Theoretical Analysis of a Transcritical Power Cycle for Power Generation From Waste Energy at Low Temperature Heat Source

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Abstract:

The present paper reports the results obtained on a carbon dioxide transcritical power cycle using an energy and exergy analysis. The procedure consisted of modifying the inlet pressure to the turbine from 66 bar, by means of the software HYSYS®, maintaining constant each evaluated turbine inlet temperature (60, 90, 120 and 150 ºC) until the net work was approximately zero. As a result, an increase up to 25% for the exergy efficiency, and up to 300% for the energy efficiency are obtained when the inlet temperature to the turbine is risen from 60 to 150 ºC. Consequently, the analysis shows the viability of implementing this process as alternative energy, because of the possibility to recovery energy from waste heat from industrial processes.

Keywords:

Carbon dioxide, energy efficiency, exergy efficiency, power generation, waste heat.

Nomenclature

CDTPC	Carbon Dioxide Transcritical Power Cycle
$\dot{E}$	Exergy rate, kW
GWP	Global Warming Potential
h	Specific enthalpy, kJ/kg
$\dot{I}$	Irreversibility rate, kW
$\dot{m}$	Mass flow, kg/s
wne	Net specific work
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
P	Pressure, bar
$\dot{Q}$	Heat flow, kW
state point	1, 2, 2_{is}, 3, 4, 4_{is}
T	Temperature, ºC
$\dot{W}$	Power, kW

Subscripts

H	Heat source
is	Isentropic
L	Heat sink
max	Maximum
p	Pump.
t	Turbine.

0 Environmental state

Greek symbols

η Efficiency

1. Introduction

After having detected environmental problems related to the use of fossil fuels, which influence in human health and ecosystems sustainability, there is a need of covering the growing energy demand with renewable energies and with an efficient use of the energy. Energy savings must be taken into account like an alternative energy source, where it is important to avoid waste heat by using new technologies that have emerged in market, with a cost saving approach as well.

A promising technology for satisfying this is the organic Rankine cycle “ORC” [1], which allows to use waste heat or low temperature sources such as some renewables [2-3]. This operates equal to the conventional Rankine cycle and the fact of working with an organic agent as the working fluid, makes possible to recovery low enthalpy energy sources. However, some limitations must be overcome as the pinching problem that can occur in the ORC’s counter current heat exchanger evaporator, which can make unsuitable the use of this equipment because of the constant evaporation temperature of the organic substance [4].

Thus, with the purpose of employing sensitive heat sources as waste heat, some authors have proposed the use of working fluids in supercritical conditions [5-7] and/or zeotrope fluids [8-9] in the traditional Rankine cycle, obtaining therefore a better fit between the working fluid and the heat source when the heat is added. The results of an exhausted themodinamic analysis realized to a transcritical power cycle with carbon dioxide as working fluid are presented in [10]. The simulation carried out in the study showed that the proposed process is suitable for the production of useful energy utilizing heat sources between 60 and 150 ºC and, additionally, emphasizes as, with an internal heat exchanger increases considerably the energy and exergy efficiency. However, the great complexity of the plant by the incorporation of this exchanger makes that the real application depends strongly on the operability and the cost of the equipment for the high pressures required for the operation of the cycle.

Carbon dioxide (CO_2) as a natural refrigerant has been employed for refrigeration applications, but its excellent chemical and physical properties, make that is also considered as working fluid in power cycles. Some of these characteristics are the non-toxicity, environment-friendly with an Ozone Depletion Potential (ODP) of zero [11] and a Global Warming Potential (GWP) of 1 over 100 years [12], its moderate critical pressure (73.8 bar), relative inertness (for the temperature range of interest), non-explosivity, low cost and abundance in nature [3]. Moreover, its thermodynamic properties are well known and it is an environmentally friendly gas. Furthermore, it may have a great potential to improve energy conversion in a more efficient way while greatly reducing the global discharge of CO_2 in the world by using waste heat as the energy source of the system. Also, although with less influence, this is a way of capturing CO_2 when it is utilized as working fluid.

The current study presents the thermodynamic power cycle with CO_2 as working fluid that acts between subcritical and supercritical states, “a transcritical cycle”, due to its low critical temperature (31.1 ºC). Despite the lack of information available in the use of this gas as working fluid in low temperature heat source for power generation, the excellent results obtained in the incipient research incite to continue in this field. In the last few years, the transcritical cycle with carbon dioxide has been studied in steady state with solar energy as the heat source at a temperature near 200 ºC [13], and an annual dynamic performance is simulated in [7]. A small-scale prototype has been built and started up [14], showing its feasibility and achieving efficiencies between 8.8%–9.5% [15]. According to the comparative study carried out in [3], the carbon dioxide transcritical power cycle (CDTPC) showed slightly higher power output than the ORC with a R123 as working fluid. Finally, the methodology presented in [4] has been applied for the study of a CO_2 transcritical cycle supplied by a steady stream of low temperature process gases. These results have been calculated varying the high pressure of the cycle and its net power output, for fixed temperature and mass flow rate of the heat source, fixed maximum and minimum temperatures in the cycle and fixed sink temperature. Some of the researchers of [4] developed the study of the influence of the inlet temperature in the turbine (80 – 99 ºC) on the total performance of the simple cycle, maintaining the same suppositions of the mentioned research, whose results are shown in [5].

