

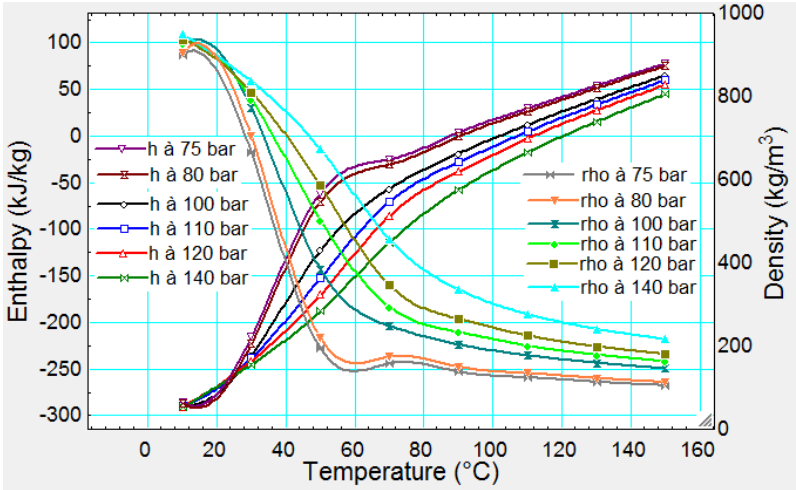


Figure 3:-Variation of density and enthalpy as a function of temperature.

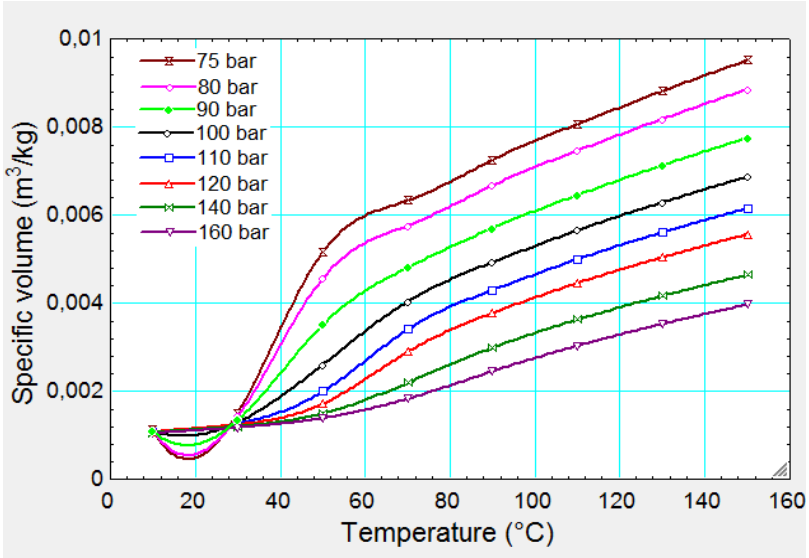


Figure 4:- Variation of volume-mass as a function of temperature.

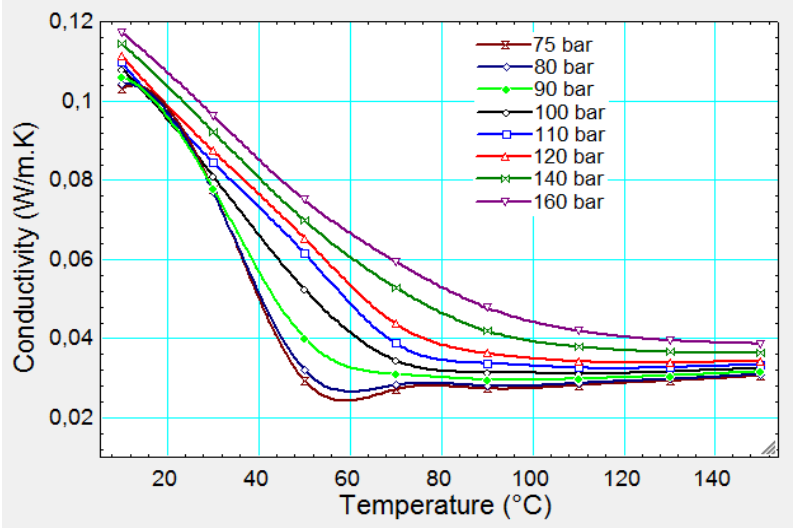


Figure 5:- Conductivity variation as a function of temperature.

