



encit 2020



18<sup>th</sup> Brazilian Congress of Thermal Sciences and Engineering  
November 16-20, 2020 (Online)

## ENC-2020-0500

# SUPERCRITICAL CARBON DIOXIDE BRAYTON CYCLES ANALYSIS FOR HIGH CONCENTRATION LINEAR FRESNEL COLLECTOR APPLICATIONS

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**Abstract.** *The present work aims the analysis of a supercritical carbon dioxide (sCO<sub>2</sub>) power cycle with heliothermic energy from a linear Fresnel collector (LFC), by the composition and numerical solution of the governing equations. Both technologies are under development and have great potential for increasing efficiency and reducing costs. Recent advances in Fresnel technology have made it possible to reach temperatures above 500 °C, enabling the use of a sCO<sub>2</sub>-based cycle. The cycle efficiency dependence on various parameters was analyzed, such as the turbine inlet temperature and pressure, the compressor inlet temperature and pressure, the by-pass ratio in the recompressor, as well as the use of recompression, reheat and intercooler. For the regenerators modeling, the minimum pinch method was used to determine the heat exchange. The performance of the cycle was analyzed for turbine inlet pressures from 160 to 340 bar and temperatures from 350 °C to 570 °C. The EBSILON Professional software was used to estimate the heat losses and efficiency of a Fresnel collector. The simple cycle showed a performance about 14% lower than the recompression cycle for temperatures above 350 °C, which limits its practical use. The use of reheating in the recompression cycle showed a 5% average efficiency increase, rising the thermal efficiency from 0.42 to 0.44 for a turbine inlet temperature (TIT) of 500 °C and pressure of 200 bar. In addition, with the use of intermediate cooling in the main compressor, a gain of up to 2% can be achieved, resulting in a thermal efficiency of up to 45% in these conditions. There was a linear gain in the power cycle thermal efficiency with the increase in TIT for all cases. However, the collector efficiency declines exponentially with increasing temperature due to heat losses. Thus, the combined peak efficiency of the collector and the power cycle in relation of TIT under the studied conditions occurs at approximately 550 °C for a perpendicular solar incidence of 1000 W/m<sup>2</sup>.*

**Keywords:** *Supercritical carbon dioxide, power cycle, Brayton cycle, concentrated solar power, Fresnel.*

## 1. INTRODUCTION

Concentrated solar power (CSP) has shown to be a great option to sustainably meet the current high energy demand. Solar concentrators combined with power cycles for generating electricity have been extensively studied over the past few years, showing the potential to compete with photovoltaic systems (Brun et al., 2017). A promising part of this evolution is the use of special working fluids, such as supercritical carbon dioxide (sCO<sub>2</sub>), which enable more efficient, compact and low-cost equipment compared to steam turbines (White, 2016). This advantage expands the field of use for smaller scale systems, without prejudice to use in large scale systems (> 1000MWe) (Brun et al., 2017).

Linear Fresnel Collectors (LFC) have a great advantage of simplicity and cost compared to other conventional forms of heliothermic concentrators, such as the parabolic trough and the solar tower (Duffie and Beckmann, 2013). Although historically known for having lower concentration factors than other types, Fresnel technology has recently made significant advances. Tests carried out in 2009 at a 1.4 MW commercial-scale solar collector at Novatec's PE1 power plant, in Murcia, Spain, proved that with a new receiver design it is possible to reach temperatures above 500 °C (Morin et al., 2014), making possible the application in a Brayton cycle with supercritical carbon dioxide.

This work aims to model, analyze and optimize the main constructive and thermodynamic parameters that influence a gas power cycle operating with sCO<sub>2</sub> heated with heliothermic energy from a high-concentration Linear Fresnel Collector capable of reaching 500 °C, in order to determine the conditions of greatest efficiency and cost-effectiveness.

## 2. THE SUPERCRITICAL BRAYTON CYCLE AND THE CONCENTRATED SOLAR POWER

The use of supercritical CO<sub>2</sub> in a Brayton cycle is an innovative concept for converting thermal energy into electrical energy. Other supercritical fluids, such as helium and nitrogen, have been studied for this purpose, however, CO<sub>2</sub> has been presented as the best candidate so far (Invernizzi, 2013). One of the main reasons is the low critical temperature of carbon dioxide, of approximately 31 °C at a pressure of 73.8 bar (Brun et al., 2017), which makes heat rejection at room temperature viable. Other factors such as safety, toxicity, availability, thermal stability, corrosion and cost were also evaluated and no other fluid was considered to be a better candidate for terrestrial applications (Invernizzi, 2013).

Supercritical CO<sub>2</sub>-based Brayton cycles have a high-density working fluid as they operate at high pressure. Furthermore, they have a low expansion ratio and, consequently, few compressor and turbine stages. These two features enable compact and potentially cheaper equipment. For example, a 1 MW cycle would have only a single-stage compressor with 10 cm in diameter and a turbine with 4 stages or less (Fleming, 2013). However, the supercritical cycle requires regenerators that usually take up a lot of space and have a significant cost.

The supercritical CO<sub>2</sub> Brayton cycle can be cheaper and safer than the Organic Rankine Cycle (ORC), which until today is one of the best alternatives for heat sources from 100 to 370 °C (Marchionni et al., 2017; Hung, 2001). For temperatures higher than this (390 to 575 °C), the sCO<sub>2</sub> cycle is an excellent option, and this is the usual temperature range of most high-concentration solar power systems (Duffie and Beckmann, 2013).

