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Effect of high temperatures on the efficiency of sub-critical CO_2 cycle

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ABSTRACT

Thermodynamic efficiency is a crucial factor of a power cycle. Most of the studies indicated that efficiency increases with increasing heat source temperature, regardless of heat source type. Although this assumption generally is right, when the heat source temperature is close to the critical temperature, increasing the heat source temperature can decrease efficiency. Therefore, in some cases, the increase in the source temperature, like using improved or more collectors for a solar heat source can have a double negative effect by decreasing efficiency while increasing the installation costs. In this paper, a comparison of the CO_2 subcritical cycle and the Trilateral Flash Cycle will be presented to show the potential negative effect of heat source temperature increase.

KEYWORDS

subcritical carbon dioxide cycle, Trilateral Flash Cycle, critical temperature, heat source, maximum efficiency

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1. INTRODUCTION

The demand for energy in the world increases; this increase is preferably satisfied by modern power plants using a clean, renewable source. A special class of these power plants is the one operated by low heat sources, using organic Rankine cycles or carbon dioxide power cycle. The transcritical CO_2 power cycle (working with heat sink temperature below and heat source temperature above the critical temperature (T_{cr}) of CO_2, namely 31 °C) has significant and sufficient potential to convert the heat to produce power (electricity) by using the carbon dioxide as a working fluid due to its good thermodynamic and environmental properties [1, 2]. Many thermodynamics cycles are applicable in the temperature below 350 °C, like the CO_2 transcritical power cycle, Organic Rankine Cycle (ORC) and Trilateral Flash Cycle (TFC), instead of high-temperature steam Rankine Cycle (RC) [1, 3–5]. CO_2 has a low critical temperature, which makes it appropriate for utilizing a low heat source – geothermal [6, 7]. CO_2 power cycles are widely used in air conditioning, heat pumps, refrigerating systems, and power cycles [8–11]. CO_2 power cycle can be used to utilize solar heat, but due to the weather-dependence of this source, a heat storage system is required to provide continuous heating. Also, integration of the absorption refrigeration system with reheat transcritical power cycle leads to improve the efficiency and maintain stable productivity by keeping the low condensation temperature at all different weather conditions. Using the compressed air energy storage help to overcome the decrease or interruption for solar energy and improves the technical flexibility in solar thermal power and storage [12, 13]. Using a regenerative heat exchanger (recuperator) in the CO_2 transcritical cycle improves overall net power and overall thermal efficiency [14]. At the low and high heat source temperature, the transcritical CO_2 power cycle using mixed CO_2 is better than the cycle that using pure carbon dioxide, thermodynamically, and exergo-economically. It has been shown by analyzing binary mixtures of carbon dioxide with other refrigerants (R32, R1270, R161, R1234yf, R1234ze, R152a), and alkanes (butane, pentane, propane, isobutene or isopentane) that the highest exergy efficiency and the lower levelized cost per unit of exergy product was with CO_2/propane at the high-temperature heat source. At lower heat source temperature, the highest exergy efficiency was with CO_2/R32, and the lowest levelized cost per unit of exergy product was with CO_2/R161 [15]. With CO_2 mixtures that consist of binary mixture of CO_2 with one of these refrigerants (R152a, R161, R290, R1234yf, R1234ze, and R1270) with a transcritical power cycle at geothermal water temperature between 100 and 150 °C and temperature of cooling tower 10–30 °C, it has been observed that the better thermal performance and economic performance was with R161/CO_2, while the R290/CO_2 was the worst due to low thermal performance. The cost per net power reduction, decrease of operating pressure and extension of the range of condensing temperature, all these occur with the blends of CO_2 more than the CO_2 in a pure state. At the low cooling water temperature, R152a/CO_2 mixed working fluid is not suitable with the proposed system [16]. The comparison between basic, recuperator, reheat, and regenerative systems for transcritical CO_2 power cycle demonstrated that reheat transcritical CO_2 cycle is the best one, concerning thermo-economic performance, and reheat system showed an increase in net power produced, energy efficiency, and efficiency of exergy compared the basic transcritical CO_2 cycle while the total investment cost is higher for reheat system due to largest heat transfer area [17]. Condensing is one of the problems facing the conventional CO_2 trans- and subcritical CO_2 power cycles by using traditional water cooling, but with self-condensing, the CO_2 carbon dioxide transcritical power cycle overcome this problem and can operate well with the cooling water as warm as 30 °C [18, 19]. The CO_2 transcritical power cycle has a better economic performance than the organic Rankine cycle in terms of cost per net power output and under a certain turbine inlet pressure. The cost per net power output in the regenerative CO_2 transcritical power cycle is even lower than that of the basic CO_2 transcritical power cycle, that which observed by analyzing the organic Rankine cycle and CO_2 transcritical cycle with a geothermal heat source and different working fluids for example isobutane, R123, pentane R245fa [20].

