First and second law analysis of supercritical CO_2 recompression Brayton cycle

M.S. Khan, U. Atikol
Eastern Mediterranean University, Faculty of Engineering, Department of Mechanical Engineering, Famagusta, North Cyprus via Mersin 10, Turkey
E mail: engr.sajidazam@yahoo.com

Abstract

The present research concentrates on the energy and exergy analysis of the S-CO_2 recompression Brayton cycle and the individual components irreversibilities by varying the different operating parameters. Results show that the cycle efficiencies and LTR effectiveness reduce by increasing minimum cycle temperature, but HTR increases. The effect of minimum cycle temperature is more critical on cycle performance than maximum cycle temperature. The reactor has the highest irreversibility followed by recuperators and pre-cooler. Exergy efficiency shows a downward trend as environment temperature enhances. However, the effect of turbine inlet temperature is very low on-cycle efficiency and optimum pressure ratio for lower compressor outlet pressure values, which is more significant by increasing this parameter.

Keywords: S-CO_2 cycle, Nuclear reactor, Exergy analysis, irreversibility, pressure ratio.

Introduction

Due to the simplicity, safety and exceptional economy, the supercritical CO_2 cycle is a suitable conversion option for the next stage of nuclear reactors. The operating outlet temperature of the reactor will be in the range of 550-750 ℃. The S-CO_2 cycle benefits from the rapid variations of the CO_2 thermal physical properties near the critical point, which helps reduce the compression work. So high cycle thermal efficiency is achieved. Likewise, a recompression S-CO_2 Brayton cycle is used and can gain considerable higher efficiency. The main advantages of supercritical CO_2 Brayton cycle are listed as below : (a) Cycle efficiency is higher at the same inlet temperature of the turbine, which is due to the reduction in compressor work near critical point [1], (b) it lies in the non-flammable group (A1) and it is non-toxic [2], (c) as compared with helium its leakage rate is low and also its initial cost is lower than other thermodynamic cycles [3-4], (d) system is compact and less complex parts than Rankine system [4]. Angelino [3], in his research, presented that recompression S-CO_2 is the best technology than the other system configurations.

The use of simple cycle configuration is confined due to the problems that occurred by the temperature pinch point in HTR. This problem's main reason is the inequality in the heat capacity rate between the cold and hot sides [5]. The S-CO_2 recompression Brayton cycle eliminates this problem as two recuperators are used in this cycle.

Sulzer [6] in 1948 presented the early proposal of the S-CO_2 system. The use of the S-CO_2 cycle for power production systems was provided by Feher [7] in 1968. Feher [1] also studied S-CO_2 based nuclear power system's efficiency by considering the cycle pressure drop. However, each component's effect on system performance was studied by Angelino [3]. The effect of one and two-stage reheating on the system performance and proposed to use only single-stage reheating in the S-CO_2 cycle was investigated by Dostal [8]. Sarkar [9] carried out the optimization and exergy analysis of SCRBC and stated that the system's heat exchangers are more crucial components than turbo-machines from an exergy destruction perspective.

This paper mainly focuses on the compressor pressure ratio, energetic and exergetic efficiencies of the whole system, and the individual component's irreversibilities at various maximum and minimum temperatures and pressures that have not been studied in detail yet. The effect of the variation in environment temperature on cycle efficiency is also analyzed. Furthermore, recuperators' effectiveness and recompression fraction at different operating parameters are assessed and presented as well.

System description and assumptions

The efficiency of the S-CO_2 system enhances by using the recompression version as heat rejected from the cycle is reduced by introducing another compressor (recompression compressor).Figs. 1 and 2 show the layout and corresponding T-s diagram of the above said system, respectively. The low-pressure flow passes from the low-temperature regenerator (recuperator, LTR) and divided into two streams at the LTR exit (point 8). Mainstream (1-x) m_a becomes cool as it proceeds to pre-cooler (8a-1) and then through the central compressor (1-2), its pressure increases and eventually enters into LTR. The remaining low fraction stream with mass flow rate (x) m_a passed through recompression compressor (8b) and joined the stream exiting LTR at state 3. Before entering the reactor, the mainstream is heated through HTR, and after that, it passes through the turbine at state 5. It is essential to concentrate that stream (8b) has non-zero flow, and due to this, there are different mass flow rates for streams in LTR. Stream 7 has a higher mass flow rate than stream 2. Furthermore, the pressure of stream 7 is less than that of stream 2.

Assumptions:

All the processes in the system are steady-state.

Heat transfer with surroundings and pressure drops are assumed to be negligible.

Changes in kinetic and potential energies are ignored.

The processes in compressors and turbines are non-isentropic but adiabatic.