### 2.1 The recompression Brayton cycle

The process consists of recompressing a portion of the working fluid, without it passing through the main cooler and the main compressor. Several studies indicate that the use of recompression increases the efficiency of the supercritical cycle (Brun et al., 2017; White, 2016; Invernizzi, 2013). This cycle can be more efficient than a water Rankine cycle, when the inlet temperature of the turbine is greater than approximately 425 °C (White, 2016). Another important factor is that sCO<sub>2</sub> cycles can operate without significant use of water in the main cooler or even only with air cooling. This feature facilitates to be used in regions with water scarcity, unlike steam turbines (Brun et al., 2017). The main downside is the need for an additional compressor and regenerator.

Figure 1 shows the diagram of a simple cycle and a recompression cycle. There is a by-pass before the cooler that leads to the secondary compressor. The fluid returns to the main cycle before the high temperature regenerator (HTR). Although this configuration increases the compression work required, there is an increase in the amount of working fluid that passes through the cycle for the same thermal energy input (Park et al., 2018), due to the increased effectiveness of the regenerators.

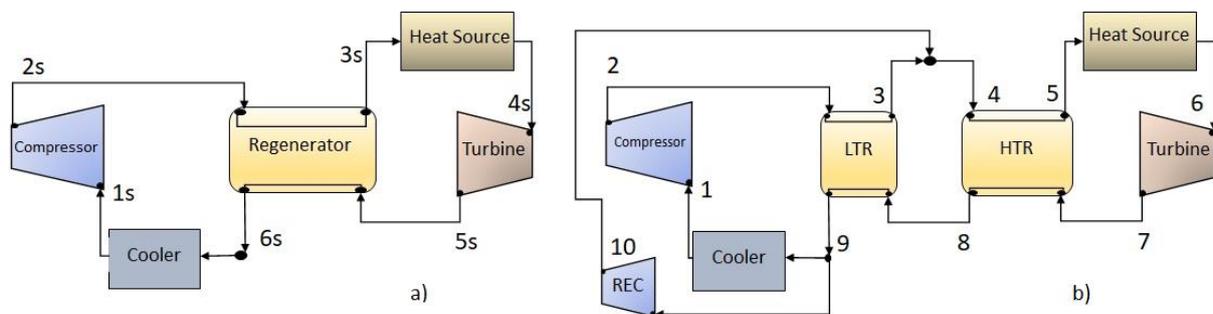


Figure 1. a) Simple Brayton cycle (SMP) diagram (with regeneration). b) Recompression Brayton cycle diagram (RC).  
 Rec: Recompressor. LTR: Low temperature regenerator, HTR: High temperature regenerator.

With the low-pressure region (points 1s, 5s, 6s, 1, 7, 8 and 9) of the cycle close to the critical point, there is a compression work reduction. However, there is a side effect that implies reducing efficiency for this condition, which is the high specific heat of the fluid at high pressure near the critical point (points 2 and 2s), as shown in Fig. 2. This reduces the effectiveness of the regenerator, because the low-pressure stream does not have enough energy to heat the high-pressure stream. The use of recompression can mitigate these effects, because in the low temperature regenerator, due to the by-pass, there is a greater mass flow passing through the hot side than the cold side, compensating for the specific heat difference. When the streams are mixed, the fluid has specific heat low enough for the heat exchange to be effective.

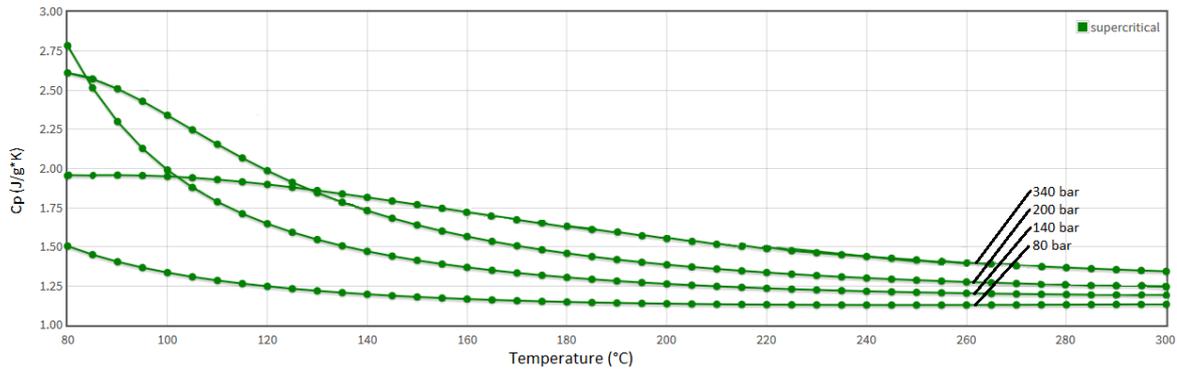


Figure 2. Temperature relation to sCO<sub>2</sub> specific heat near critical point. Adapted from NIST (2018).

## 2.2 The reheat and intercooler Brayton cycle

The use of reheating between stages of the turbine has the potential to increase the expansion work and the efficiency of the cycle. However, it increases the complexity of the power block. This configuration can make it difficult to use direct heating of the working fluid, as it requires an additional heating stage.

Similarly, the use of a cooler between the stages of the main compressor reduces the required compression work, with the disadvantage of requiring an extra stage in the compressor and an additional cooler. Figure 3 shows the diagram of a cycle with recompression and reheating in one stage and a cycle with recompression, reheat and intermediate cooler (intercooler), also in one stage. Many of the cycles analyzed in this work were first studied and discussed by Angelino (1969).