The subcritical CO_2 power cycle (CO_2 Rankine cycle) can use the ambient temperature or low-temperature geothermal as a heat source; these two sources are classified as a low enthalpy source [21]. To properly characterize the transcritical cycle, it requires a deep knowledge of the subcritical carbon cycle [22]. When the ambient temperature or some other source with similar temperature (for example, thermal water not above 35–40 °C) are used as a heat source, then - depending on the weather conditions - transcritical cycles might shift to the sub-critical region (i.e., the maximum temperature will be below the critical temperature of CO_2). In that case, it is better to use a subcritical CO_2 cycle. Still, engineers should know that there is a narrow temperature range near to the critical temperature, where the thermodynamic efficiency has inverse maximum cycle temperature dependence. Therefore, there is a temperature range, which should be avoided during this application. The aim of the study is to demonstrate to engineers and researchers that the efficiency does not always increase with increasing heat source temperature, but sometimes the increasing of the heat source temperature close to the critical temperature leads to decreasing the efficiency. That can also happen with other thermodynamics cycles using various working fluids, for example Rankine cycle and organic Rankine cycle. This study focused on the subcritical CO_2 power cycle with low heat source temperature close to the critical points.

2. METHODS

The water and some organic materials are working fluids used in the power plants using steam and organic Rankine cycles, respectively. In contrast, in the CO_2 power cycle, carbon dioxide is used as a working fluid. The subcritical CO_2 power cycle operates under temperature and pressure not exceeding the critical point for CO_2. Therefore, the low heat source temperature, like ambient temperature (T_{amb}) or geothermal, is sufficient for the subcritical CO_2 power cycle. The ideal cycle was used in this study that was an isobaric process at the heat exchangers and isentropic steps at the expansion and compression.

2.1. Components and processes of CO_2 power cycle

The subcritical CO_2 cycle is very similar to the simple steam Rankine cycle; therefore it is called the CO_2 Rankine cycle. Evaporator, turbine, condenser, and pump are demand to configuration the CO_2 power cycle, as it is shown in Fig. 1a. The T–s diagram shows the processes of the CO_2 power cycle in Fig. 1b. The CO_2 compressed from point 1 to point 2 by a pump in an isentropic process. A slight increase in temperature occurs, together with the rise of the pressure. Then, CO_2 enters the heat exchanger (sometimes called evaporator) at point 2; here, in the initial part, the temperature will increase. Then, after reaching the boiling temperature for the given pressure, evaporation happens, even the total mass of the fluid reaches a vapor state (point 3). At point 3, the fluid is in high pressure and high temperature saturated vapor state. Here, the working fluid enters the turbine (or expander) with high pressure and temperature to produce the mechanical work by expansion between points 3 and 4. During this process (taken as ideal adiabatic, i.e., isentropic one) pressure and temperature of the working fluid is decreased. In stage 4, the fluid is in a low-enthalpy, low-temperature, low-pressure saturated vapor state. Between points 4 and 1, part of the heat is removed from the system isobarically in a second heat exchanger, called condenser, causing complete condensation from saturated steam to saturated liquid state in order to start a new cycle.

