Energetic and exergetic studies of modified CO_2 transcritical refrigeration cycles

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Abstract

Thermodynamic analysis including energetic and exergetic analysis is carried out employing Engineering Equation Solver for the five modified cycles: dual expansion cycle, internal heat exchanger cycle, work recovery cycle, work recovery with internal heat exchanger cycle and vortex tube expansion cycle. Contours are developed to study the effect of gas cooler temperatures and evaporator temperatures on the system performance and optimum gas cooler pressure. The modified cycle with work recovery turbine offers relatively higher COP and higher exergetic efficiency with lower compressor discharge pressure. The exergy loss in compressor, gas cooler, throttle valve and vortex tube (VT) are considerably higher than that in internal heat exchanger (IHX), evaporator and turbine. It is observed that COP of modified cycle with VT is slightly less than that with IHX, whereas the cycle with work recovery turbine brings the highest COP with the improvement of 25% at the gas cooler exit temperature of 305 K and evaporator temperature of 248 K.

Keywords: CO_2; exergy; COP; transcritical

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1. INTRODUCTION

The two menaces, global warming and ozone depletion, are the threats to the environment sustainability and compelling us to look for new alternatives. Use of natural refrigerants is always the preferred choice being the ecological safe in search of suitable alternatives. Being the natural refrigerant and inherited excellent thermodynamic and heat transfer properties, carbon dioxide (CO_2) has proven to be a promising substitute [1]. Further, safety characteristics such as non-flammability, non-toxicity of CO_2 also favor it as the attractive replacement. CO_2 has been successfully commercialized in low ambient temperature environment; however, use of CO_2 in high ambient temperature (HAT) applications is still a challenge as the system performs poorly at HAT [2]. One of the distinguish features of CO_2 is the low critical temperature (31.1°C) that makes the CO_2 cycle transcritical, consequently the CO_2 system performance vulnerable at high temperature applications. Several modifications have been suggested by various researchers in the past to improve the CO_2 system performance at high temperature applications.

The results of the experimental investigation by [3] showed that the use of internal heat exchanger (IHX) has trivial influence on the system performance at low and moderate gas cooler exit temperatures; however, taking longer length of IHX reduces the optimum discharge pressure. Sarkar *et al.* [4] carried out the exergetic analysis of a transcritical CO_2 system using IHX and reported that the minimum exergy destruction in IHX compared to other components. Dia *et al.* [5] investigated the performance of a direct mechanical subcooling integrated to transcritical CO_2 heat pump system. Coefficient of performance (COP), optimum discharge pressure, city-specific seasonal performance factor are estimated based on the energetic and economic performance. Response surface methodology is employed to study the quantitative difference of optimum gas cooler pressure of transcritical CO_2 refrigeration and water heat pump system [6]. It is shown that gas cooler outlet temperature is the most dominating factor on optimum pressure with 96.38% contribution.

Agrawal and Bhattacharyya [7–10] in their seminal studies investigated capillary tubes as an expansion device in transcritical CO_2 system to run the system optimally. Sarkar [11] evaluated the system performance using vortex tube (VT) as an expansion device and found that the VT expansion system is more effective for high temperature lift applications and lower optimum discharge pressure. Elbel and Hrnjak [12] performed experimental investigations with flash gas bypass and found that the COP and cooling capacity increase by 7% and 9%, respectively. Robinson

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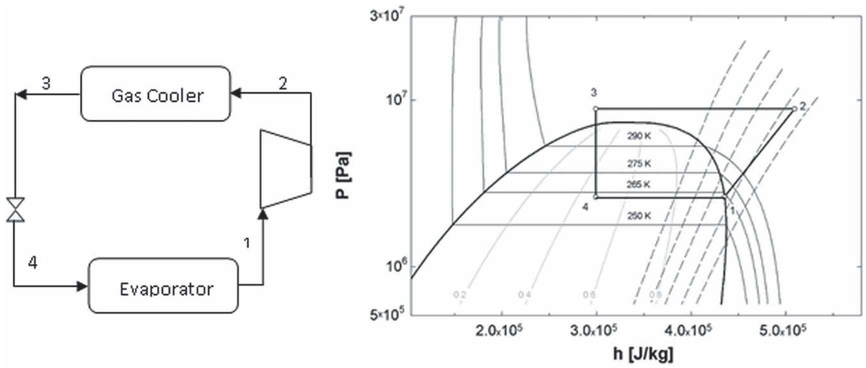