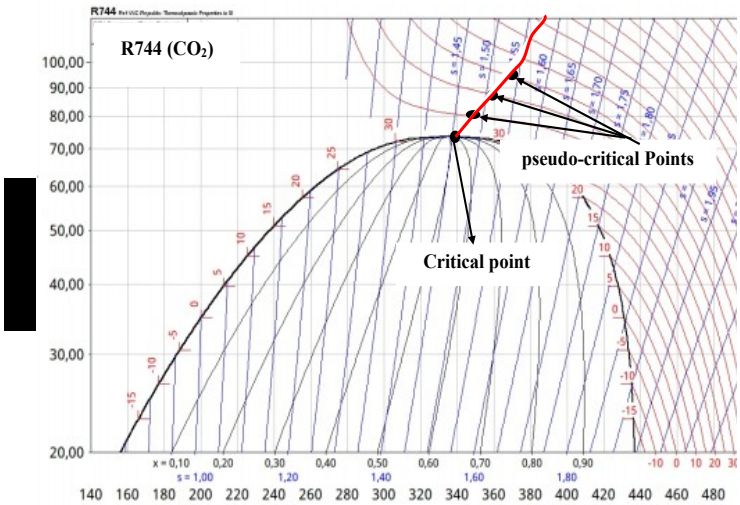


Figure 6:- Illustration of pseudo-critical points.

Ambient temperature in the south of Benin (Cotonou):

During the year, the temperature generally varies between 24°C and 32°C and is rarely below 21°C or above 33°C. Figures 7 and 8 show the average annual temperature in southern Benin (Cotonou) and the average daily temperature in southern Benin (Cotonou), respectively. These data were collected using Meteonorm 7 software.

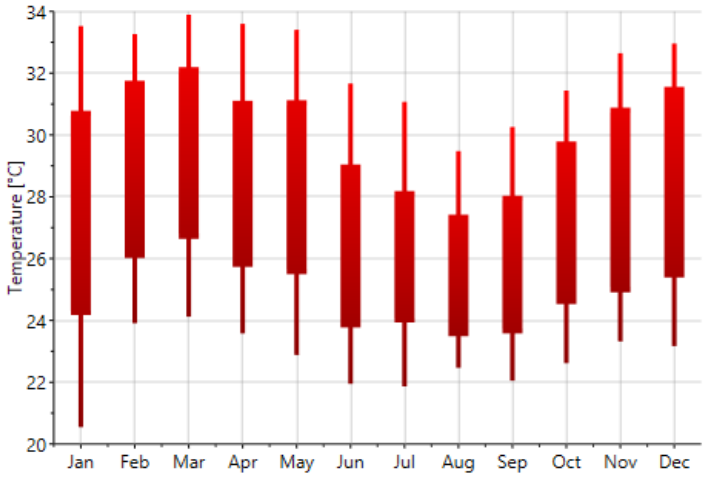


Figure 7:- Average annual temperature in southern Benin (Cotonou).