In view of what has been stated, there is great interest in the use of this fluid for the energy use of sources below 150 ºC. However, there are not many works reported in the literature on the CDTPC [3-7,10,13-15], and as it is commented in [10], it is notable the lack of studies in this theme. Therefore, the main novelty aspects provided in the present paper, are addressed to contribute in the increase of these studies showing, between others, the influence of the inlet temperature and pressure to the turbine in this process, avoiding all type of “distortions” or “noises” by external parameters on the behavior of the fluid within the cycle. Thus, in this paper, it is only calculated how much energy is required to heat the process flow directed to the turbine. Because the analysis of the heat source conditions is not our aim (i.e. it is not restricted the study to an only application), this study can be therefore used directly for analyzing its applicability with heat sources of low temperatures as waste heat from process industrial sectors, and from renewable sources, as solar, geothermal, biomass, etc.

2. Description of the carbon dioxide transcritical power cycle.

Water is the working fluid most frequently used in power systems with steam because is abundant, cheap, non-toxic, chemically stable and relatively non-corrosive. Water has also a large specific enthalpy when is vaporized at ordinary pressures in the steam generator, which allows to reduce the mass flow rate for a given net power of the cycle. The properties of liquid water and steam are also such that work ratios obtained are very small and the reheating and regeneration techniques are effective in improving the thermal performance of the plant. However, water is less satisfactory than some other working fluids with respect to other features. For instance, the critical temperature of water is 374.14 °C. Therefore, to achieve a high average temperature of heat absorption and consequently to obtain a high thermal performance, it may be necessary for the steam generator operating at supercritical pressures. This requires expensive piping and heat exchanger tubes able to support high pressures. Another undesirable feature related to water is that the saturation pressure at normal temperature of the condenser is much lower than atmospheric pressure. As a result, the air can enter the system, requiring the use of special nozzles in the condenser of feedwater heaters with degassing to remove air.

Although water has some inconvenient as working fluid, it doesn't exist any other simple working fluid more satisfying for the electrical power generation plants. Thereby, the growing interest in extracting the mechanical energy from low temperature sources has induced the development of numerous conversion techniques. One of the most promising is this CDTPC, in which a pump pressurizes the liquid CO_2, and it is injected in an evaporator to produce a vapor that is expanded in a turbine connected to a generator; finally, the exit vapor is condensed, starting the new cycle (Fig. 1).

According to the state points shown in the schematic diagram of the simple process in Fig. 1, Fig. 2 presents the CDTPC in a T-s-diagram plotted with [16] data. As an example, an ideal cycle process is shown by segments that are built from the state points 1, 2_{is}, 3 and 4_{is} marked with (○). The line segment 1-2_{is} represents an isentropic expansion with a production of output work. Heat is extracted from 2_{is} to 3 along a constant subcritical pressure line. Then, an ideal compression of the subcooled liquid from pressure at state point 3 to state point 4_{is} is carried out. Finally, the segment 4_{is}-1 represents the heat addition at constant supercritical pressure to the highest temperature of the cycle at state point 1.

In the evaporator, the fluid temperature is increasing continuously because; being above the critical point, there is no coexistence between liquid and gas phases. Thereby, fewer and more uniform differences of temperature for the transcritical cycle will be produced with respect to the thermal source, causing a considerable reduction of irreversibilities.

The previous case, but operating in conditions in which the expansion process, as well as the compression process, have certain efficiency, i.e., a cycle more closer to the reality, is represented by the segments built from the state points 1, 2, 3 and 4 marked with (○) in the same Fig. 2, which are also related to Fig. 1. State point 3 is at the lowest temperature of the cycle and above the temperature of the heat sink (receiving reservoir). Net input heat to the cycle occurs from 4 to 1 at constant pressure. Net work output is the difference between output work from state points 1 to 2 and the input work pump from state points 3 to 4.

3. Modelling of the process.

The modelling of the process, which allows analyzing the behavior and determining theoretically the performance of the CDTPC, is presented widely in [11]. However, in this section and using the first and second law of thermodynamic, it is presented a summary with the main equations, assumptions and conditions used for the theoretical study proposed. Thus, the performance of a CDTPC could be evaluated under diverse working conditions, assuming constant isentropic efficiencies of 75% for the pump as well as for the turbine, steady state conditions and no pressure drop or heat loss in the evaporator, condenser or the pipes.

The cycle's total energy efficiency is:

$$\eta = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_e} \tag{1}$$

where $\dot{W}_t$ is the power out from the turbine; $\dot{W}_p$ is the power input in the pump defined as:

$$\dot{W}_t = \dot{m} \times (h_1 - h_2) \tag{2}$$

$$\dot{W}_p = \dot{m} \times (h_3 - h_4) \tag{3}$$

and the $\dot{Q}_e$ is the heat input in the evaporator defined as:

$$\dot{Q}_e = \dot{m} \times (h_1 - h_4) \tag{4}$$

The exergy efficiency is defined as:

$$\eta_E = 1 - \left(\frac{\sum_i \dot{I}_i}{\sum_i \left(1 - \frac{T_0}{T_i}\right) \times \dot{Q}_i + \sum_i \dot{E}_{in,i}} \right) \tag{5}$$

where $\dot{I}_i$ is the exergy loss (destruction) of each component i (evaporator, turbine, condenser and pump) that can be found from an exergy balance:

$$\dot{I}_i = \sum_i \left(1 - \frac{T_0}{T_i}\right) \times \dot{Q}_i + \sum_i \dot{E}_{in,i} - \sum_i \dot{E}_{out,i} - \dot{W}_i \tag{6}$$