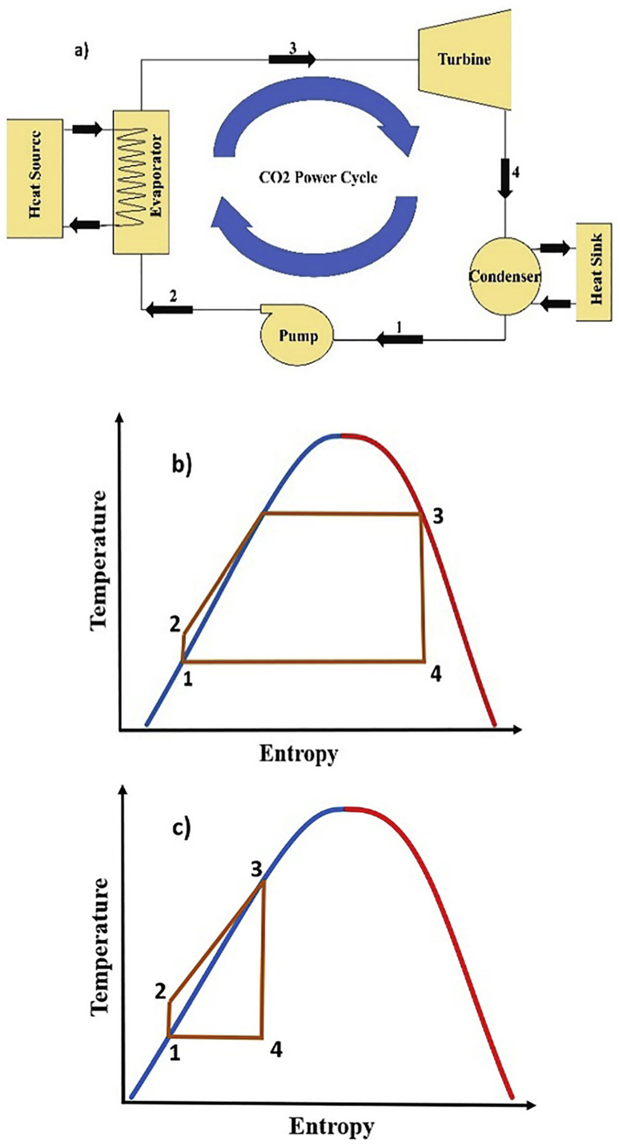

Fig. 1. Schematic diagram for CO_2 power cycle, a) Main components of cycle, b) *T–s* diagram for CO_2 power cycle, and c) *T–s* diagram for TFC cycle

There is a slightly different cycle, called TFC; it is characterized by its simplicity. Heat addition terminates, when the saturated liquid state reached (point 3, Fig. 2). In this way, the fluid volume between points 2 and 3 is not significantly different, and therefore the heat exchanged for TFC can be simpler, than for ORC. In general, the process 1–2–3–4 is similar to the CO_2 cycle used in the study. The two differences are the lack of evaporation in the 'evaporator' and the "wet expansion" (i.e., expansion starting from the liquid state) in the expander [23]. The difference can be seen in detail in Fig. 1b and c.

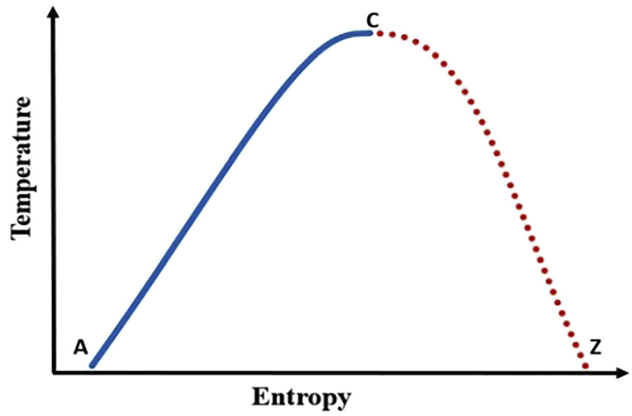

Fig. 2. T–s diagram for carbon dioxide as wet working fluid with characteristic points (see in text)

2.2. Characteristics of CO_2 as working fluid

Due to the thermal stability, the CO_2 is one of the natural working fluids considered suitable for the refrigeration and power cycles, also; it has significant advantages compared to other working fluids that lack whole or part of these characteristics, which include physical, chemical, environmental, and economic features. There are some crucial factors that must be considered when selecting a working fluid, or it characterizes the working fluids, for chemically factors like non-flammability, low toxicity; physical factors, like high critical pressure (P_{cr}), low critical temperature, low boiling point (B_p), and good heat transfer; the environmental ones, like low Global Warming Potential (GWP), low Ozone-Depleting Potential (ODP), and environmentally friendliness (very safe to use); and finally, low cost as the economic factor. Carbon dioxide is one of the liquids whose properties satisfy the above-mentioned characteristics; additionally, it is a "natural" fluid. The CO_2 properties and American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) for CO_2, are shown in Table 1. The CO_2 can be used in all types of CO_2 cycles like subcritical, transcritical, supercritical; also, it can be used as pure fluid as well as in mixtures with other working fluids especially with hydrocarbons, especially because of its ability to reduce the flammability of hydrocarbons while preserving all the other desired thermodynamic properties [24]. According to the traditional working fluid classification, CO_2 is a wet working fluid; in the novel classification, it belongs to the so-called ACZ class [25].