Thermal analysis and performance criteria

The mathematical modeling and design of the supercritical carbon dioxide Brayton system is considered and presented here. The Engineering Equation Solver (EES) software is developed to analyze the proposed model. The input data is indexed in Table 1. The equations that are used to model the proposed system are given in [9].

https://doi.org/10.24949/njes.v13i1.342

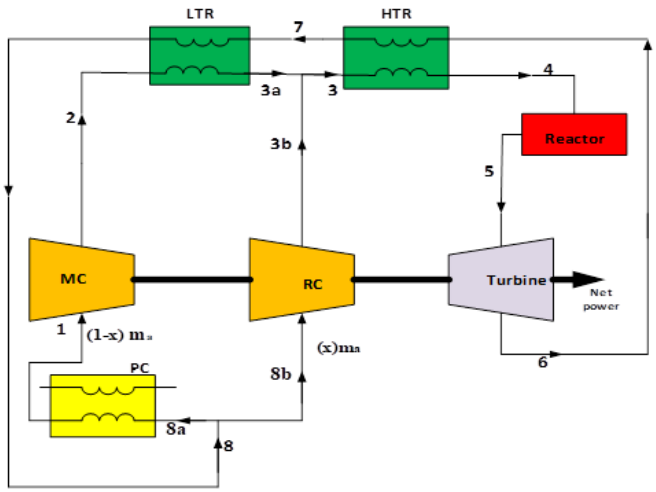

Figure 1: Schematic diagram for the S-CO₂ recompression Brayton cycle.

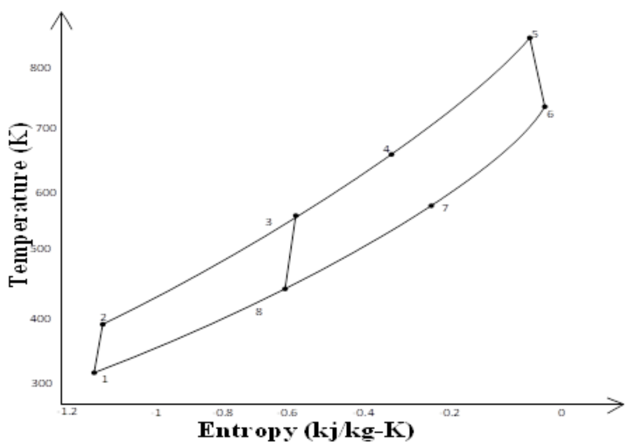

Figure 2: T-s diagram of recompression S-CO₂ Brayton cycle.

Thermodynamic relations for both recuperators are:

$h_6 - h_7 = h_4 - h_3$ (For HTR) (1)

$(1 - x)(h_3 - h_2) = (h_7 - h_8)$ (For LTR) (2)

Effectiveness of HTR can be calculated as:

$\varepsilon_{HTR} = (T_6 - T_7)/(T_6 - T_3)$ (3)

For the heat capacity of low pressure fluid is greater than that of high pressure fluid, effectiveness of LTR is given by equation (4).

$\varepsilon_{LTR} = (T_3 - T_2)/(T_7 - T_2)$ (4)

For reverse case equation (5) will be used.

$\varepsilon_{LTR} = (T_7 - T_8)/(T_7 - T_2)$ (5)

Reactor specific heat input (q_r) is presented as:

$(h_5 - h_4) = q_r$ (6)

Thermodynamic relations for other components can be expressed as:

$\dot{W}_{tur} = (h_5 - h_6)$ (7)

$\dot{W}_{mc} = (1 - x)(h_2 - h_1)$ (8)

$\dot{W}_{recomp} = (h_3 - h_8)$ (9)

Thermal efficiency of system is given by:

$\eta_{th} = (\dot{W}_{tur} - \dot{W}_{mc} - \dot{W}_{recomp})/\dot{Q}_{add}$ (10)

Exergy analysis

Exergy is also known as availability and is defined as the maximum theoretical work obtained by the system and specified reference surroundings (environment). The second law analysis or exergy analysis allows overcoming many of the deficiencies of energy analysis. Exergy analysis depends upon the second law of thermodynamics and is used to identify the reasons, positions, and quantity of the system's process inefficiencies.

The exergy, exergy efficiency, and exergy destruction rate of the S-CO_2 system are assessed at all the relevant points and presented here.

Input exergy to the system is expressed as:

$e_{x,in} = \left[1 - \frac{T_0}{T_r}\right] . q_r$ (11)

Where T_r = 800℃ [8] and T_0 = 25℃ [9]

Exergy at the inlet of main compressor can be calculated as:

$e_{x.1} = (1 - x)(h_1 - h_0) - T_0.(s_1 - s_0)$ (12)

Where (1-x), h_1, and s_1 are the fractions of mass flow rate, enthalpy, and entropy at state 1, respectively.

Similarly, exergy at all other states can be determined using the same procedure.

Furthermore, irreversibility or exergy destruction by different components can be expressed as:

$X_{dest,tur} = (e_{x,5} - e_{x,6}) - \dot{W}_{tur}$ (13)

Exergy destruction for both compressors can be calculated as:

$X_{dest,mc} = \dot{W}_{mc} - (1 - x)(e_{x,2} - e_{x,1})$ (14)

$X_{dest,rc} = \dot{W}_{rc} - x(e_{x,3} - e_{x,8})$ (15)

Similarly, irreversibility for both recuperators is given by Eqs. (16) and (17).