The term $\dot{Q}_i$ represents the flow of heat transfer at the location on the boundary where the instantaneous temperature absolute is T_i and the 0 subscripts are relative to the environmental state from which no interaction is possible. Finally, the exergy rate for each state point is expressed as:

$$\dot{E} = \dot{m}\left[\left(h - h_0\right) - T_0\left(s - s_0\right)\right] \tag{7}$$

An inlet temperature of the condensation water T_7=15 ºC, dead state T_0=15 ºC and a working fluid condensation temperature T_3=25 ºC are considered. Otherwise, a pinch point of 5 ºC is maintained between T_3 and the output temperature of the condensation water (T_8). In the heating process, the temperature T_i in the high temperature reservoir, i.e. the waste heat of the heat source, can be treated as a constant heat source at T_H. In order to transfer heat, a differential temperature ΔT=20º C is considered and therefore,

$$T_H = T_1 + \Delta T_H \tag{8}$$

In the condensation process the temperature T_i is the temperature of the low-temperature reservoir T_L. This temperature is considered to be equal to $T_L = T_7$.

The thermodynamic analysis of the CDTPC was performed using a process simulator HYSYS® (Hyprotech Co., Canada). The simulation flow diagram is the same as that presented in Fig. 1.

3. Results and discussion.

The procedure for analyzing the behavior of the CDTPC was to evaluate the thermodynamic cycle in terms of net specific work and energy and exergy efficiency, the behavior of the cycle, for different temperatures: 60, 90, 120 and 150 ºC and varying the inlet pressure to the turbine from 66 bar until the net work was nearly zero or until pressure and temperature conditions no longer allowed the fluid to be in a gaseous state for injection to the turbine. These values of temperature are selected with base on typical temperatures found in different conditions of heat waste at the industry; e.g. wet gas from drying process, hot gas from heating process, waste hot water at textile industry and other more.

3.1. Energy analysis.

Results achieved in the simulations carried out are shown in Fig. 3, where the blue tendency lines and the blue open symbols indicate the energy efficiencies (η) and the discontinuos green tendency lines represent the evaporator heat flow ($\dot{Q}_e$) or quantity of waste heat. On the other hand, in Fig. 4, the red tendency lines represent the net specific work (*wne*). Symbols represented with triangle (▲), square (■), circle (●) and the rhombus (♦) are linked with the analyzed temperatures of 150°C, 120ºC, 90ºC and 60 ºC, respectively.

A glance at Figs. 3 and 4 shows a parabola-like behavior and, hence, the existence of a maximum for both the energy efficiency and the net specific work for the four inlet temperatures to the turbine. However, it is also appreciated that these maximum points, in energy efficiency as net specific work are different. The reason for this variation is due to the different amount of heat flow required in the evaporator ($\dot{Q}_e$) for achieving the inlet conditions imposed in the turbine.

In the same Fig. 3 it is analyzed this behavior for the four inlet temperatures to the turbine and, as it was expected, the $\dot{Q}_e$ decreases as the inlet pressure to the turbine increases. This is because of the rise in the outlet temperature of the pump due to the heat given off by the pump inefficiencies, achieving thus a higher temperature to the entrance of the evaporator, which supposes a lower heat transfer requirement in this equipment.

Tables 1 and 2 have been included to summarize the results and provide a better visualization of the information exhibited in previous paragraphs, according to the values presented in Figs. 3 and 4, respectively. That is, Table 1 presents for each one of the four inlet temperatures to the turbine studied, the point of maximum energy efficiency and net specific work and the pressure corresponding to this mentioned maximum point of efficiency. Also, Table 2 shows for each one of the four inlet temperatures to the turbine studied, the maximum values for the net specific work and the values of energy efficiency and pressure corresponding to this point of maximum net specific work.

From this Fig. 3 and 4, and therefore, from Tables 1 and 2, it is inferred that the benefits from the inlet temperature of the turbine and the design criteria for the chosen cycle (maximum energy efficiency, maximum net specific work or a value between both) are different. A more exhaustive analysis of this appreciation is explained in the following sections.

3.1.1. Effect of the inlet temperature to the turbine regarding to the maximum energy efficiency.

In Fig. 3 it is observed that for a concrete pressure value, the energy efficiency increases as the inlet temperature to the turbine increases (a similar behavior can be inferred from the results in [9]). Considering the maximum value of the energy efficiency obtained (see Table 1), the effect of rising the inlet temperature to the turbine, from 60 to 150 ºC, causes an increase in the energy efficiency of 5.6% (a change from 2.5% to 8.1%, i.e., 224% more). For these same maximum points presented in Fig. 3, and comparing with the results showed for the inlet temperatures to the turbine from Fig. 4, the value of net specific work increases as well. According to Table 1, this increase is 14.2 kJ/kg (from 3.3 to 17.5 kJ/kg, i.e., 430% more). However, these points which reflect maximum efficiency energy suppose an increase in the pressure of the system in 77 bar (a change from 95 to

172 bar, i.e., 81% more); which can suppose technical problems as well as an increase of the costs.

3.1.2. Effect of the inlet temperature to the turbine regarding the maximum net specific work.

In Fig. 4 it is observed as, for other value of pressure and at equal than occurs with the energy efficiency (Fig. 3), the net specific work increases with the rise of the inlet temperature to the turbine. Thus, and according to Table 2, if the obtained maximum value is taken, the effect of increasing the inlet temperature to the turbine, from 60 to 150 ºC, causes a rise in the net specific work of 14.7 kJ/kg (a change from 3.5 to 18.2 kJ/kg, i.e., 420% more). In these points, the value of the energy efficiency rises 5.3% (from 2.4% to 7.8%, i.e., 218% more). However, these maximum points of net specific work suppose an increase in the pressure of the system in 60 bar (a change from 90 to 150 bar, i.e., 66% more). According to previous remarks, the effect of increasing the inlet temperature of the turbine, whereas it is possible, supposes an enormous increase of the energy efficiency in the system as well as the net specific work.