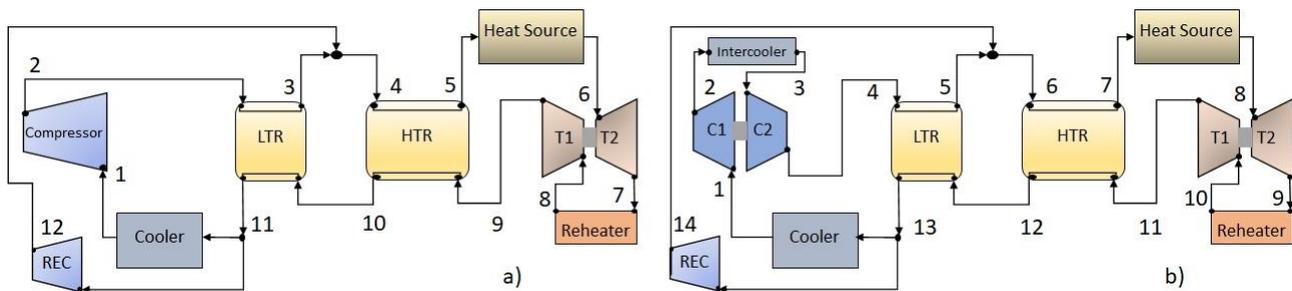


Figure 3. a) Diagram of the recompression Brayton cycle with reheat in one stage (RCRH). b) Diagram of the recompression Brayton cycle with reheat and intercooler in one stage (RCRHIC).

## 2.3 Concentrated solar power and the linear Fresnel collector

Concentrated solar power represents about 1% of the solar energy used worldwide in 2019 (about 6.2 GW), the rest is from photovoltaic systems (REN21, 2020). However, CSP systems are still preferable for large-scale generation (Brun et al., 2017). Fresnel collectors still have a small portion of the CSP systems used, generally with applications for temperatures from 200 °C to 300 °C. Due to its simplicity, it has the advantage of being cheaper and more durable (Abbas et al., 2013). LFCs have a fixed absorber, which reduces sealing and bearing costs in relation to the parabolic trough collector. The absence of moving parts in the absorber facilitates the use of direct heating of the working fluid, even gases at high pressure. In this way, the need to use a heat transfer fluid is eliminated and the efficiency of the plant can be increased. However, to use direct CO<sub>2</sub> heating, the collector must withstand a pressure of approximately 200 bar. It may be economically unfeasible to store CO<sub>2</sub> at this pressure, thus being subject to variations in solar incidence. These characteristics suggest the use of an intermediate heat transfer fluid with a thermal storage system (TES) or a backup system with fossil fuels. In the case of indirect heating, it is possible to use oils or molten salts as heat transfer fluid. In this way, the power cycle is separated from the solar field and it may be possible to use higher pressures.

## 3. CALCULATION PROCEDURE

This analysis was made by the composition and numerical solution of the governing equations for the turbine inlet pressure (TIP) variation from 160 bar to 340 bar and the compressor inlet pressure (CIP) from 74 bar to 94 bar, with intermediate pressures from 84 bar to 220 bar. Pressure losses in the circuit were neglected. The thermodynamic properties of CO<sub>2</sub> were obtained with the NIST Chemistry WebBook Software (NIST, 2018), which uses a series of correlations described by Span and Wagner, 1996. The EBSILON Professional software was used to estimate the heat

losses and efficiency of a Fresnel collector. The results were obtained in the stationary condition and in the design condition.

### 3.1 Regenerator’s modeling

The modeling of the regenerators was done by discretizing their length in small parts of the same heat exchange, considering the minimum temperature difference of the streams (pinch). This approach proved to be preferable in relation to simply defining the effectiveness of the regenerator, as already noted by White et al. (2014). Figure 4 shows the temperature distribution of the regenerator in the simple cycle for the 200 bar outlet compressor stream and the 80 bar turbine outlet stream. There is a large temperature difference between the streams at the regenerator hot end, of approximately 90 °C. In this case, the cold stream is heated to up 308 °C.

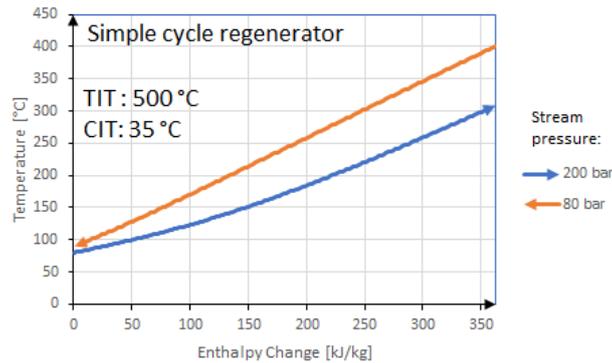


Figure 4. Temperature distribution in the simple cycle regenerator ( $bpr = 0.0$ ).

## 4. RESULTS AND DISCUSSION

With recompression, an increase in efficiency is obtained by increasing the by-pass ratio, due to the increased effectiveness of the regenerators. However, this increase is limited by the internal pinch point. An excessive large heat exchanger would be required to indefinitely decrease the stream’s temperature difference. Figure 5 shows the temperature distribution in the regenerators for two by-pass ratios ( $bpr$ ). For a  $bpr$  of 0.37, which corresponds to an internal pinch of 10 °C, the cold stream is heated up to 365 °C. As the greatest heat exchange effectiveness occurs with the greatest temperature difference in the streams, the union of the recompression stream and the LTR stream should occur where both have the same temperature. In this way, a sudden variation in temperature is avoided. Otherwise, there would be a “step” in the curve and an effectiveness loss.