Table 1. Properties of carbon dioxide
(*Source:* on the basis of [26])

Type	Category	ASHRAE NO.	Formula	ASHRAE level for safety	ODP	GWP	Critical temperature [K]	Critical Pressure [MPa]	Boiling Point [K]
Wet	ACZ	744	CO_2	A1	0	1	304.1282	7.3773	194.75

2.3. Thermodynamic analysis

A calculation was performed to find the efficiency of the CO_2 power cycle with the temperature variation of the heat source by using the MATLAB software and data from the NIST webbook [27]. On the T–s diagram of CO_2, A and Z marks the minimum temperature for liquid and vapor phases, and C is the critical point, as shown in Fig. 2. The blue solid line represents the saturated liquid state, and the red dashed line represents the saturated vapor state. General equations were used. The efficiency was calculated by

$$\eta = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}, \tag{1}$$

where the $(h_3 - h_4)$, $(h_2 - h_1)$, and $(h_3 - h_2)$, are the difference in enthalpy at the turbine, pump, and evaporator respectively, and the enthalpy and other data values as entropy, pressure, and dryness fraction found by NIST webbook.

In this study, some assumptions and parameters applied like the steady-state of the operation cycle, and ignored the pressure losses during the flow. At the saturated vapor phase, the working fluid entering the turbine, and the inlet pressure changes based on the increasing the heat source temperature, while in the TFC cycle, the working fluids enter the turbine at the saturated liquid. Depended on the condenser temperature and dryness fraction, the working fluid enters the condenser then leaves it at the saturated liquid. Ten thousand steps, using different temperature pairs were used in this study (with 1,000 readings for each curve, 500 for the CO_2 cycle, and 500 for the TFC cycle). The condenser line, green solid line is at low temperature, for example, 217 K, and all the horizontal lines above the condenser line, green solid line are the evaporator lines (increase with increasing heat source temperature starting from orange dot line to dark blue long dash line), up to the critical point. First, the 1–2–3–4 cycle efficiency was calculated then evaporation temperature was shifted to a higher value, while the condensation temperature was kept and the efficiency of the new cycle (1–2a–3a–3b) was determined. The process continued upwards, to the vicinity of the critical temperature (see cycle 1–2d–3d–4d) as it is shown in Fig. 3. In the next step, a new (increased) condenser temperature was taken (238 K instead of 217 K green solid line shifts upward). For this new value, efficiencies related to changing evaporation temperature (orange dot, gold dash, purple long dash dot, blue long dash dot dot, and dark blue long dash lines), were also calculated. In the following steps, the process was repeated with new condenser temperatures. Finally, obtained the whole set of efficiency values for various condensation and evaporation temperature pairs from 217 K to the critical temperature was obtained. The same processes were applied for the TFC cycle, except that the entering parameter in the turbine would be saturated liquid, as it is shown in Fig. 4. All boundary conditions are shown in Table 2.

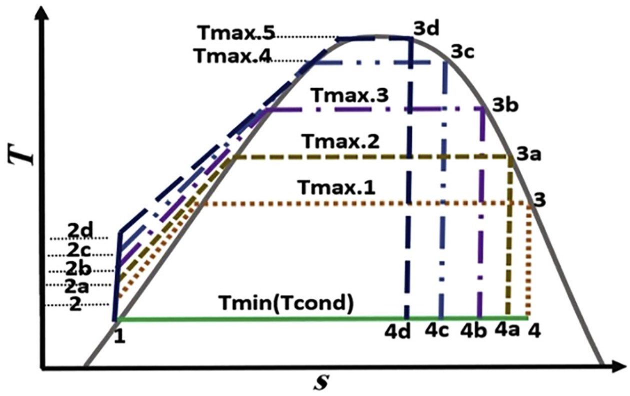

Fig. 3. The procedure of the calculation on a T–s diagram for CO_2 power cycle

3. RESULTS AND DISCUSSION

Carbon dioxide can be applied as working fluid in refrigeration cycles as well as in and power cycles. This study focused on the sub-critical CO_2 power cycle by utilizing low-temperature heat sources for example ambient temperature or geothermal one with several of condenser temperatures. In general, the efficiency increases with increasing the heat