$X_{dest,HTR} = (e_{x,6} + e_{x,3}) - (e_{x,7} + e_{x,4})$ (16)

$X_{dest,LTR} = (e_{x,7} - e_{x,8}) - (1 - x)(e_{x,3} - e_{x,2})$ (17)

Pre-cooler and reactor subjected to exergy destruction can be determined as:

$X_{dest,pc} = (1 - x)(e_{x,8} - e_{x,1})$ (18)

$X_{dest,React} = e_{x,in} - (e_{x,5} - e_{x,4})$ (19)

Finally the overall exergy efficiency, η_{ex} is calculated as:

$\eta_{ex} = (\dot{W}_{tur} - \dot{W}_{mc} - \dot{W}_{rc})/e_{x,in}$ (20)

Table 1:Basic design and operating parameters for the analysis of S-CO₂ Brayton cycle [10].

Inlet temperature of the main compressor	305 K
Inlet temperature of the turbine	824 K
Inlet pressure of the turbine	200 bar
Pressure ratio	2.6
Effectiveness of HTR and LTR	0.85
Compressors isentropic efficiencies	0.90
Turbine isentropic efficiencies	0.90

Results and Discussion

The system's performance of the recompression S-CO_2 cycle is assessed by developing a complete simulation code. Conservation of mass, energy as well as exergy balance are applied. Thermodynamic relations and energy exergy equations are solved simultaneously using EES. The main focus is paid to the second law of thermodynamics, and

availability values at different locations are calculated. The basic design parameters are listed in table 1.
$\gamma_p, T_{min}, P_{max}, P_{in}, T_{max}, \varepsilon_{HTR}, \varepsilon_{LTR}, \eta_{is,tur}, \eta_{is,mc}, \eta_{is,rc}$:The input parameters, are used and based on the energy balance equations of individual parts. Furthermore, the minimum cycle temperature is higher than the critical point, so no pinch point problem has happened in the simulation. Detailed exergy and irreversibility of the components are presented in table .2. The model has been validated with the reference [11]. The system performance was compared for minimum cycle temperature =305 K, maximum cycle temperature=824 K, maximum cycle pressure ratio=200 bar, and pressure ratio =2.6 indicates that cycle thermal and exergetic efficiencies are 44.70% and 57.56% with recompression mass fraction 0.39 is almost the same as mentioned in [11]. Furthermore, the relationship between entropy and temperature at designed conditions is validated with the reference [11], and our values are well meet with them. A property code was established and used to simulate recompression S-CO_2 system, based on Span and Wagner correlations [12].

Table 2

An exergy balance sheet of S-CO_2 recompression cycle

Exergy	Kj/kg	irreversibility	Kj/kg	%
Reactor	171.35	Reactor	29.78	27.66
		Pre-cooler	16.85	15.65
		HTR	17.64	16.38
		LTR	12.15	11.28
Turbine	148.21	Turbine	6.39	5.93
Main-compressor	22.65	Main-compressor	7.11	6.60
Re-compressor	18.34	Re-compressor	3.01	2.80
Net exergy output	107.22	Total irreversibility	92.9	86.3

The effect of compressor inlet temperature on the system's performance is crucial due to the property variations symbolically adjacent to the critical point of the S-CO_2 cycle**.** The change in optimum compressor pressure ratio, thermal efficiency, and exergy efficiency for S-CO_2 Brayton system at different compressor (main) inlet temperatures (maximum operating pressure and temperature are 200 bar and 305 K, respectively) are shown in Fig.1. As the difference between the minimum and maximum cycle temperature becomes lower (at the compressor inlet, specific enthalpy reduces because the specific heat capacity of CO_2 is less), the cycle thermal efficiency reduces linearly with an increase in central compressor inlet temperature. That is why work done by the central compressor grows up. However, re compressor and turbine works are minimal as compared to the main compressor. Finally, efficiency shows a downward trend if compressor inlet temperature will increases.

The effect of increasing cycle minimum temperature on components exergy destruction rate at constant operating conditions is shown in Fig.2. The reactor's irreversibility shows a downward trend due to the decrease in mean cycle temperature difference when minimum cycle temperature enhances. For HTR and LTR's case, the exergy destruction rate also decreases by increasing compressor inlet temperature. The reason beyond this, as the temperature difference between interacting streams of both recuperators is more significant.