3.2. Exergy analysis.

In this process, the pressure ratios ($P/P_{critical}$), on one hand, must be sufficiently high as for producing an approach of temperatures as more uniform as possible between the heat source and the fluid during the heat transfer. On other hand, the pressure should not be too high to prevent the expansion in the turbine in the two-phase region, with the consequent deterioration of the latter and the decrease in performance. A fluid such as CO_2 than allows to work in the supercritical region with significantly lower temperature values and, additionally, increase the conversion of energy from a more efficient way to use waste heat or renewable as energy sources (reducing, thus, the emission of CO_2 to the environment).

An exergy analysis of the process has been carried out on the basis of the results obtained from the simulations using both the model and the suppositions and parameters mentioned in section 2. The procedure used to analyze the behavior of the CDTPC was the same as that mentioned in section 3.1, whose results were shown in Figs. 3 and 4. The realization of the process exergy analysis reveals that, contrary to what happens to energy efficiency, exergy performance (η_E) has not a maximum at the conditions in that we had evaluated the cycle, but with the increase of operating pressure of the process, rises the exergetic performance for all four inlet temperatures to the turbine, as it is shown in Fig. 5.

Figure 5 presents how, unlike what happens with the energy efficiency, the exergy efficiency has not a maximum, since that with the definition which we had proposed here, if inlet pressure to the turbine increases, the exergy efficiency also rises, (similar results are mentioned in [4]).

However, it can be concluded that, depending on the turbine inlet temperature and pressure, there can be differences in the value of the exergy efficiency; hence, they can condition the design criterion for the obtaining of a maximum energy efficiency or a maximum net specific work. A more exhaustive analysis of this appreciation is explained in the following sections.

3.2.1. Effect of the inlet temperature to the turbine on the exergy efficiency.

It can be seen in Fig. 5, that for a specific pressure, the rise in the inlet temperature to the turbine causes a decrease of the exergy efficiency of the cycle as a result of the increase of irreversibility due to the rise in heat flow required. For instance, at a pressure of 150 bar, and two different turbine inlet temperatures of 120 ºC and 150 ºC, the exergy efficiency decreases by 10% (a change from 40% to 36%, respectively).

3.2.2. Effect of the inlet pressure to the turbine on the exergy efficiency.

When a high discharge pressure is imposed on a pump (and therefore on the turbine inlet), the output temperature of the work fluid rises, causing a decrease in the irreversibility by heat flow in the evaporator, as the temperature of this fluid approaches to the heat source. Hence, a rise of the inlet pressure to the turbine causes an increase in the exergy efficiency, as presented in Fig. 5. Therefore, the increase in the inlet pressure to the turbine makes to rise the exergy performance asymptotically (above 50% as maximum for temperatures of 90, 120 and 150 °C, while for the temperature of 60 °C is limited by its maximum operating pressure to 40%).

Nevertheless, a decrease in the inlet temperature to the turbine causes a decrease in the sensitivity of the process in relation with the exergy efficiency. In other words, the lower the inlet temperature to the turbine, the lower the exergy performance gains for a slight increase in pressure. This effect evidently decreases whenever it approaches the value of its asymptote in each case. For instance, an increase of 20 bar (from 80 to 100 bar) and for the temperature of 150 ºC, causes an increase in the exergy efficiency of 110% (a change from 10% to 21%). While the same increase of pressure for the temperature of 90 ºC produces an increase of 87% (a change from 15% to 28%).

3.2.3. Irreversibilities of the process in each device.

Figure 6 presents the effects of the inlet pressure to the turbine over the irreversibilities of each one of process devices for the turbine inlet temperature at 150 ºC. Due to a similar behavior of the rest of the temperatures evaluated, only the results for this inlet temperature are represented.

Furthermore, Fig. 6 shows that when increases the inlet pressure turbine, the irreversibility of the heat flow decreases in the evaporator, as well as in the condenser and it increases at the turbine and pump; therefore we will have the conditions with which the total effects cause a maximum in exergy efficiency. The explanation for this is related to the rise in the output temperature of the fluid in the pump when a higher discharge pressure is imposed on the pump (and therefore on the inlet pressure to the turbine). This causes the irreversibility by heat flow in the evaporator to decrease when the temperature of the fluid approaches the temperature of the heat source (a hypothesis in keeping with that explained in [17]). The same happens with respect to the condenser when a lower discharge temperature is produced. However, the increase of the exergy destruction by work is less than that of these exchangers, and hence the total exergy destruction decreases in line with the increase in the inlet pressure to the turbine, being more appreciable at pressures below 150 bar, as it is showed in the same Fig. 6.

3.3. Optimum conditions of design.

Optimum conditions for the design have been considered taken into account the results presented in the previous section, these optimum points selected linked with the mean values located between the maximums of the energy efficiency and the net specific work for each of the temperatures studied. Table 3 compiles these results where the values of net specific work and the energy and exergy efficiency corresponding to this optimum pressure point. The fact of considering mean values supposes a drop of the energy efficiency of 0.15% (equivalent to 2.3%) when is compared with the maximum values whereas for the case of net specific work, a decrease of only 0.3 kJ/kg as maximum (also equivalent to only 2.3%). On the other hand, the exergy efficiency is always increasing, so that taking into account a mean design point, it is influenced in a variation (decrease or increase) of around 2% as maximum (equivalent to 6%), depending on the variable which it is compared (η or *wne*). This means that the exergy efficiency drops when is related with the one obtained with the point of maximum energy efficiency and it increases if it is compared with the point of maximum net specific work obtained.