Figure 1. *BC and corresponding P-h plot.*

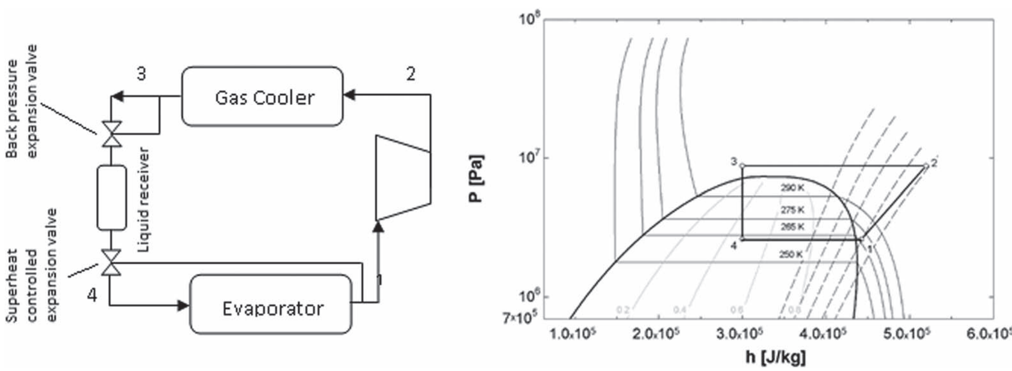


Figure 2. *DEC and corresponding p-h plot.*

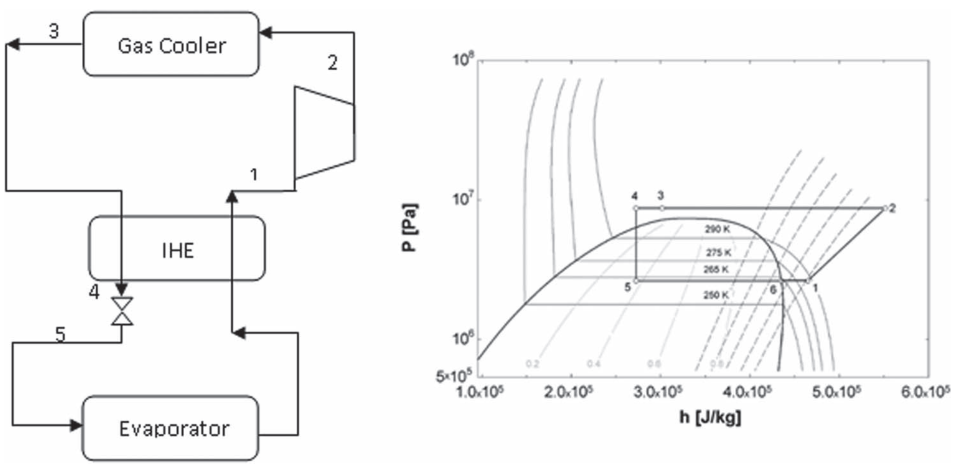


Figure 3. *IHEC and corresponding P-h plot.*

and Groll [13] reported the reduction on cycle irreversibility by employing expansion turbine. Baek *et al*. [14, 15] carried out the theoretical and experimental investigations of a transcritical CO_2 system using work-producing reciprocating piston-cylinder device and reported 10% improvement in system COP. Pérez-García [16] carried out the comparative study of various transracial CO_2 cycle configurations and found that the configuration with turbine as an expansion device performs better in comparison to other configurations. Li *et al*. [17] carried out the thermodynamic analysis of transcritical CO_2 system with vortex tube as expansion device and found COP improvement by 37%. It is also reported that the efficiency of VT (ratio of enthalpy drop of cold mass to the isentropic enthalpy drop of total mass) should be above 0.38 to achieve higher COP.

Kronhauser [18] proposed the transcritical CO_2 system with ejector as an expansion device and reported the reduction in both throttling losses and work of compression. Li and Groll [19] in their study found 16% improvement in COP with the