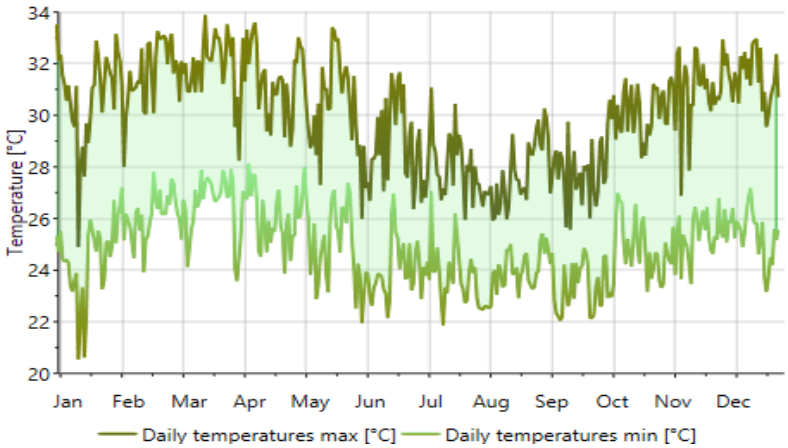


Figure 8:-Mean daily temperature in southern Benin (Cotonou)

Effect of subcooler efficiency on the coefficient of performance of the refrigeration booster system:
The liquid subcooler is an arrangement often used in transcritical refrigeration plants to achieve heat exchange between suction and liquid lines. Its advantage is that there are no other exchanges than heat. It maintains the physico-chemical characteristics (pressure, concentration of chemical elements, etc.) of the fluids unchanged apart from their temperature or state. Figure 9 shows the coefficient of performance (COP) as a function of subcooler efficiency. For this simulation, the efficiency is varied by keeping constant the ambient temperature (Tamb = 30°C), the low temperature evaporator temperature (TLT = -35°C) and the medium temperature evaporator temperature (TMT= -10°C).

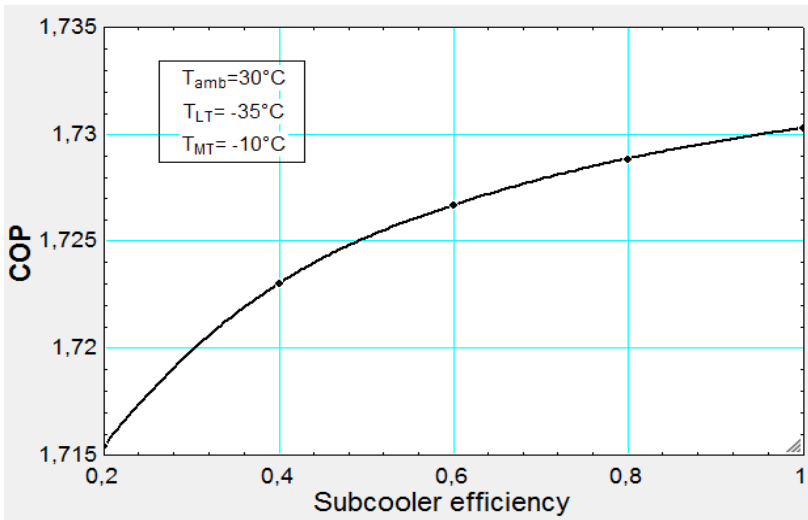


Figure 9:- Variation in coefficient of performance as a function of subcooler efficiency.

Variation of the coefficient of performance as a function of the ambient temperature:
The medium-temperature (Q_{MT}), low-temperature (Q_{LT}) refrigeration capacities and evaporator evaporation temperatures (T_{LT}= -35°C, T_{MT} = -10°C) are kept fixed. Figure 10 shows the variation of the coefficient of performance (COP) as a function of ambient temperature. The highest COP value was 2.37 at a temperature of 21°C. The COP becomes less than 1 when the temperature exceeds 45°C. The COP of the refrigeration booster system increases when the ambient temperature decreases.