Figure 7, provides for the four inlet temperatures to the turbine studied, the percentage of the irreversibilities of each of the devices that make up the cycle of the process proposed in the optimum operating conditions suggested (midpoint between the maximus of energy efficiency and net specific work). In general lines, it can be inferred of this figure which was discussed in section 3.2.2, i.e, a decrease in the inlet temperature to the turbine causes a decrease in the sensitivity of the process in relation with the exergy efficiency. In other words, the lower the inlet temperature to the turbine, the lower the exergy yield increases (mainly due to a larger percentage of irreversibilities in the evaporator). Additionally, it is interesting to note in this Fig. 7, the proportion of participation of the irreversibilities of each one of the devices on the proposed process, which occurs mainly in the evaporator, followed by the condenser and in a lesser way, by the turbine and pump (the irreversibilities of the pump are less than half related to the turbine).

Conclusions.

The CDTPC are suitable to be used in heat waste recovery at the industry, which permits to take advantage of hot streams that are wasted to atmosphere. It is possible to produce power from heat waste at low temperature (e.g. 60 °C), like those occurred in drying process. The technology is simple, compact and easy enough to be transported from one point to other and it is recommendable to select the operation conditions of CDTPC with base on the criterion of maximum work generated instead of maximum energy efficiency.

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TABLES.

Table 1. Main results obtained at maximum energy efficiency conditions.

Temperature, T_1 (ºC)	Parameter		
	Pressure, P_1 (bar)	η_{max} (%)	*wne* (kJ/kg)
150	172.0	8.1	17.5
120	145.0	6.5	12.0
90	120.0	4.6	7.4
60	95.0	2.5	3.3

Table 2. Main results obtained at maximum net specific work conditions.

Temperature, T_1 (ºC)	Parameter		
	Pressure, P_1 (bar)	η (%)	wne_{max} (kJ/kg)
150	150	7.8	18.2
120	128	6.2	12.8
90	108	4.4	7.8
60	90	2.4	3.5

Table 3. Optimum conditions of design.

Temperature, T_1 (ºC)	Parameter			
	Pressure, P_1 (bar)	η (%)	η_E (%)	*wne* (kJ/kg)
150	161.0	8.0	38	18
120	136.5	6.4	36	12.5
90	114.0	4.5	34	7.6
60	92.5	2.5	30	3.4

FIGURES.

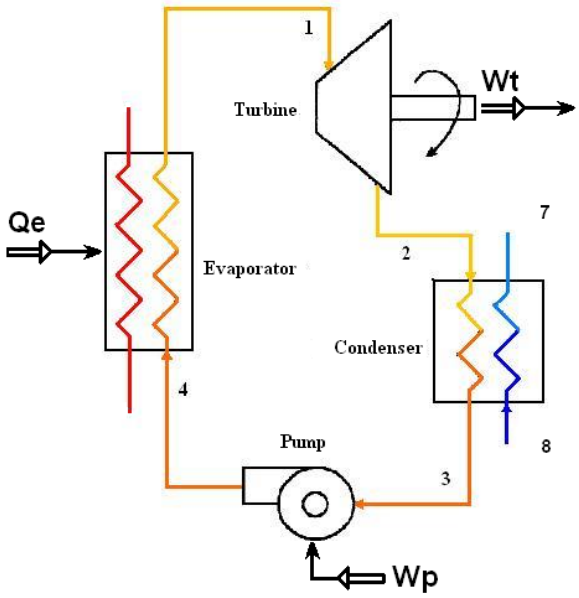


Fig. 1. Schematics diagram of the process.

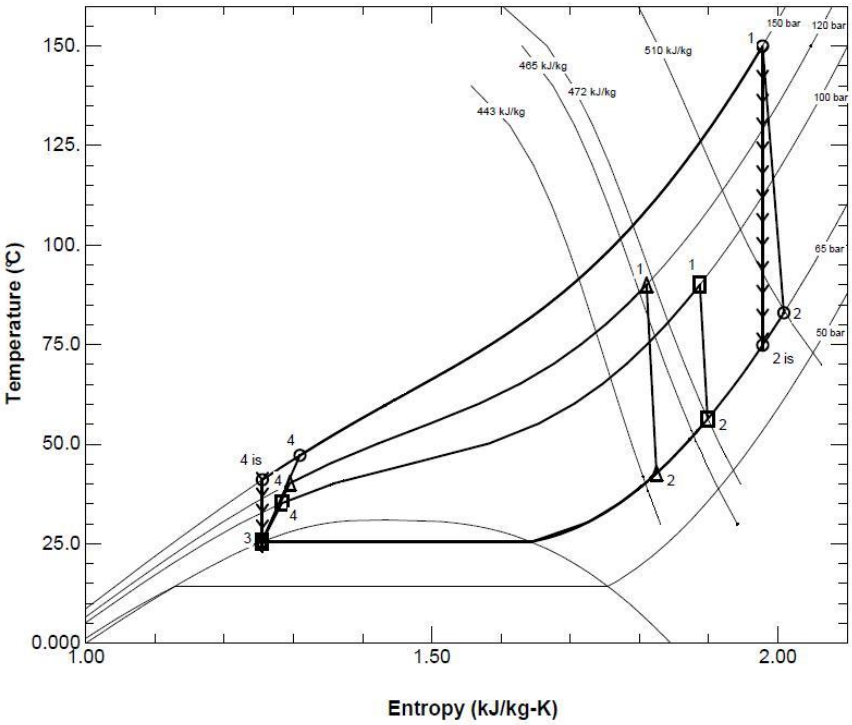


Fig. 2. T-s Diagram for differents CO_2 transcritical power cycles.