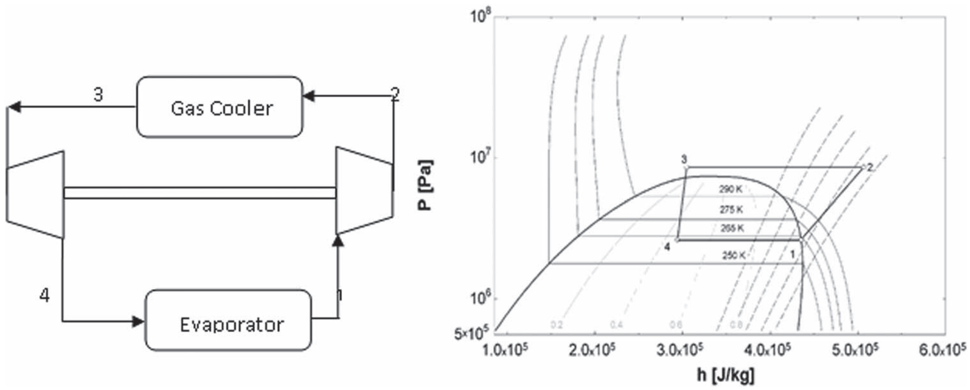


Figure 4. *WRC and corresponding P-h plot.*

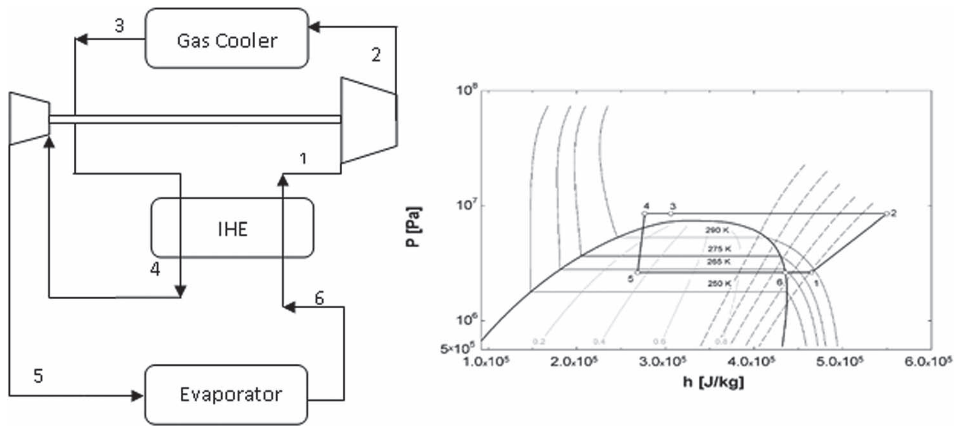


Figure 5. *WRCIHE and corresponding P-h plot.*

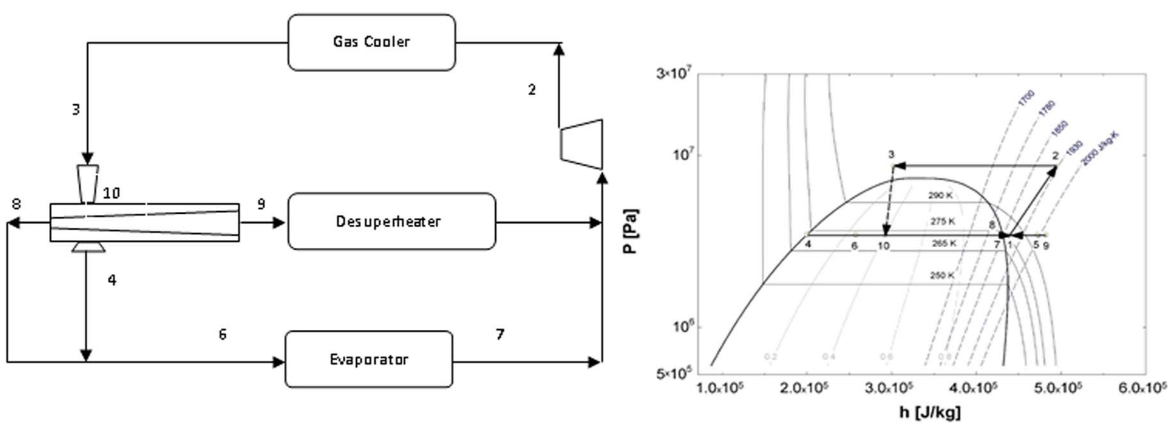
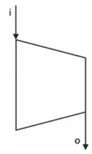
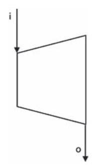
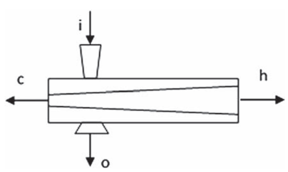


Figure 6. *VTEC and corresponding P-h plot.*

use of ejector as expansion device and also reported that gas cooler pressure can be controlled by varying throat area of motive nozzle of ejector. Eskandari *et al*. [20] studied two stage transcritical CO_2 cycle with two ejectors and concluded that the exergy destruction in compressor and gas cooler lowers due to use of two ejectors. Studies on cycle modifications to improve the transcritical CO_2 cycle performance are reported in open literature. No attempt has been made to compare among the various cycle modifications based on energetic and exergetic approach together. The core objective of the present study is to compare the effect of various transcritical CO_2 cycle modifications including prediction of optimum operating conditions based on first law and second laws of thermodynamics together. The various cycle configurations are simulated using Engineering Equation