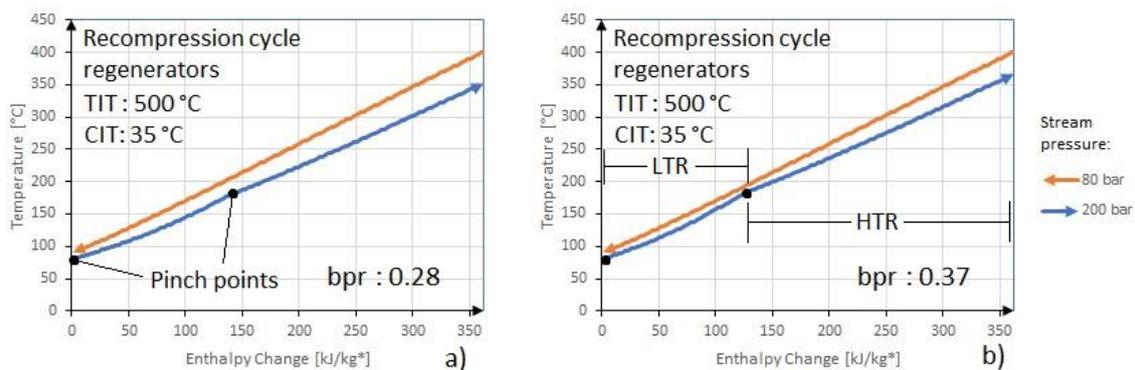


Figure 5. Temperature distribution in the recompression cycle regenerators for: a) A by-pass ratio of 0.28. b) A by-pass ratio of 0.37. \* refers to 1 kg of the hot stream.

As the curves of the cold stream have a concave upwards, a pinch point occurs at the union of the curves. This point coincides with the end of the LTR and the beginning of the HTR. In this way, there are two pinch points to be considered, one at the coldest end and the other at the union of the regenerators, where there is the junction of the recompression stream. For higher pressure ratios, a third pinch point can occur in the LTR. The by-pass ratio is not a function of turbine inlet temperature (TIT). Thus, by setting the compressor inlet temperature (CIT) at 35 °C and the minimum pinch at 10 °C, the by-pass ratio is constant for each TIT.

#### 4.1 Cycle performance and behavior analysis

An analysis of the cycle efficiency and other properties were made in relation to the compressor inlet temperature and for the turbine inlet temperature for high-concentration LFC temperatures, between 350 °C and 570 °C, for several pressure ratios. The minimum difference between the temperature of the cold stream and the hot stream in the heat exchanger (pinch) was set at 10 °C. Isentropic efficiency was set at 0.85 and 0.92 for the turbine and the compressor, respectively. Intermediate pressures were adjusted to the condition of greatest efficiency.

The influence of the compressor inlet temperature was analyzed for several conditions. Figure 6 shows, in a timeless way, the energy variation in the various components in the form of work or heat per kilogram of working fluid circulating in recompression cycle and in the recompression reheat intercooler cycle with pressure ratio of 300/90. The reheater inlet pressure (RIP) is 160 bar and the intercooler inlet pressure (IIP) is 120 bar in this case.

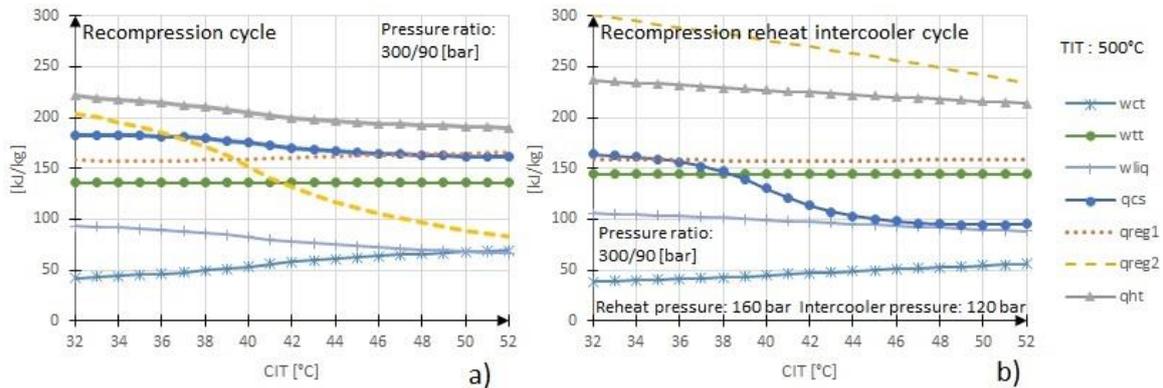


Figure 6. CIT relation to energy variation in the various components in the form of work or heat per kilogram of working fluid in: a) recompression cycle. b) recompression reheat intercooler cycle. wct: Total compression work. wtt: Total turbine work. wliq: Liquid work. qcs: Cooler heat output. qreg1: Enthalpy change in LTR. qreg2: Enthalpy change in HTR, qht: Hot source heat input plus reheater input.

The turbine work remains constant as it depends only on the TIT and the pressure ratio. By increasing CIT, there is a decrease in the heat input and in the liquid work due to the increase of the compression work. In the reheated cycle, there is a slight increase in the turbine work and in the hot source heat input. The regenerated heat increases in a greater proportion. With the use of intercooler, a significant reduction in the compression work is observed.

The variation of thermal efficiency in relation of the compressor inlet temperature was analyzed for several cycle configurations for a 500 °C TIT, as shown in Fig. 7. For the simple cycle, the maximum efficiency does not occur for the lowest compressor inlet temperature, as expected. The simple cycle benefits from a higher CIT, despite increasing the required compression work. This occurs due to the increase in temperature and to the decrease in the specific heat of the high-pressure stream, providing a higher outlet regenerator temperature. The simple cycle maximum efficiency occurs for a CIT of 35 °C to 37° C in the studied conditions, for TIT above 350 °C. It is observed that the use of recompression significantly increases the efficiency of the cycle.