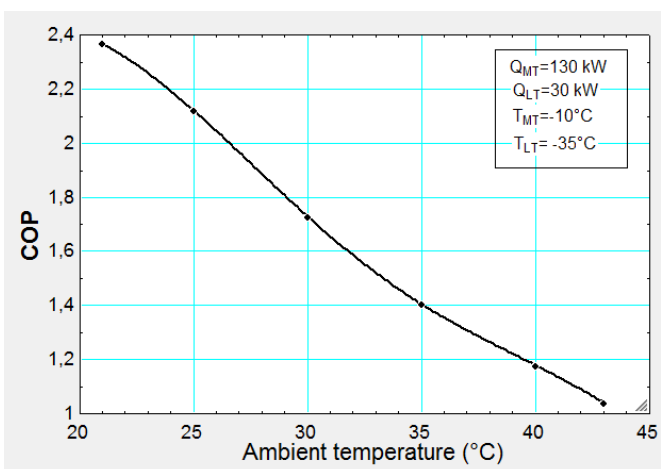


Figure 10:- Variation of the coefficient of performance (COP) as a function of ambient temperature.

Comparison of mass flow rates as a function of room temperature:
The refrigerant mass flow rate in the medium temperature evaporator ($\dot{m}_{MT}$), the refrigerant mass flow rate in the low temperature evaporator ($\dot{m}_{LT}$)and the refrigerant mass flow rate in the gas cooler ($\dot{m}_{gc}$) of the booster system are modelled. Mass flow analysis can help determine the amount of refrigerant charged into the system. The variations of those mass flow rates as a function of ambient temperature are shown in figure 11. After 26 °C, the value of the

mass flow rate ($\dot{m}_{MT}$) in the medium temperature evaporator first decreased slightly and then gradually increased, resulting in a decrease in the rate of growth of the mass flow rate ($\dot{m}_{gc}$) in the gas cooler; then decreased slightly. This is due to the operating pressure of the gas cooler.

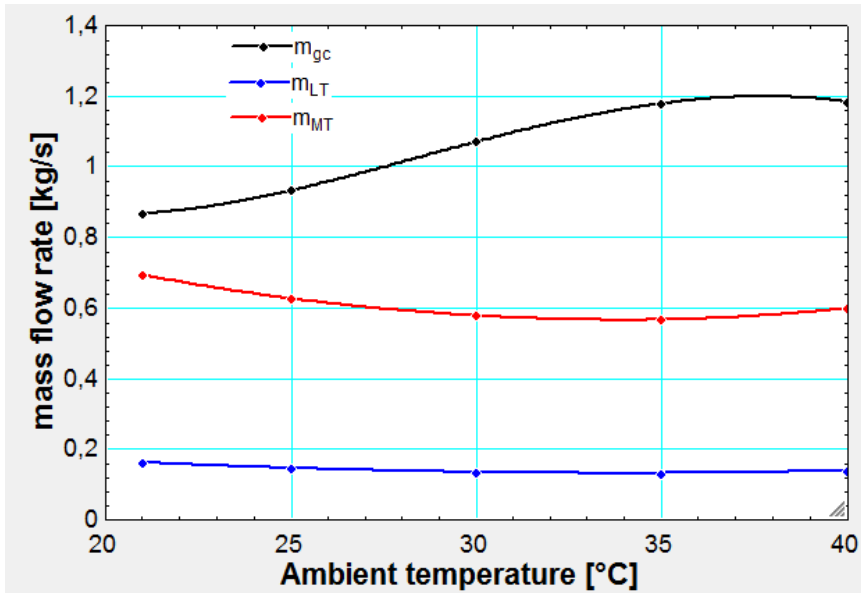


Figure 11:-Variation of mass flow rates as a function of ambient temperature.

Variation in compressor work requirements as a function of gas cooler pressure:-

Figure 12 shows the work required at high and low pressure compressors as a function of gas cooler pressure. Both work requirements increase as a function of the gas cooler pressure. The work required at the high pressure compressor (W_{HP}) at 80 bar is 86 kW when the work required at the low pressure compressor (W_{LT}) is 7 kW. The work required at the compressors of transcritical booster refrigeration systems is high because of the high volumetric power generated by carbon dioxide ($_{CO2}$) due to the high working pressures, which limits the performance of these systems.