Table 1. *Component wise steady state thermodynamic modeling.*

Component	Symbol	Characteristic equations
Evaporator		$q_e = h_o - h_i$ $I = mT_o[(s_o - s_i) - q_e/T_{sf,e}]$
Compressor		$\eta_c = (h_{o,s} - h_i)/(h_o - h_i)$ $w_c = (h_o - h_i)$ $I = mT_o(s_o - s_i)$
Gas cooler		$q_{gco} = h_i - h_o$ $I = m[h_i - h_o - T_o(s_i - s_o)]$
Expansion valve		$h_i = h_o$ $I = mT_o(s_o - s_i)$
Turbine		$\eta_t = (h_i - h_o)/(h_i - h_{o,s})$ $w_t = (h_i - h_o)$ $I = mT_o(s_o - s_i)$
IHX/desuperheater		$\varepsilon = (T_{co} - T_{ci})/(T_{hi} - T_{ci})$ $I = mT_o[(s_{Tco} - s_{Tci}) + (s_{Tho} - s_{Thi})]$
Vortex tube		$m_i = m_c + m_h + m_o$ $m_i hi = m_c h_c + m_o h_o + m_h h_h$ $I = T_o[m_c s_c + m_o s_o + m_h s_h - m_i s_i]$
Mixing of two streams		$m_o = m_1 + m_2$ $m_o ho = m_1 h_{i1} + m_2 h_{i2}$ $I = T_o[m_o s_o - m_1 s_{i1} - m_2 s_{i2}]$

Solver (EES) and results are compared to find the suitability of the modification.

2. CYCLE CONFIGURATIONS

The basic transcritical CO_2 cycle and corresponding P-h plot is shown in Figure 1. As shown in the figure, at state 1, the exit of the evaporator is taken as saturated vapor. Unlike subcritical cycle point 3 lies above the critical point where pressure and temperature are independent. Further, there is an existence of the optimum pressure for a given gas cooler outlet temperature due the slope of isothermal lines in supercritical region [1]. Expansion of liquid takes place in two stages in dual expansion cycle (DEC) as shown in Figure 2, where second expansion is governed by the superheat in the evaporator. Combining superheating of vapor with liquid subcooling for mutual benefits through liquid vapor regenerative heat exchanger known as IHX is a common practice in all practical subcritical systems. Adopting the same technique an IHX is employed to cool the gas cooler outlet stream with the evaporator outlet stream and corresponding flow and P-h representation is shown in Figure 3.

Being the large pressure difference between gas cooler and evaporator and gas cooler outlet lies in supercritical region, there is an opportunity of work recovery in transcritical CO_2 cycle by carrying out expansion process in work recovery turbine. A cycle with work recovery turbine and corresponding P-h plot is shown in Figure 4. A further modification is suggested to use an IHX in the work recovery turbine cycle. This may enhance the refrigerating effect. The cycle representation and corresponding P-h chart is shown in Figure 5.

As discussed earlier that in the transcritical CO_2 cycle, the exit of the gas cooler lies in the supercritical state where refrigerant is in gaseous state and makes possible to use the VT as an expansion device. The cycle with VT as an expansion device and corresponding P-h chart is shown in Figure 6. The cycle