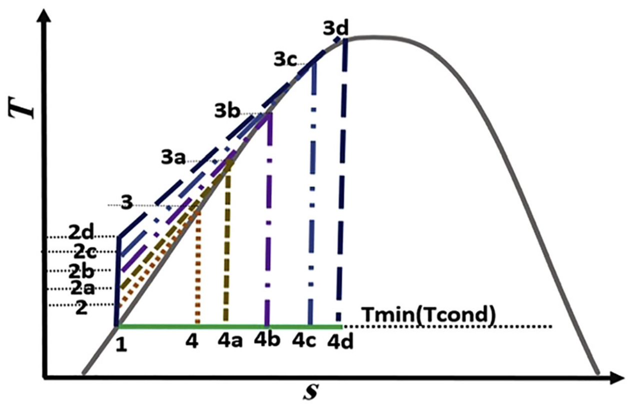

Fig. 4. The procedure of the calculation on a T–s diagram for TFC power cycle

Table 2. The boundary conditions

Run	Heat source temperature [K]	Heat sink temperature [K]	Run	Heat source temperature [K]	Heat sink temperature[K]
1	217 to T_{cr}	217	6	278 to T_{cr}	278
2	238 to T_{cr}	238	7	284 to T_{cr}	284
3	252 to T_{cr}	252	8	290 to T_{cr}	290
4	266 to T_{cr}	266	9	296 to T_{cr}	296
5	272 to T_{cr}	272	10	302 to T_{cr}	302

source temperature. It has been found in sub-critical CO_2 cycles (and can be generalized to all cycles having similar, ACZ-type *T–s* diagram) that while this increase is usually true, choosing the maximal cycle temperature close to the critical point, inverse dependency can be seen a narrow, but definitely non-zero temperature range. In contrary, for TFC, the maximum efficiency was at the critical temperature as it is shown in Fig. 5; the upper curves represent the efficiency values of CO_2 power cycles and the lower curves for TFC cycles. For example, the first curve for CO_2 and TFC, the range of evaporator temperature (heat source temperature) between 217 and 304.1282 K, calculated point by point with an increment equal to (total temperature range)/500, with fixed condenser temperature 217 K. For the next curve, the condenser temperature was increased, and the calculation was repeated.

Separate lines in Fig. 6 represent the variation of efficiency for CO_2 power cycle with increasing heat source temperature and with fixed condenser temperature (represented by the lowest temperature value on each curve), the efficiency increases for a while with increasing heat source temperature, but then a maximum appears, close, but definitely below the critical temperature, followed by an efficiency decrease, supported by results obtained with other working fluids for ORC and TFC [28]. The red dots (diamond marker) show the positions of the efficiency maximum, which is always below the critical temperature (located at the end of the curves), also the dots showing that maximum efficiency shifted close to the critical temperature with a minimum value of efficiency in the and the maximum condenser temperature 302 K. However, even this new maximum is definitely below (although closer, than the previous maximum) the critical point. It is approaching the critical temperature and its value is very small; 0.066, located at 304.10 K (while the critical temperature is 304.1282 K). Figure 7 shows efficiency for the CO_2 cycle and TFC cycle in the range where this maximum appears, up to the critical temperature. The curve represents the case with condenser temperature located at 217 K, and showing that the two efficiency curves meet at one point at the critical temperature. Figure 8 shows the efficiency values at the maximum, together with the temperature of this maximum and how it decreases with increasing condenser temperature.