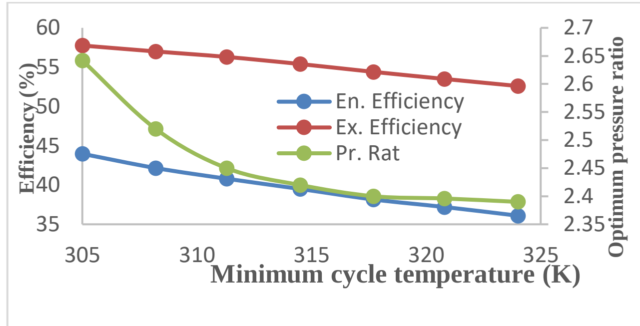

Figure 3:. Effect of minimum cycle temperature on cycle performance

So the heat transfer takes place efficiently and irreversibilities will decrease. However, this variation in irreversibility for LTR is less than HTR. The pre-cooler's irreversibility boosts up considerably due to the difference in temperature between two flows increases in this section. Furthermore, the specific heat capacity will degrade by increasing inlet temperature, which intensifies the central compressor's irreversibility. However, the re compressor and turbine effect is insignificant as the specific heat capacity remains unchanged away from the critical point. Increasing the main compressor inlet temperature, LTR and HTR's effectiveness shows different variations as depicted in Fig.3. The reason is that they perform more in the ideal gas regime as away from the critical point. The specific heat of both fluids flows in the recuperators remains the same, disturbing the recuperators' temperature difference shape. However, this effect is substantial in HTR (as the mass flow rate is identical on both sides of HTR). This phenomenon lowers the effectiveness of a high-temperature recuperator.

Moreover, the effectiveness of LTR is contrasting to HTR as recompressed fraction varies on both sides of LTR that will increase low-temperature recuperator performance. Furthermore, Fig.4. shows the effectiveness of a high-temperature recuperator at various maximum operating temperatures and pressures. HTR has a considerable effect on cycle efficiency as there is a more significant amount of heat reproduced in HTR. Beyond the compressor outlet pressure of 20 MPa, HTR effectiveness is constant or decreasing slightly.

The variations caused by maximum operating pressure on cycle energetic and exergetic efficiencies and optimum compressor pressure ratio for different turbine inlet temperatures are shown in Fig. 5. As the value of specific heat capacity is greater near the critical point, the optimum compressor pressure ratio enhances by increasing cycle maximum pressure. By increasing the temperature, an improvement in efficiency almost linearly. There is a substantial improvement in efficiency up to 23 MPa. However, this development becomes minor for higher pressure as the cycle diverts from its critical value.

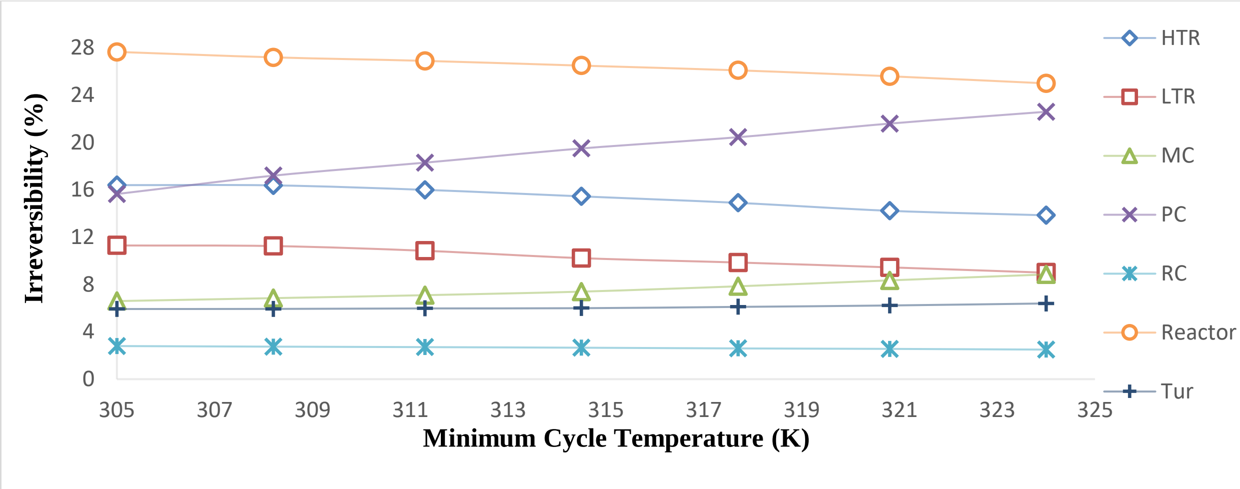

Figure 4: Min. cycle temp. effect on components irreversibility.