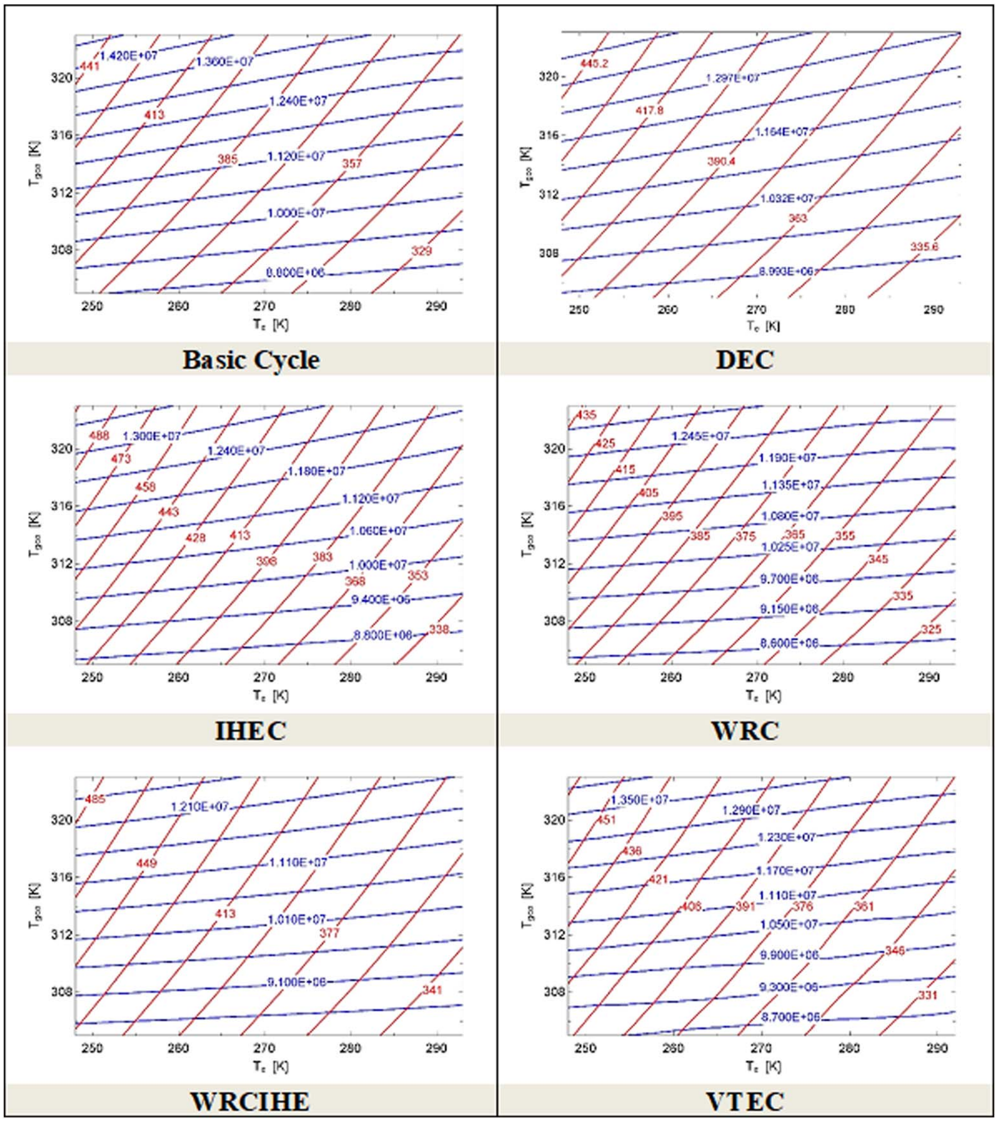


Figure 7. *Contours of optimum discharge pressure and discharge temperature of chosen configurations.*

Table 2. *Typical values optimum discharge pressure and discharge temperature.*

Sr. no.	Cycle	Evaporator temperature (K)	Gas cooler exit temperature (K)	Optimum discharge pressure (bar)	Discharge temperature (K)
1	BC	260	312	104.4	386.8
2	DEC	260	312	106.5	387.4
3	IHEC	260	312	100.4	412.8
4	WRC	260	312	99.4	382.4
5	WRCIHE	260	312	97.3	419.2
6	VTEC	260	312	103.2	399.0

is based on the Maurer model [11]. The stream 8 is the cold stream gaseous phase while stream 4 is in liquid phase and stream 9 is at superheated state. Stream 9 is cooled in desuperheater. Note that the outlet temperature of desuperheater will always be higher than the ambient temperature. The evaporator outlet and the desuperheater outlet are combined and compressed isentropically.

3. MODELLING AND SIMULATION

All the cycles are modeled employing steady-state mass, energy and exergy balance across the individual component. The isentropic efficiencies of the compressor and turbine are considered as 70% and 60%, respectively [13, 40]. The effectiveness of IHX and desuperheater is taken as 50% and 85%, respectively [11, 17]. The