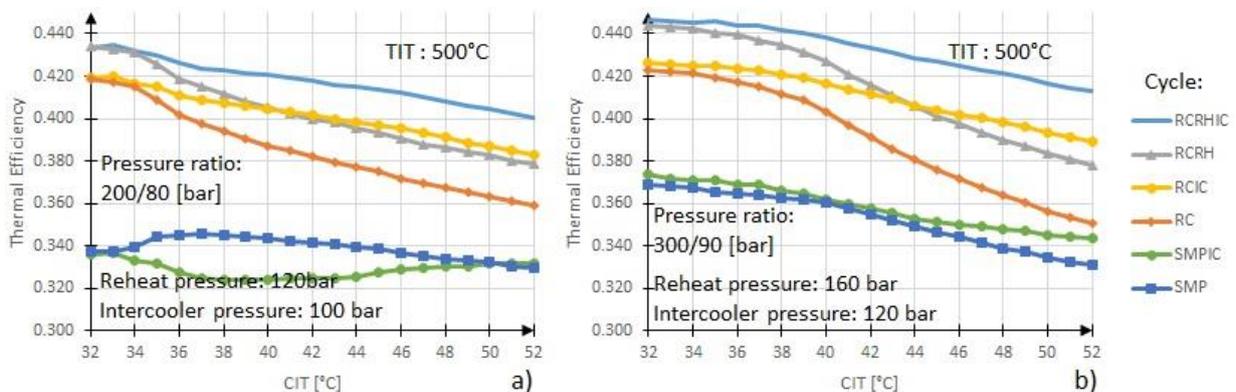


Figure 7. CIT relation to thermal efficiency for two pressure ratios. a) 200/80. b) 300/90. SMP: Simple cycle. SMPIC: Simple intercooler cycle. REC: Recompression cycle. RCIC: Recompression intercooler cycle. RCRH: Recompression reheat cycle. RCRHIC: Recompression reheat intercooler cycle.

In the recompression cycle for an 80 bar CIP, there is a slight drop in efficiency for CIT up to 35 °C, followed by a sharp drop for higher temperatures. For a 90 bar CIP, the efficiency drop is more pronounced after 38 °C. It is observed that the use of the intercooler is more significant for high CIT, and its use should be considered in hot or air-cooled regions. However, its use does not always show efficiency gains, as in the simple cycle with a pressure ratio of 200/80. The optimal intermediate pressure of the intercooler is lower than the optimal intermediate pressure of reheating.

Intermediate cooling in the main compressor reduces the compressor outlet temperature (from 80 °C to 46 °C in the 200/120/80 cycle at TIT of 500 °C and from 69 °C to 49 °C in the 300/160/90 cycle at TIT of 500 °C). This affects the performance of the entire cycle, also reducing the compression work of the secondary compressor. This happens because the high-pressure stream enters at lower temperature in the LTR and act lowering the temperature of the low-pressure stream further. This causes more heat to be exchanged and reduces the temperature at the inlet of the secondary compressor. Its use increases the cycle efficiency by 2% on average.

The influence of the turbine inlet temperature was analyzed for several cases. Figure 8 shows, in a timeless way, the behavior of the cycles with recompression only and of the cycle with recompression, reheat and intercooler in relation of TIT for a CIT of 35 °C. It is observed that the influence of TIT is approximately linear for all components.

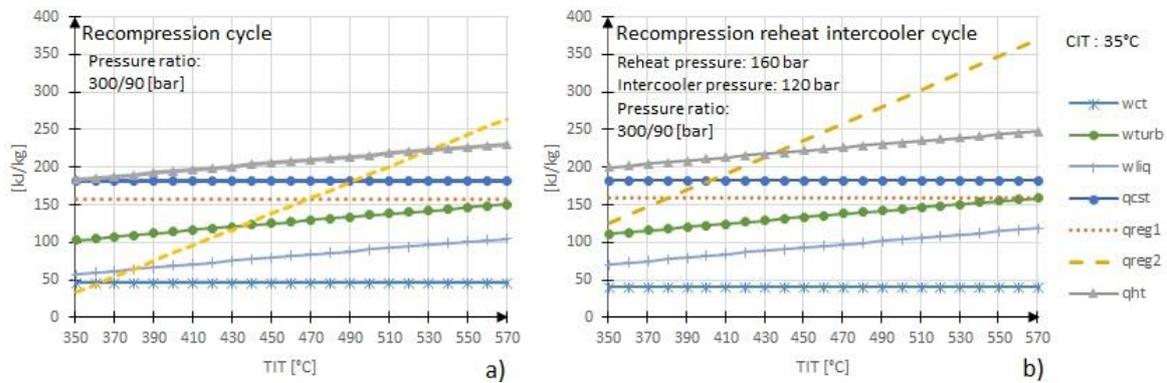


Figure 8. TIT relation to energy variation in the various components in the form of work or heat per kilogram of working fluid in: a) Recompression cycle. b) Recompression reheat intercooler cycle. wct: Total compression work, wturb: Total turbine work, wliq: Liquid work, qcst: Cooler heat output plus intercooler heat output, qreg1: enthalpy change in LTR, qreg2: enthalpy change in HTR, qht: hot source heat input plus reheat input.

With the use of intercooler, the heat lost in the cooler of the cycle decreases. However, it is compensated by the heat lost in the intercooler, making the total energy lost in the two cycles similar. With the use of reheating, there is an increase in the turbine outlet temperature and more energy is exchanged in the HTR. This causes the high-pressure stream to come out at a higher temperature. In this way, there is a reduction in the heat input from the main source. However, the total heat input increases due to the reheater.