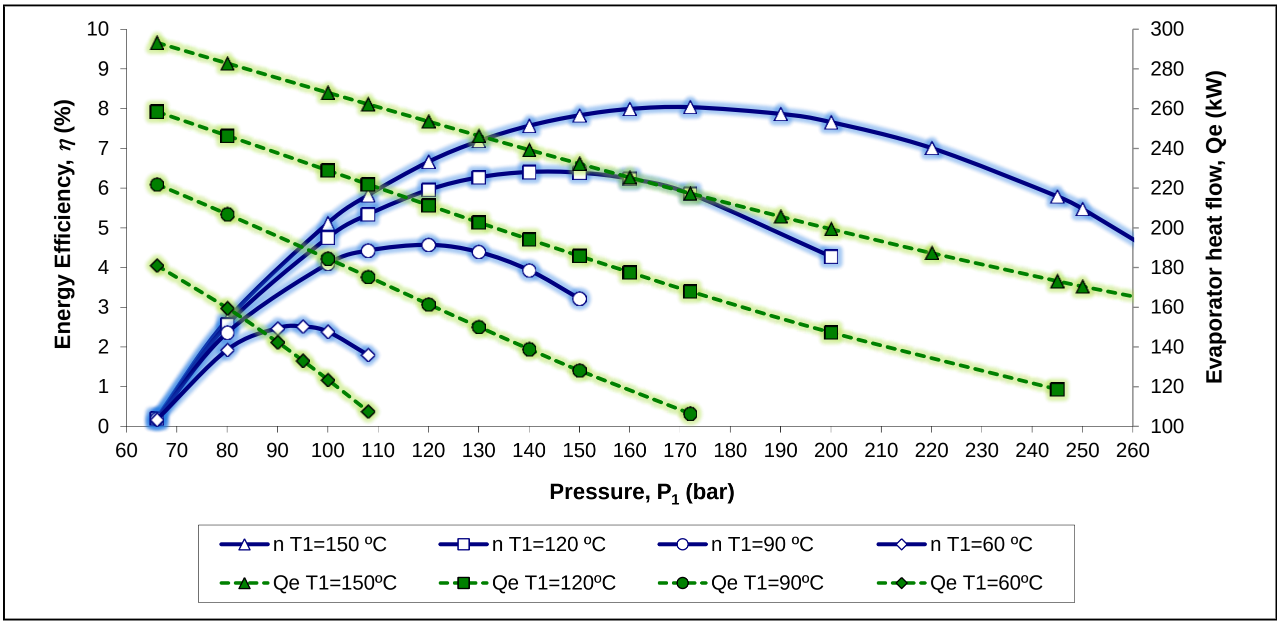


Fig. 3. Energy efficiency (open symbols) and heat flow required in the evaporator (bold symbols) vs. Pressure P_1 for T_1= 150 ºC (triangle), 120 ºC (square), 90 ºC (circle) and 60 ºC (rhombus).

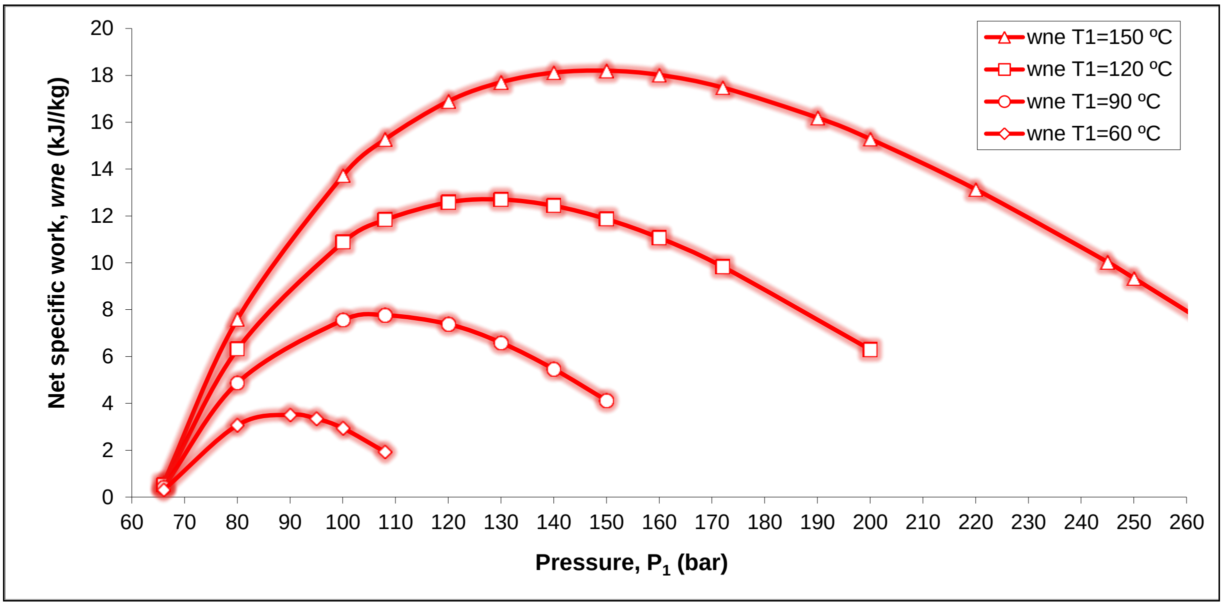


Fig. 4. Net specific work (wne) vs. Pressure P_1 for T_1= 150 ºC (triangle), 120 ºC (square), 90 ºC (circle) and 60 ºC (rhombus).