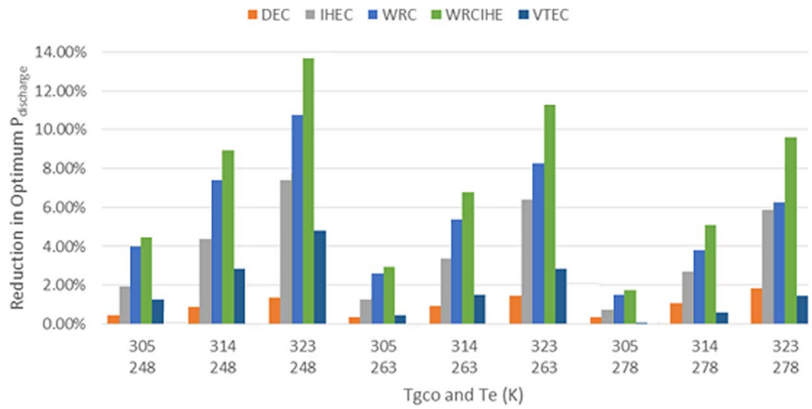


Figure 8. *Percentage reduction in optimum gas cooler pressure.*

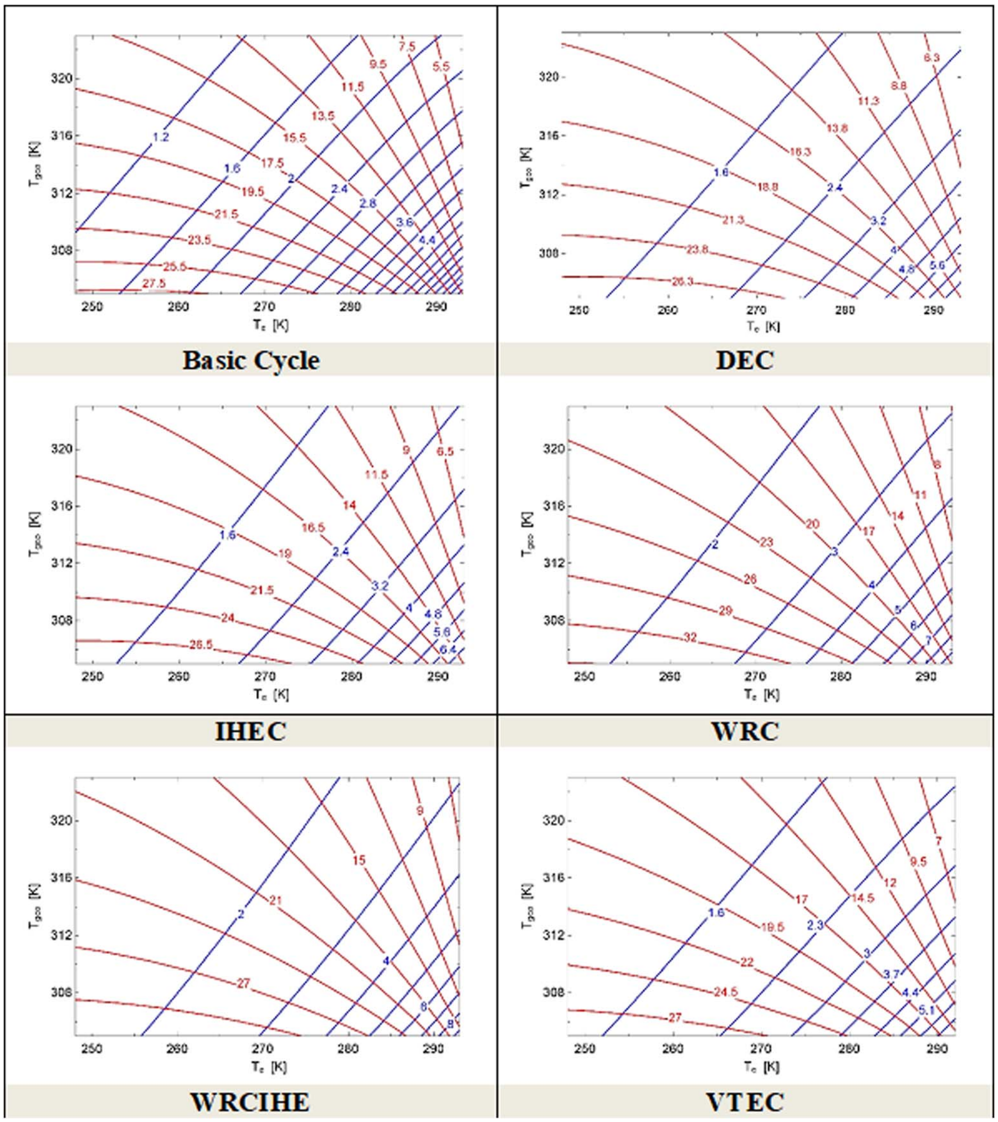


Figure 9. *Contours of COP and II law efficiency of chosen configurations.*

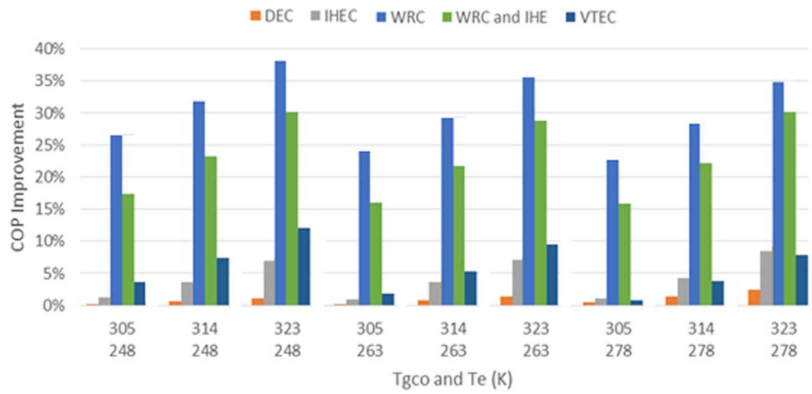


Figure 10. *Percentage improvement in COP.*

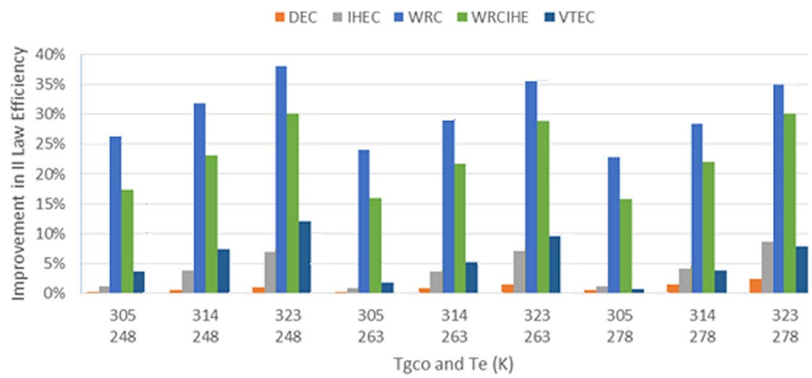


Figure 11. *Percentage improvement in II law efficiency.*

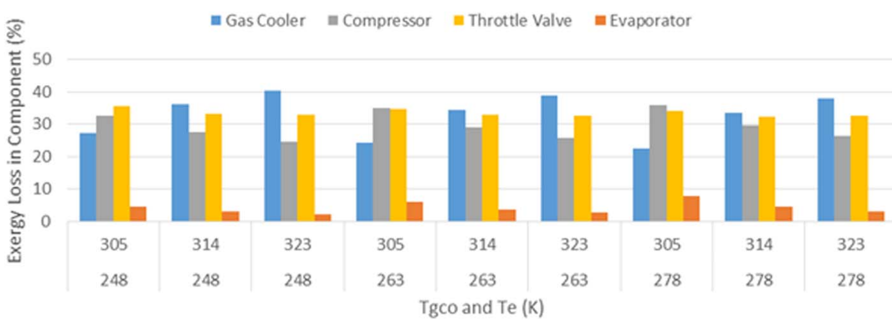


Figure 12. *Component-wise percentage exergy loss in DEC.*

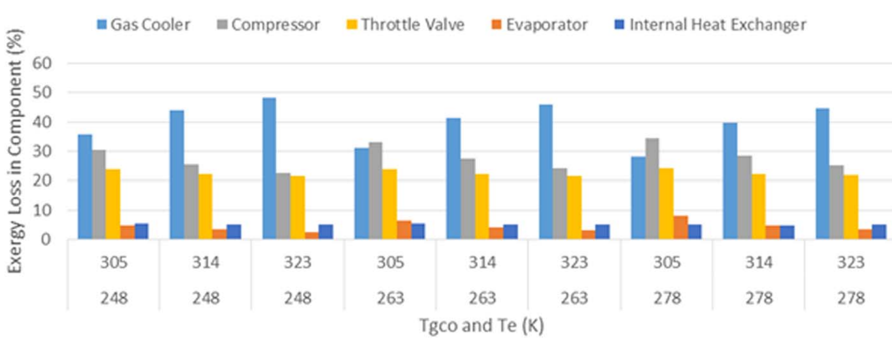


Figure 13. *Component-wise percentage exergy loss in IHEC.*

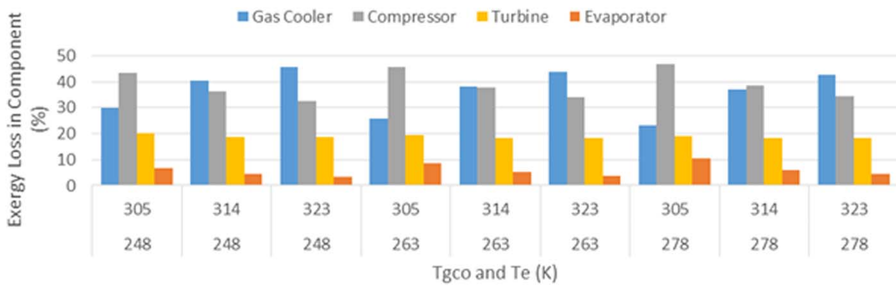


Figure 14. *Component-percentage exergy loss in WRC.*

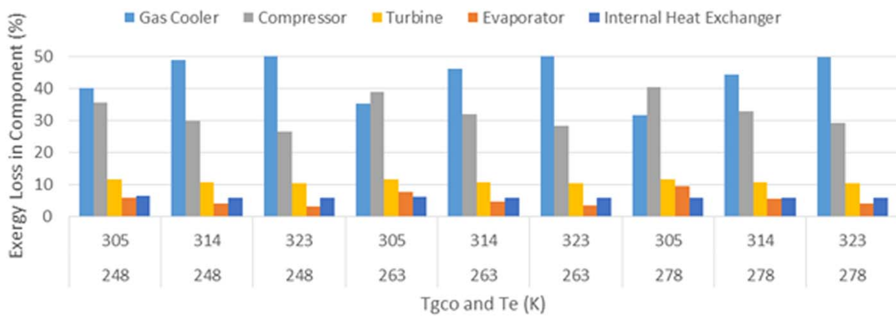


Figure 15. *Component-wise percentage exergy loss in WRCIHE.*

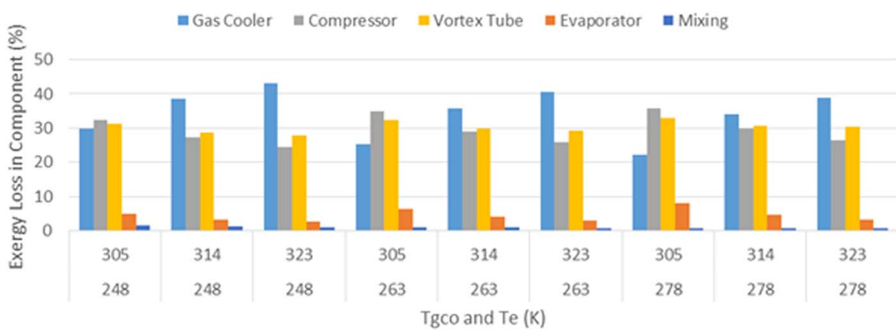


Figure 16. *Component-wise percentage exergy loss in VTEC.*

component-wise characteristic equations are shown in Table 1. The cold mass fraction for VT is taken as 0.5 while the isentropic efficiency of nozzle is taken as 80% [11]. The pressure drop in all components is assumed as zero. Heat transfer with the ambient is assumed to be negligible. Mixing processes is considered as isobaric. The simulation is carried out with steps of 0.5 K for chosen evaporator temperatures (248 K $\leq$ Te $\leq$ 293 K) and gas cooler exit temperatures (305 K $\leq$ Tgco $\leq$ 323 K) with the optimization of the pressure and corresponding COP at each step. Exergy analysis (based on