The reheating between the turbine stages considerably increases the efficiency of the cycle with recompression, by about 5% on average. For a 500 °C TIT and intermediate pressure of 160 bar in a cycle with a pressure ratio of 300/90, the work generated by the turbine increases by about 6%. This represents a 9% increase in net work for just 4% higher heat input. The use of recompression and reheating act in a similar way, increasing the energy regenerated in the cycle and consequently increasing the thermal efficiency. However, unlike recompression, reheating does not increase the effectiveness of the regenerators, but increases the temperature of both streams in the regenerator, as shown in Fig. 9.

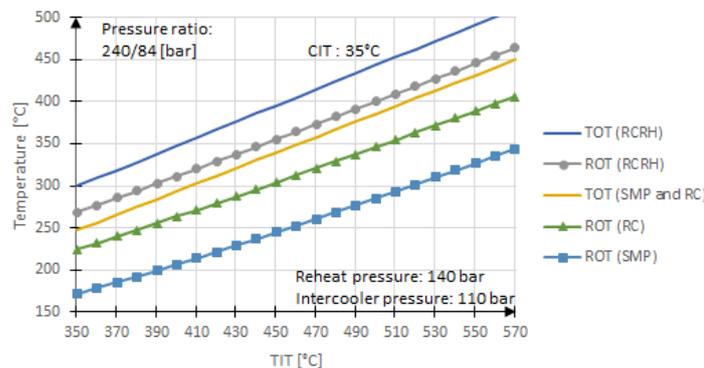


Figure 9. Stream temperature at the hot end of HTR in relation of TIT for simple cycle, recompression cycle and reheat-recompression cycle. TOT: Turbine outlet temperature (Same as HTR low-pressure hot stream inlet temperature). ROT: Regenerator outlet temperature of high-pressure stream.

The turbine outlet temperature (TOT) is equal to regenerator inlet temperature, which represents the temperature of the low-pressure stream at HTR inlet. The regenerator outlet temperature (ROT) is the temperature of the high-pressure stream at HTR outlet before the main heat source. It is observed that the use of recompression reduces the temperature difference of the cold and hot streams in relation to the simple cycle. In the reheated cycle, there is a slight loss of effectiveness of the regenerators, shown by the greater temperature difference between streams. In the meantime, the temperature of both streams is increased.

#### 4.2 Optimal cycle configuration analysis

Figure 10 shows the variation of the efficiency of the simple, recompression, reheat and intercooler cycles in relation of the turbine inlet temperature. It is observed that the efficiency increases gradually with the increase of TIT in all cycles. Simple cycles are unlikely to be used for temperatures above 350 °C. The CIP chosen for each TIP is not the optimum for all cases, since the optimal CIP varies for each cycle configuration and for each turbine inlet temperature. Thus, an average CIP was chosen according to the TIP, which presents satisfactory performance for all cases.

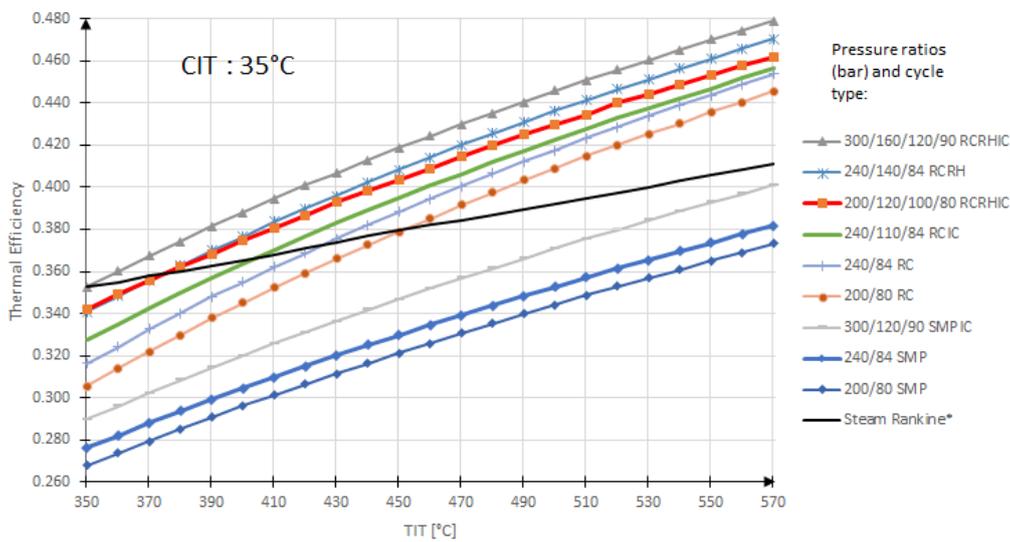


Figure 10. TIT relation to thermal efficiency for several cycle configurations and pressure ratios. Pressure format: TIP/RIP/IIP/CIP. \* Steam Rankine cycle with a single reheat, adapted from White (2016).

The reheated cycles proved to be much more efficient for all temperatures, being the main candidates for high performance installations. The 200/80 recompression cycle has a satisfactory performance for low cost application and indicates that it may be possible to use direct heating for a TIT of 500 °C, with thermal efficiency of approximately 41% with a by-pass ratio of 0.37.

The ideal CIP for the Supercritical CO<sub>2</sub> cycle for a 500 °C TIT ranges from 78 to 86 bar for a CIT of 35 °C. For a CIT of 32 °C, the ideal CIP ranges from 76 to 80 bar, depending on the cycle used. Other CIPs have not shown satisfactory performance for this temperature. Figure 11 shows the CIP influence in thermal efficiency and the optimal CIP for two cases, for 35 °C and 33 °C CIT.