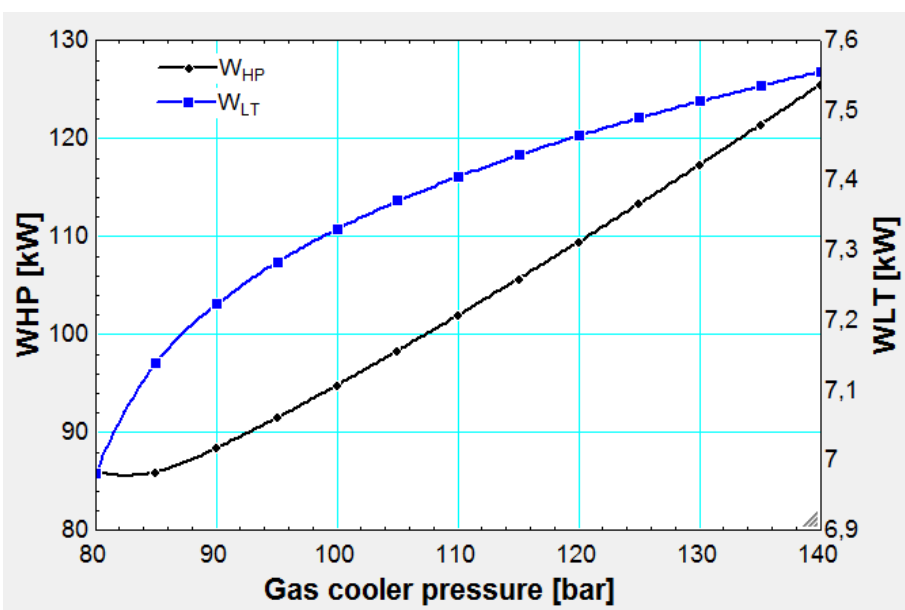


Figure 12:- Variation in compressor work requirements as a function of gas cooler pressure.

The effect of the coefficient of performance as a function of the gas cooler pressure for different evaporating temperatures:

The coefficient of performance (COP) value is plotted against the gas cooler pressure for different evaporating temperatures of the two evaporators (Figure 13). As the evaporating temperatures (T_{MT} and/or T_{LT}) of the evaporators increase, the COP increases slightly up to a maximum and then decreases with higher pressure values.

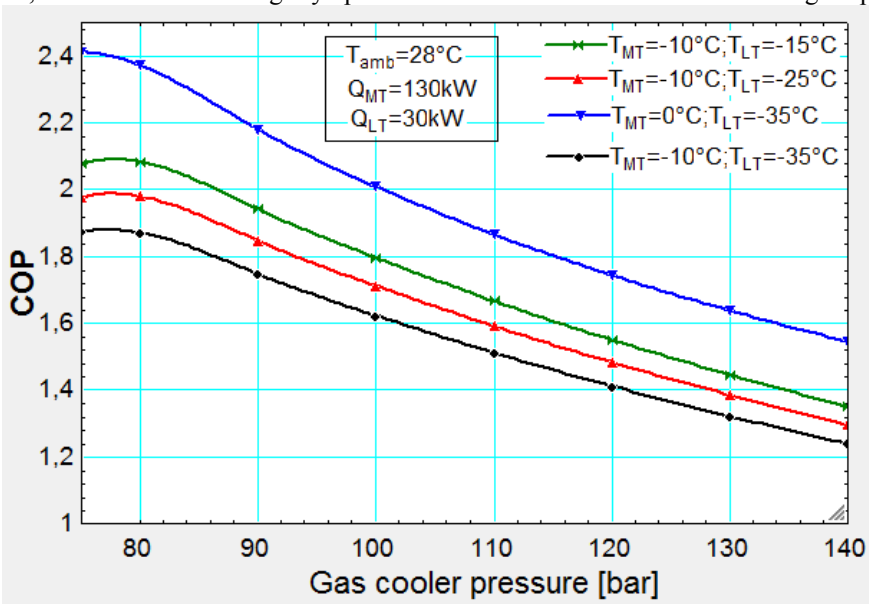


Figure 13:- COP variation as a function of gas cooler pressure for different evaporator temperatures.