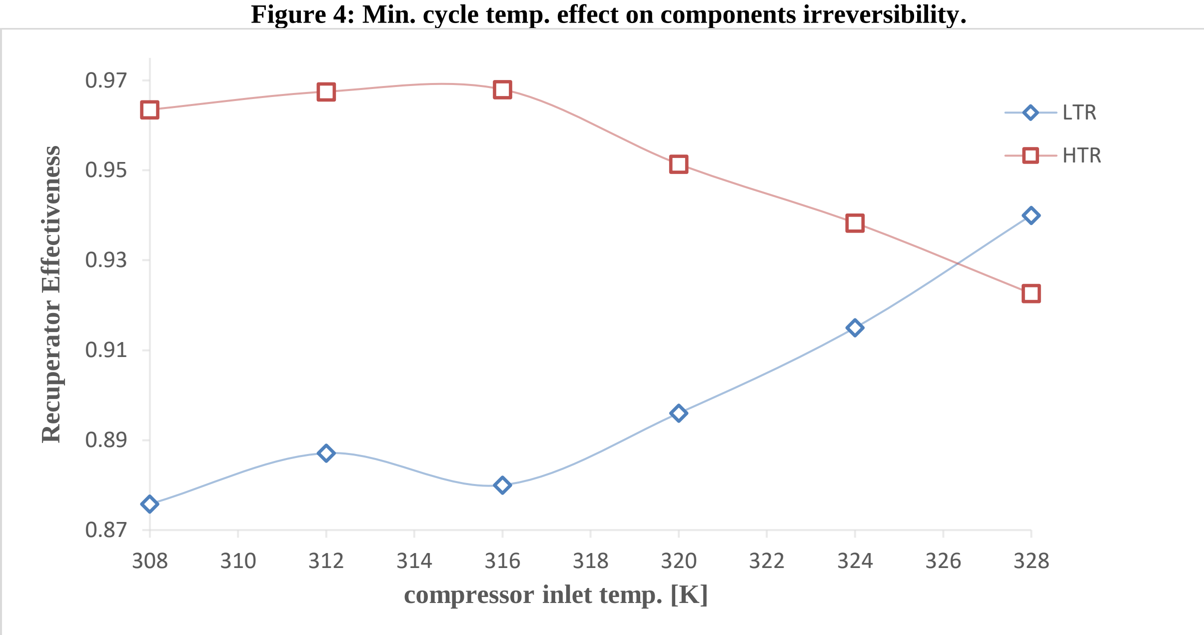

Figure.5: Minimum temp. effect on recuperator performance

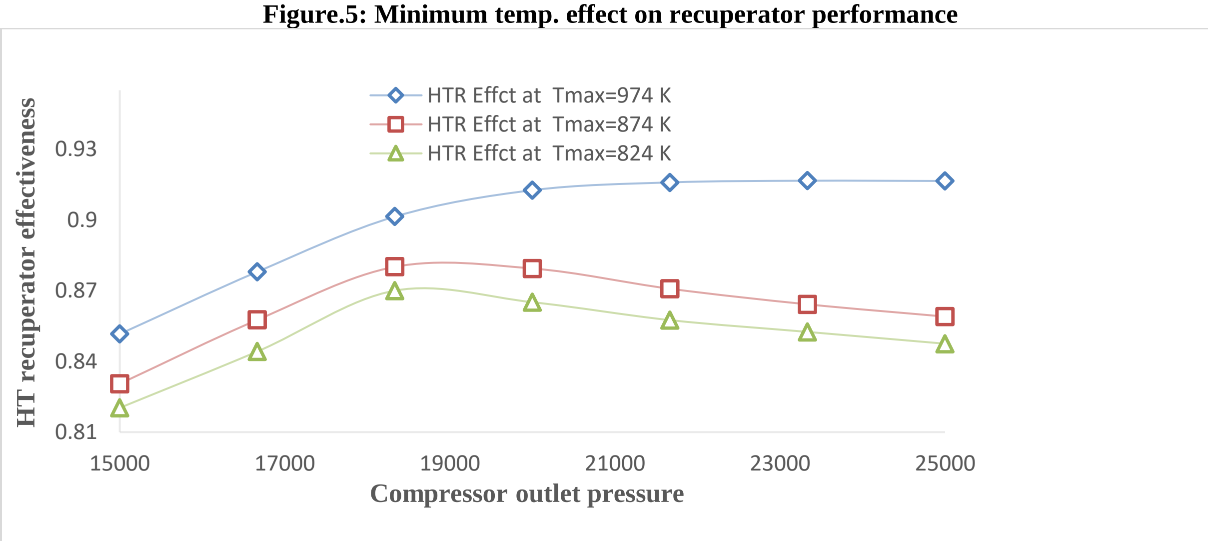

Figure. 6: HTR effectiveness for recompression cycle at various turbine inlet temperatures