second law efficiency defined as the ratio of exergy recovered to the exergy supplied) that is carried out based on the maximum COP where the temperature of secondary fluid (water) is assumed to be 5°C above the evaporator temperature, and the temperature of the inlet water for desuperheater is assumed to be 300 K. The ambient temperature is taken as 303 K.

4. RESULT AND DISCUSSION

It is shown that at a given gas cooler exit temperature, there exists an optimum gas cooler pressure for the transcritical CO_2 cycle where system demonstrates maximum COP [1]. Six different transcritical CO_2 cycle configurations starting from basic cycle (BC), DEC, internal heat exchanger cycle (IHEC), work recovery cycle (WRC), work recovery with internal heat exchanger cycle (WRCIHE) and vortex tube expansion cycle (VTEC) are studied based on energetic and exergetic analysis. The optimum discharge pressure and discharge temperature contours for all the chosen cycle configurations are shown in Figure 7. It can be seen that the effect of evaporator temperature on optimum gas cooler pressure and discharge temperature is relatively less as compared to the gas cooler exit temperature for all the chosen cycle configurations.

For a given evaporator temperature as gas cooler exit temperature increases, both optimum gas cooler pressure and temperature increase significantly. However, increase in evaporator temperature brings marginal change, it increases in optimum gas cooler pressure and temperature. These contours help to find the optimum gas