4. CONCLUSION

The CO_2 power cycle has reasonably good efficiency operating with low-temperature heat sources. It uses CO_2 as a working fluid, with suitable physical, chemical, environmental, and economic characteristics, compared to other working fluids. It has been known to researchers and

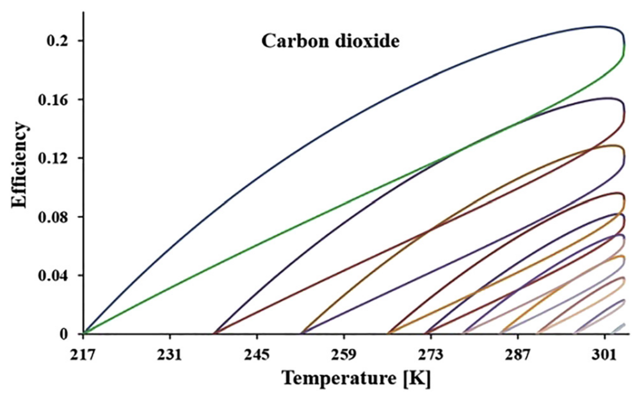

Fig. 5. Efficiency variation with heat source temperature for CO_2 and TFC cycle

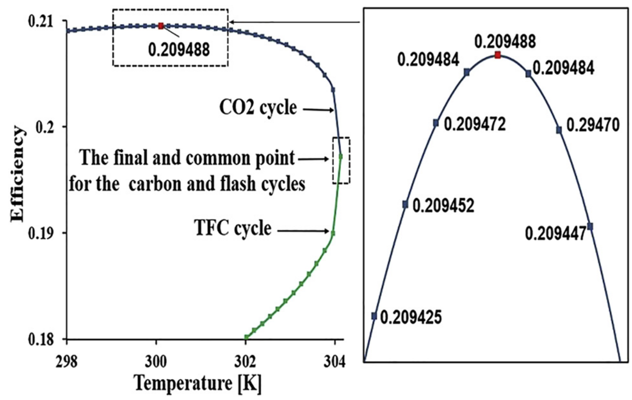

Fig. 7. Efficiency for CO_2 and TFC at Condenser temperature 217 K

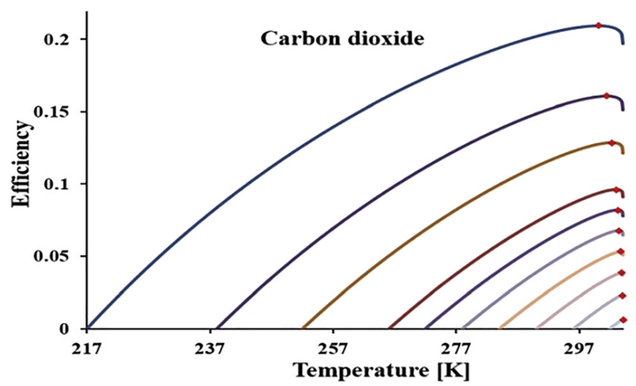

Fig. 6. Maximum efficiency (red spots) for CO_2 power cycle

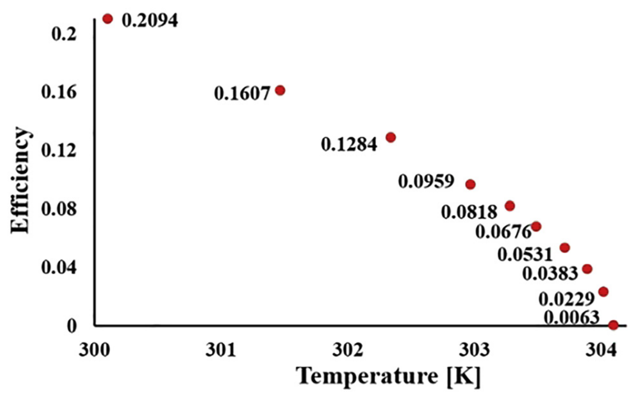

Fig. 8. The shift of the efficiency-maxima

engineers that for thermodynamic cycles the efficiency increases with the increase in the source temperature. This study showed that the efficiency of the cycle does not always increase with increasing maximal cycle temperature (and heat source temperature); close to the critical temperature; on the contrary, it reduces efficiency. It means that an efficiency maximum can exist, and the existence of this maximum should be considered upon designing subcritical CO_2 Rankine cycles with maximal cycle temperature close to the critical one. In some cases (like with solar heat), the increase of the heat source temperature goes together with the increase in installation costs. This cost increase is justified only when it is associated with proper efficiency increase; in the vicinity of the critical temperature, it is not justified. The maximum of the efficiency goes closer to the critical temperature as the condenser temperature increased, while its absolute value decreases. Also, it has been shown that the efficiency of the subcritical CO_2 power cycle higher than the efficiency of TFC and their efficiency equal at the critical point.

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