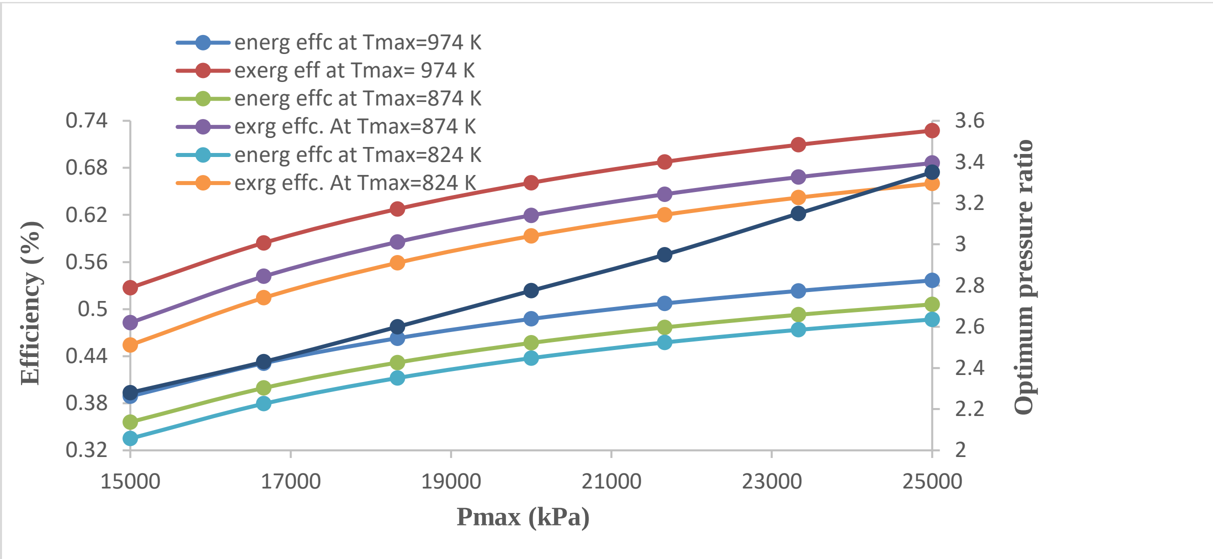

Figure.7:. Effect of compressor outlet pressure. on system performance

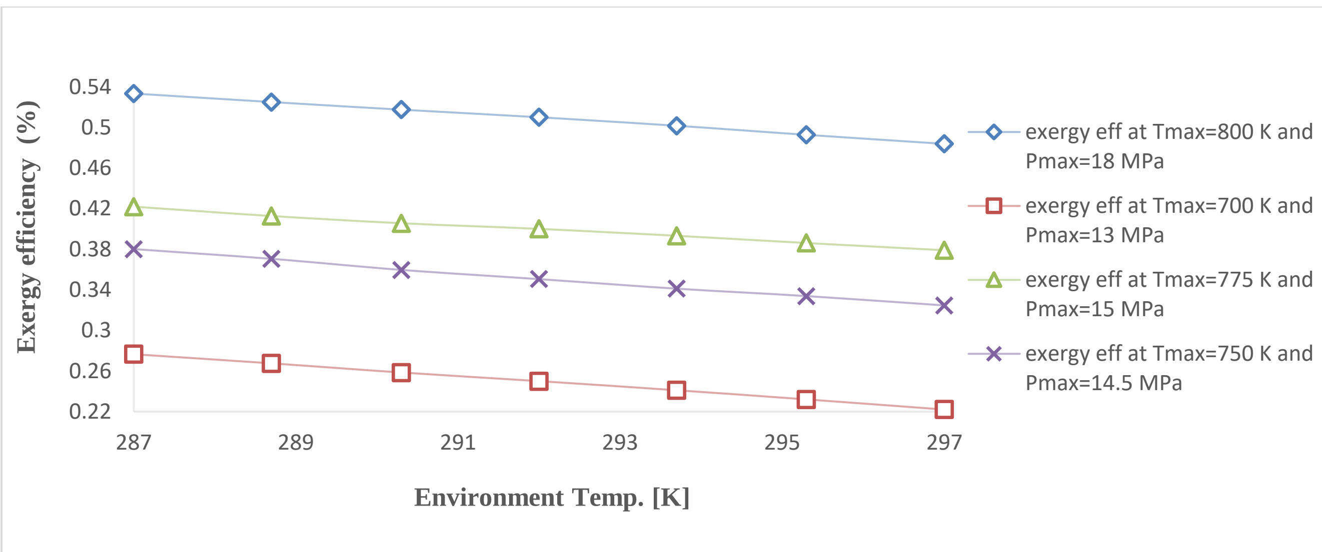

Figure 8: Effect of $T_{ambient}$ on the exergy efficiency for different turbine inlet temperatures.

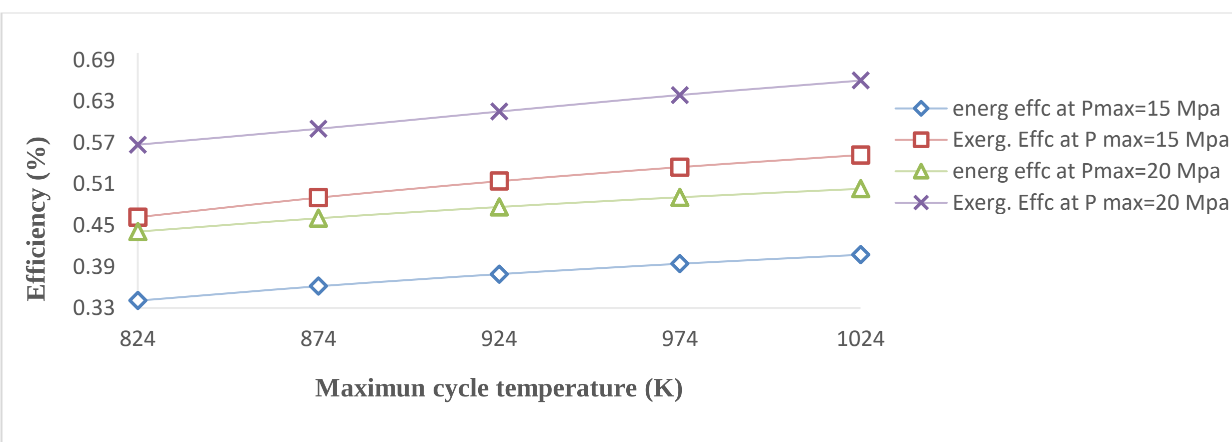

Figure 9: Effect of turbine inlet temperature on the efficiency for different compressor outlet pressures.