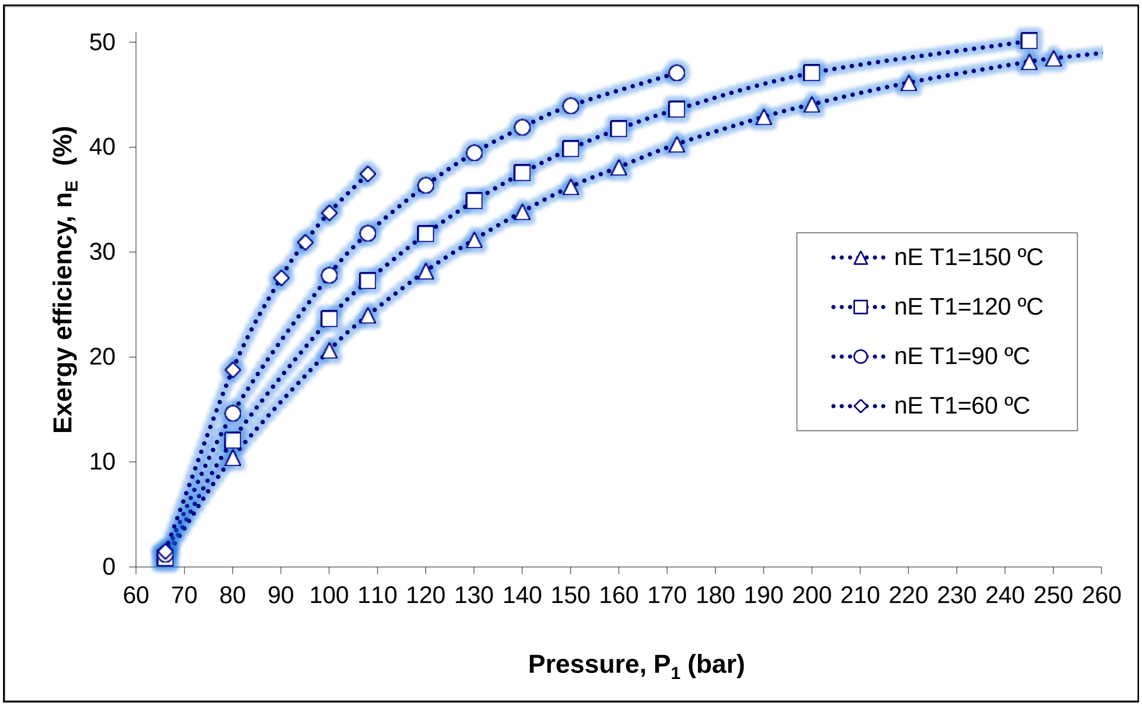
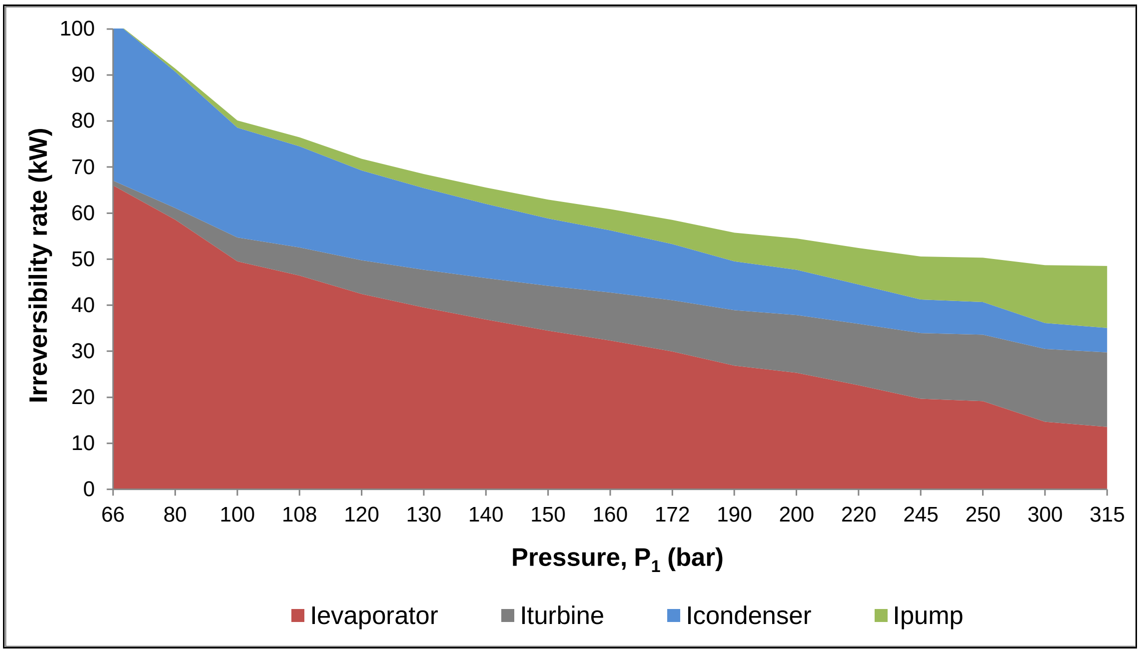
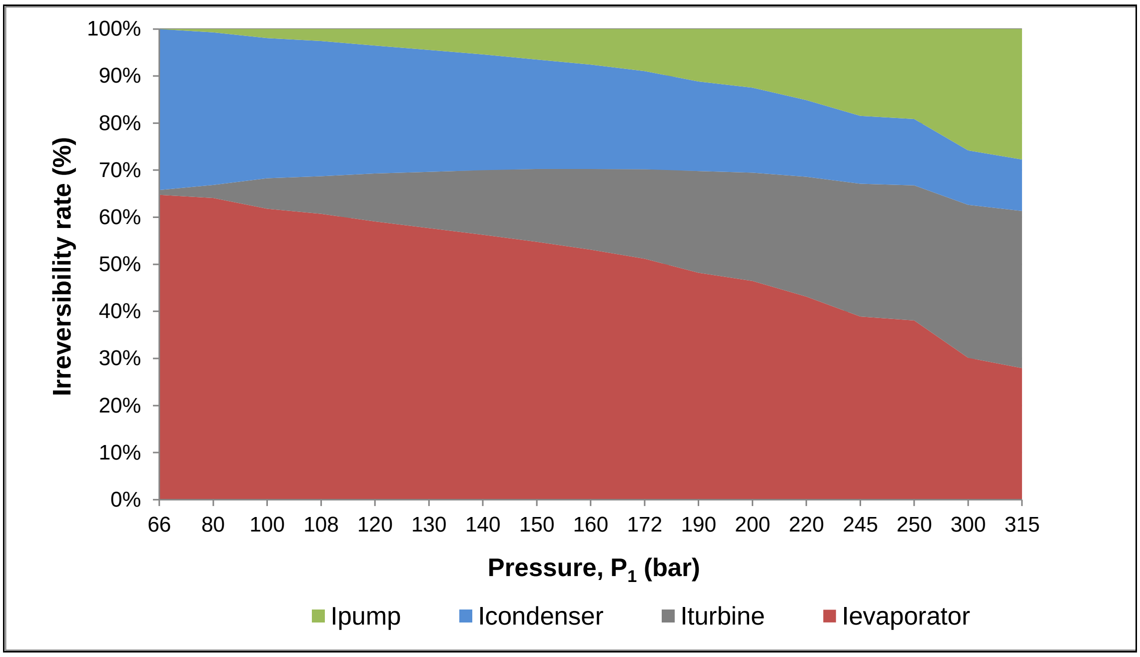