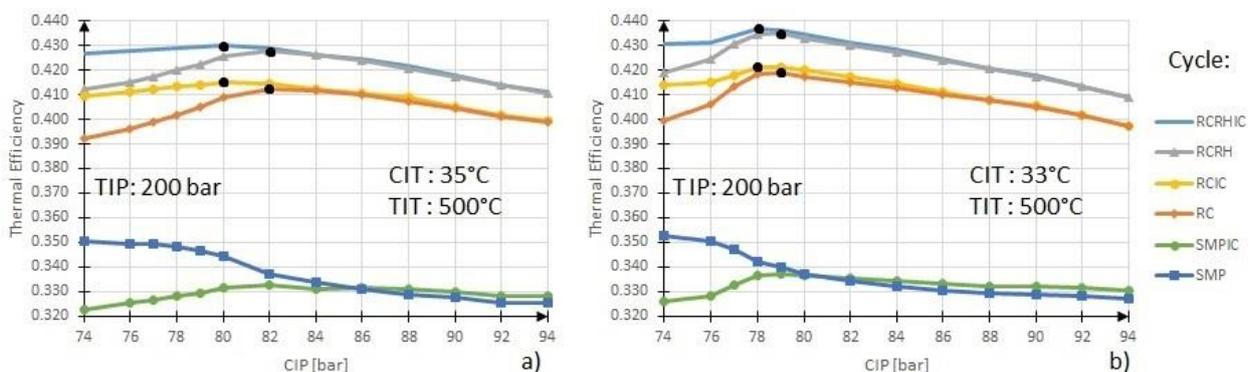


Figure 11. CIP Optimization a) CIT of 35 °C. b) CIT of 33 °C.

The optimal CIP varies for each cycle, TIP and CIT used. when CIT approaches the critical temperature, the optimal CIP approaches the critical pressure. However, when approaching the critical temperature, possible risks of liquefaction must be considered.

The cycles defined as the best candidates according to TIP and CIP for a 500°C TIT application are shown in Fig. 12, divided into 4 levels of complexity (recompression only; recompression and intercooler; recompression and reheating; recompression, reheating and intercooler).

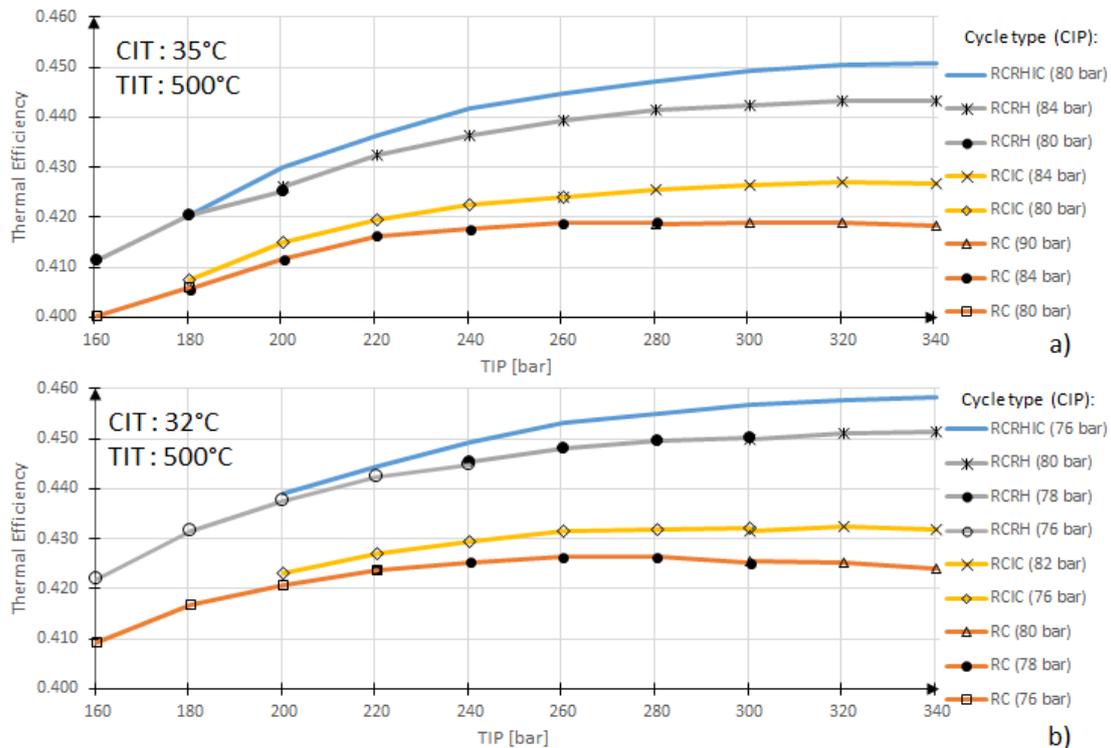


Figure 12. Optimal cycle configurations for a TIP range from 160 bar to 340 bar, for a TIT of 500 °C and a) CIT of 35 °C. b) CIT of 32 °C.

For a CIT of 35 °C, the recompression, reheating and intercooler cycle proved to be the most promising, with thermal efficiency of approximately 45% for a TIP of 300 bar and 43% for a TIP of 200 bar. For a 160 bar TIP, the recompression cycle has 40% thermal efficiency and the recompression reheat cycle has 41% thermal efficiency. The use of intercooler for this pressure is not advantageous. However, the use of intercooler is advantageous for TIP above 180 bar for all cycle configurations (except simple cycle). In the cycle with recompression only, there is no significant gain in efficiency for TIP greater than 260 bar for this TIT. For a CIT of 32 °C, the thermal efficiency is about 2% higher in all cases.