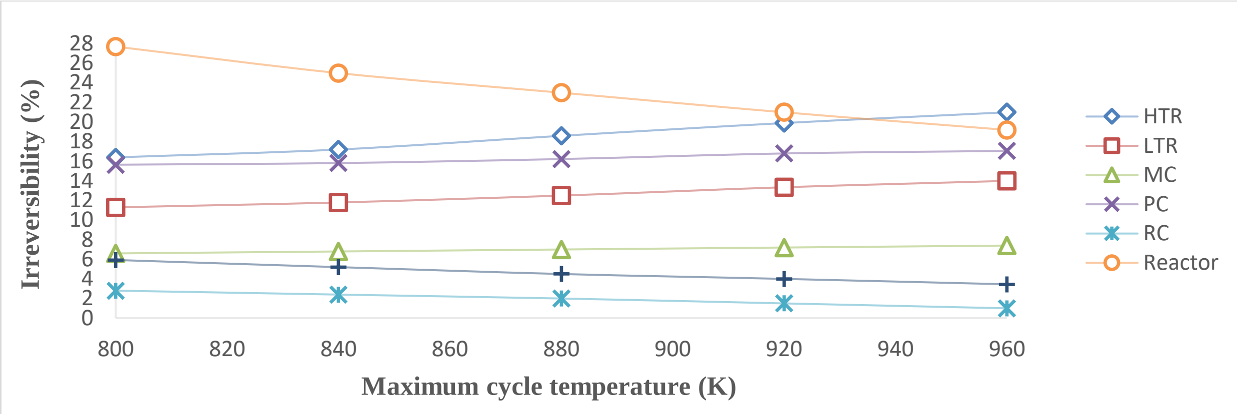

Figure 10: Effect of maximum cycle temperature on Components irreversibilities**.**

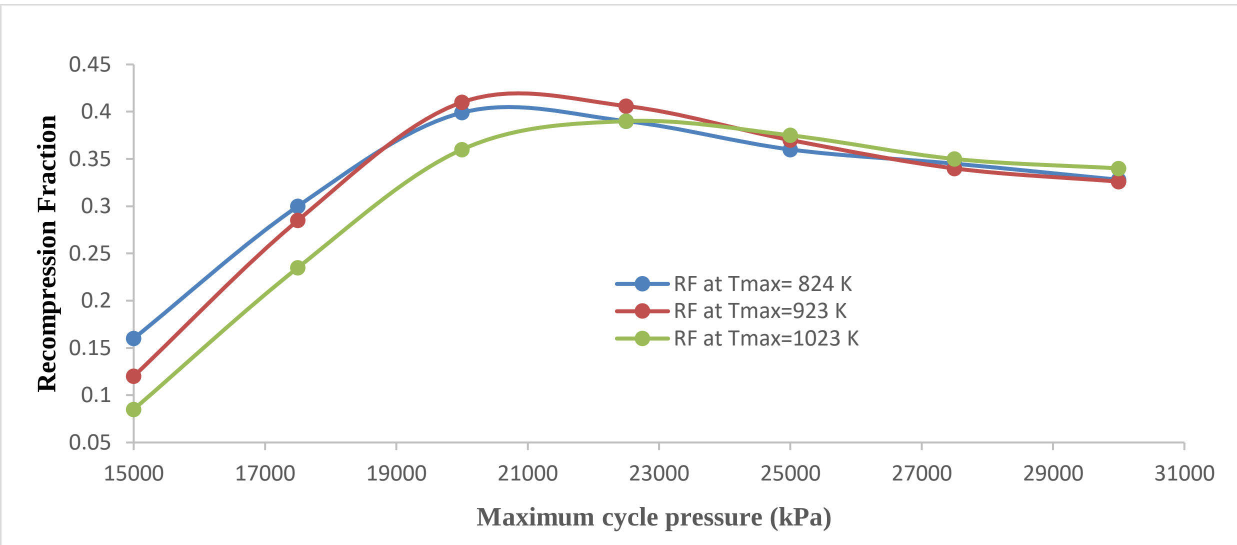

Figure 11: Recompressed fraction for recompression cycle

Fig. 6. explains how the variations in environment temperature affect the cycle's exergetic efficiency under various turbine inlet temperatures and pressures. Efficiency reduces gradually as ambient temperature increases due to the reduction in turbine power.