Variation of the coefficient of performance as a function of the gas cooler pressure:

Figure 14 shows the variation in the coefficient of performance (COP) as a function of gas cooler pressure (Pgc) at different ambient temperatures (or as a function of gas cooler condensing temperature (T_{gc}): T_{gc}= T_{amb}+3°C). The intermediate pressure is maintained at 50 bar. We can see that the COP of the system increases rapidly for low Pgc and after reaching the maximum value, it decreases slowly as the Pgc increases. This phenomenon is observed for ambient temperatures above 25°C. At 25°C the COP decreases as the pressure increases. It can be deduced from this that the ambient temperature and the COP have a significant effect on the optimum pressure of the gas cooler.

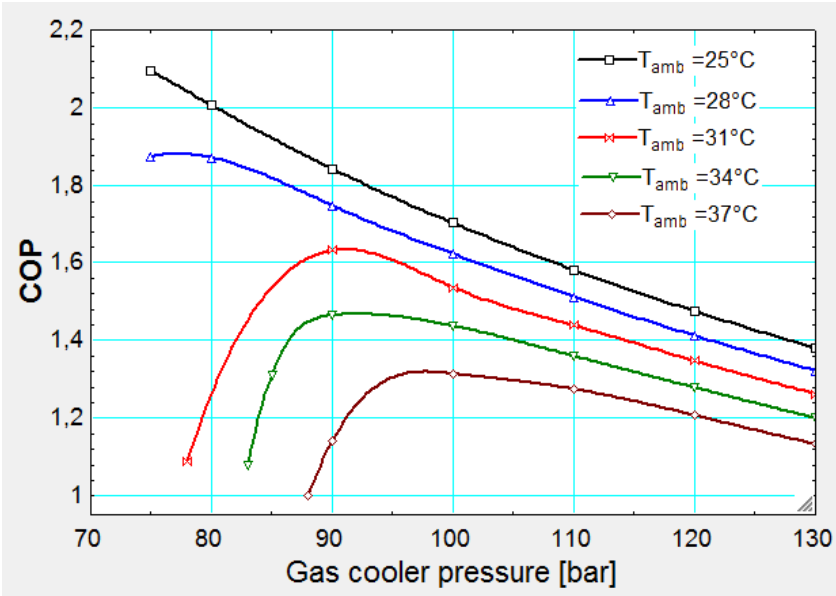


Figure 14:- Effect of gas cooler pressure on coefficient of performance.

Optimisation and correlation of gas cooler and intermediate pressures:

By performing polynomial regression analysis on the data obtained from the cycle optimisation, the following relationships were established to predict the optimum gas cooler pressure and the optimum intermediate pressure

$P_{opt,gc} = a + b.T_{amb} + c.T_{amb}^2 + d.T_{amb}^3 + e.T_{amb}^4 + f.T_{amb}^5 + g.T_{amb}^6$; with $R^2 = 1$ (1)

$P_{opt,int} = a + b.T_{amb} + c.T_{amb}^2 + d.T_{amb}^3 + e.T_{amb}^4 + f.T_{amb}^5 + g.T_{amb}^6$; with $R^2 = 0.9621$ (2)

These correlations are valid for ambient temperatures ranging from 25°C to 40°C.

Table 3 illustrates the coefficients of the developed correlations of the optimal pressures $P_{opt,gc}$ and $P_{opt,int}$.

Table 3:- Coefficients of developed correlations of optimal pressures.