### 4.3 Collector efficiency and combined efficiency

The heat losses and the efficiency of a Fresnel collector was estimated using EBSILON Professional software. Figure 13 shows the collector efficiency variation and the combined cycle-collector efficiency variation in relation of TIT, for a normal direct irradiance of 1000 W/m<sup>2</sup> and perpendicular incidence. For modeling, data from the Fresnel SPG type 3 linear collector, manufactured by Solar Power Group, were used. Another comparisons have sometimes been difficult to make without access to what continues to be proprietary commercial information.

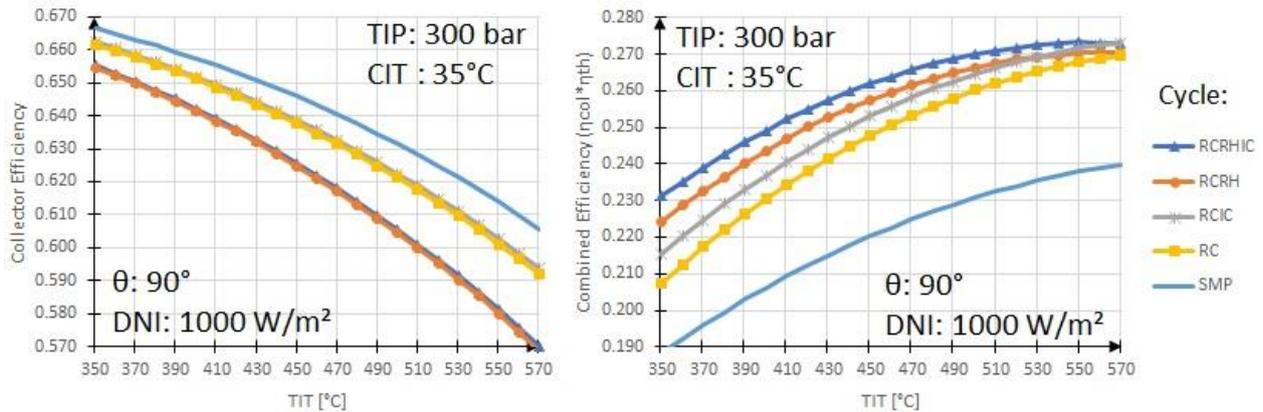


Figure 13. Collector efficiency variation and the combined cycle-collector efficiency variation in relation of TIT.

The collector efficiency varies depending on the cycle due to the variation of the temperature of the heat transfer fluid (HTF) at the entrance of the collector. Higher temperatures, as in reheating cycles, cause the average collector temperature to be higher and have more heat losses. In the simple cycle, the efficiency of the collector is higher because the average temperature of the HTF in the collector is lower. In all cases, the efficiency of the collector decreases with the increase in TIT.

The combined peak efficiency of the collector and the power cycle in relation to TIT occurs at approximately 550 °C, with a value of 27%. At this temperature, the advantage of using reheating is reduced due to higher thermal losses. For lower cycle pressures, the collector efficiency is slightly reduced. For a TIP of 200 bar the combined efficiency peak is approximately 26% for a TIT of 530 °C. However, considering the other thermal losses characteristic of each CSP plant, this optimum temperature may be lower.

## 5. CONCLUSION

The analysis showed that the low temperature regenerator (LTR) should be designed in such way that the outlet temperature be close to the recompressor outlet temperature, for the greatest regenerator's effectiveness. In addition, it's shown that the optimum compressor inlet pressure for most cases varies from 78 to 86 bar. The recompression cycle only indicates little efficiency gain for TIP above 260 bar. On the other hand, the more complex cycles show an efficiency gain by increasing the turbine inlet pressure (TIP) by up to 320 bar. The simple cycle has a thermal efficiency about 14% lower than the cycle with recompression for temperatures above 350 °C, which limits its applicability.

The use of intercooler in the main compressor reduces the compression work of the cycle, including indirectly in the secondary compressor, presenting efficiency gain in most cases by 2% on average. Its use is even more significant for high CIT, essential for air-cooled cycles or hot regions. However, the use of intercooler is not advantageous for TIP of 160 bar or less. The cycle with reheating between stages of the turbine proved to be essential to achieve high performance in the cycle with recompression, increasing thermal efficiency by 5% on average and its use should be considered.

Studies have shown that the supercritical CO<sub>2</sub> cycles are favorable for an application with a temperature of 500 °C in a linear Fresnel collector. The thermal efficiency of 41% in a cycle with recompression and a maximum pressure of 200 bar indicates the possibility of direct heating of the working fluid. Furthermore, it is possible to achieve a thermal efficiency of 45% with indirect heating and a TIP of 300 bar in the most complex cycles.

There is a linear gain in thermal efficiency of the power cycle with the increase in turbine inlet temperature for all cases. However, the efficiency of the collector declines exponentially with increasing temperature due to heat losses. Thus, for a perpendicular solar incidence of 1000 W/m<sup>2</sup>, the combined peak efficiency of the collector and the power cycle in relation of TIT under the conditions studied occurs at approximately 550 °C for a 300 bar TIP and at approximately 530 °C for a 200 bar TIP, with a value of 27% and 26% respectively. These are approximately the maximum temperatures attainable by linear Fresnel collectors today, indicating an excellent synergy with CO<sub>2</sub>-based supercritical cycles.

## 6. ACKNOWLEDGEMENTS

This work was realized with the financial support of PETROBRAS (ANEEL: PD-00553-0042/2016), through the Teaching and Engineering Foundation of Santa Catarina (FEESC). We would like to extend our sincere thanks to Steag Energy Services for gently allowing the use of the EBSILON Professional software.

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