The effect on the cycle's energetic and exergetic efficiencies by varying the turbine inlet temperature under maximum pressure of 15 MPa and 20 MPa, respectively, is shown in Fig. 7. The system's second and first law efficiency improves from 57% to 66% and 43% to 50.5%, respectively, by enhancing the turbine inlet temperature from 824 K to 1024 K, for compressor outlet pressure of 20 MPa. However, when operating under 15 MPa, both of the efficiencies have lower values than earlier. This is because by increasing the maximum operating temperature, the turbine's work output increases considerably as inflow enthalpy for the turbine will increase. This reason leads to reduce the turbine and re-compressor irreversibility, as explained in Fig.8.

Furthermore, by increasing the maximum cycle temperature, heat in and heat out temperature difference becomes more significant, resulting in higher efficiency. Another justification can be made as the turbine inlet temperature is away from the critical point. The effect of critical point on compressor inlet temperature is more pronounced than turbine inlet temperature. By increasing the turbine's inlet temperature, heat exchange in the reactor will be greater, further decreasing the irreversibility. On the other hand, exergy destruction rates of HTR and LTR increase because the temperature difference between two (high and low) pressure stream flows increases.

Variations in compressor outlet pressure cause a more significant impact on recompressed fraction at three different turbine inlet temperatures, as depicted in **Fig. 9.** The recompressed fraction value is very low (about 10-12 % of the total mass flow rate) as the primary compressor outlet pressure is less. A main portion of the streamflow is forced to the pre-cooler, resulting in more heat extraction from the cycle and decreasing cycle efficiency. The recompressed fraction attains its peak around 20 MPa and the cycle operates its best thermodynamically at this pressure. However, beyond this point, further increment in pressure will gradually decrease the recompressed fraction.

Conclusions

Comprehensive first and second law efficiency and components' irreversibilities have been carried out by varying the different operating parameters, including minimum cycle temperature, maximum cycle pressure, temperature, and ambient temperature. Outcomes present that irreversibilities of compressors and turbines are less considerable than the heat exchangers. The recompression fraction attains its maximum value of almost 39 % when the maximum cycle pressure will be 200 MPa. The impact of main compressor inlet temperature on system performance and pressure ratio is more prevailing than the turbine inlet temperature due to the symbolic effect on specific heat capacity variation near the critical point. The reactor's irreversibility is maximum approximately 27.66%, followed by HTR, pre-cooler and LTR with 16.38%, 15.65% and 11.28%, respectively. So, better chances are available for exergy improvements, but this part of exergy destruction cannot be completely rectified because of physical constraints.

Nomenclature

e_x	specific exergy (kj/kg)
h	specific enthalpy (kj/kg)
q	specific heat input (kj/kg)
γ_p	main compressor pressure ratio
s	specific entropy (kj/kg K)
T	temperature (K)
T_0	reference temperature (K)
T_r	reactor temperature (K)
$\dot{W}$	specific work done (kj/kg)
x	recompressing mass fraction
X_{dest}	exergy destruction (kj/kg)
ε	effectiveness of heat exchanger
η_{ex}	exergy efficiency (%)
η_{th}	thermal efficiency (%)

Subscripts

HTR	high temperature recuperator
is	isentropic
LTR	low temperature recuperator
min	minimum
max	maximum
mc	main compressor
pc	pre-cooler
React	reactor
rc	recompressing compressor
$S\text{-}CO_2$	supercritical carbon dioxide
tur	turbine
th	thermal

REFERENCES

1. Feher EG. The supercritical thermodynamic power cycle. Douglas Paper No.4348. In: Proceedings of the IECEC, Florida; 1967.
2. ASHRAE. ASHRAE 15-2013, ASHRAE; 2013.
3. Angelino G. Real gas effects in carbon dioxide cycles. ASME Paper No. 69-GT-103; 1969.
4. Turchi C, Ma Z, Dyreby J. Supercritical CO2 for application in concentrating solar power systems. In: Proceedings of supercritical CO2 power cycle symposium; 2009.
5. Turchi CS, Ma Z, Neises TW, Wagner MJ. Thermodynamic study of advanced supercritical carbon dioxide power cycles for concentrating solar power systems. J Sol Energy Eng 2013;135(4):041007.
6. Sulzer G. Process for generating work from heat (in Swiss). Swiss Patent CH 269599; 15 July 1950.
7. Feher EG. The supercritical thermodynamic power cycle. Energy Convers 1968;8:85e90 [Printed in Great Britain].
8. Dostal V, Driscoll MJ, Hejzlar P. A supercritical carbon dioxide cycle for next generation nuclear reactors. MIT-ANP-TR-100. March; 2004.
9. Sarkar J. Second law analysis of supercritical CO2 recompression Brayton cycle. Energy 2009;34:1172e8.
10. Sarkar J, Bhattacharyya S. Optimization of recompression S-CO2 power cycle with reheating. Energy Convers Manage 2009;50:1939e45.
11. Akbari AD, Mahmoudi SMS. Thermoeconomic analysis & optimization of the combined supercritical CO_2 (carbon dioxide) recompression Brayton/organic Rankine cycle. Energy 2014;78:501-12.
12. Span R, Wagner W. A new equations of state for Carbon dioxide covering the fluid region from triple point temperature to 1100 K at pressure up to 800 MPa. J Phys Chem Ref Data 1996;25:1509–96.