	$P_{opt,gc}$	$P_{opt,int}$
a	$-7.59881832.10^4$	$1.56325294.10^2$
b	$1.48294836.10^4$	$-1.72313442.10^1$
c	$-1.19283100.10^3$	1.44502124
d	$5.06751408.10^1$	$-6.77026153.10^{-2}$
e	-1.19958296	$1.74679283.10^{-3}$
f	$1.50092124.10^{-2}$	$-2.35462322.10^{-5}$
g	$-7.75807959.10^{-5}$	$1.35685059.10^{-7}$

Figure 15 illustrates the variations in the optimum gas cooler pressure and the optimum intermediate pressure of the transcritical carbon dioxidebooster refrigeration system as a function of the ambient temperature. The optimum pressure of the gas cooler increases with ambient temperature. The intermediate pressure drops slightly (56.4 bar at 25°C) as the temperature rises, reaching 53.4 bar at 31°C before rising (62.56 at 40°C). The curves are compared with experimental results from the literature:

$P_{opt,int} = -0.0105.T_{gc}^2 + 1.1237.T_{gc} + 25.5207$[18] (3)

$P_{opt,gc} = 2.312.T_{gc} + 11.9$; with $R^2 = 0.9991$[18] (4)

$P_{opt,gc} = 2.3426.T_{amb} + 11.541$ [19] (5)

$P_{gc,gc} = \exp(4.361 - 0.020.T_{amb} + 0.0007.T_{amb}^2)$[17] (6)

The values of the optimal pressures $P_{opt,gc}$ and $P_{opt,int}$ of the system simulated in this article are close to the results obtained by the authors (Table 4). The discrepancy observed may be due to the simulation tool and modeling assumptions.

Table 4:- Comparison of optimal $P_{opt,gc}$ and $P_{opt,int}$ pressures.

T_{amb} (°C)	Intermediate pressure $P_{opt,int}$			Gas cooler pressure $P_{opt,gc}$		
	This study	[18]	Relative error (%)	This article	[17]	Relative error (%)
28	54.36	50.26	7.54	76.32	77.46	1.493
31	53.4	51.59	3.38	86.05	82.57	4.04
34	54	52.72	2.37	94.24	89.14	5.41
37	56.77	53.67	5.46	100.5	97.45	3.03

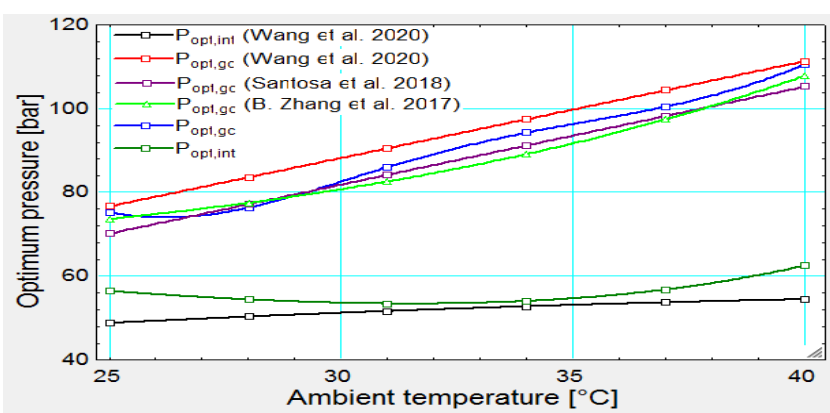


Figure 15:- Variations of the optimum gas cooler pressure and intermediate pressure as a function of ambient temperature.

Conclusion:-

The transcritical carbon dioxide booster refrigeration cycle model was simulated thermodynamically within Benin climatic conditions. The high working pressure characterises the transcritical CO_2 cycle. The performance of the transcritical booster cycle becomes low when the ambient temperature rises and also at low evaporation temperatures. This led to the search for the optimal high pressure of the gas cooler and the optimal intermediate pressure at which performance reaches its maximum value; this pressure is very sensitive to the variation of the ambient temperature. The production of transcritical CO_2 refrigeration is therefore feasible in Benin. The transcritical CO_2 booster refrigeration system is an interesting solution for the environment because it reduces the carbon footprint and can be adapted to the weather conditions in Benin. In order to reduce the pressures within the gas cooler and improve heat exchange in hot periods, booster refrigeration systems can then be equipped with a safety degassing valve and the gas cooler can be fitted with sprinkler ramps to lower the temperature.

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