Fig. 5. Exergy efficiency vs Pressure P_1 for the studied temperatures.



a.



b.

Fig. 6. Effect of the inlet pressure to the turbine over the irreversibilities in kW (a) and percentage (b) of the process in an inlet temperature to the turbine at 150 ºC.

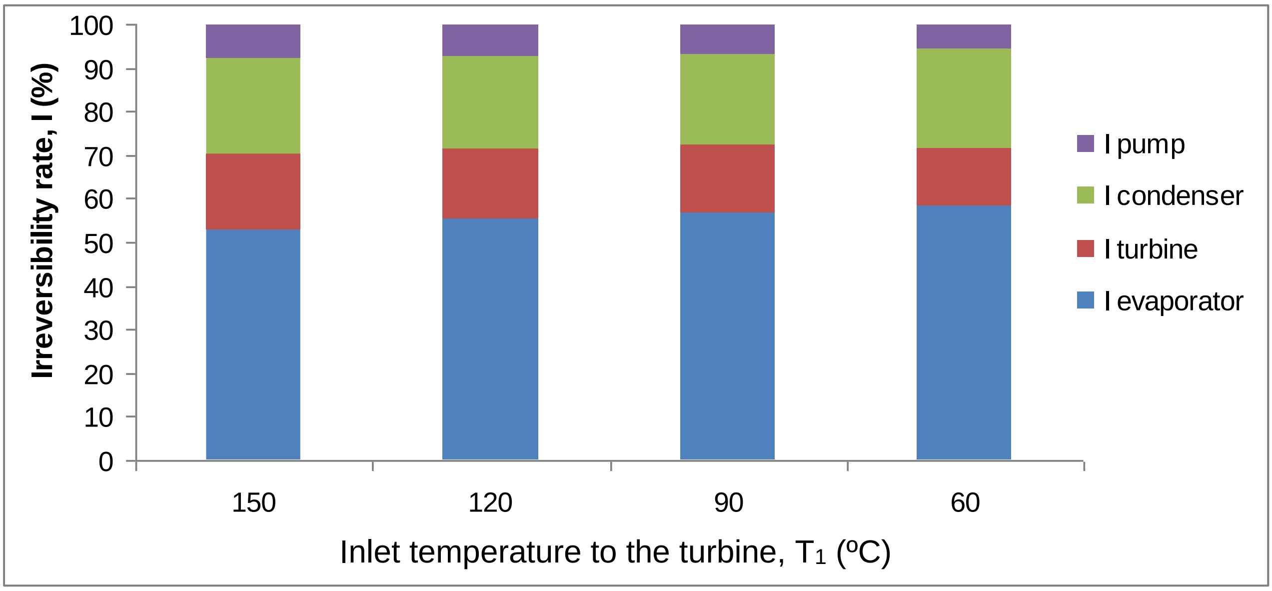


Fig. 7. Irreversibilities of the process in optimum conditions of design for the studied temperatures.