cooler pressure and discharge temperature for the chosen gas cooler exit temperature and evaporator temperature in a given configurations. Iso-discharge pressure and iso-discharge temperature contour lines are nearly parallel. Typical values of optimum discharge pressure and corresponding discharge temperature at given evaporator and gas cooler exit temperatures are tabulated in Table 2.

It is desirable to keep the optimum discharge pressure low. For a given evaporator temperature and gas cooler outlet temperature, WRCIHE has lowest optimum discharge pressure, followed by WRC, IHEC and BC. Optimum discharge pressure for DEC and VTEC is in between IHEC and BC. The percentage reduction in optimum discharge pressure compare to BC of the selected cycle configurations is shown in Figure 8. In DEC, decrease in vapor quality at the inlet to the evaporator enables increase in specific refrigeration effect and decrease in discharge pressure. Use of IHX or expansion turbine lowers optimum pressure as compared to conventional cycle. Further, use of IHX enables subcooling that leads to increase in the specific refrigeration effect in turn decrease in optimum pressure. The combined effect of availability of the work and subcooling using work recovery turbine and IHX, respectively, enable lowering the optimum discharge pressure in WRCIHE cycle.

Thermodynamic analysis based on energetic and exergetic approach to calculate COP and II law efficiency, respectively, provides comprehensive information of a thermal system. The COP means ratio of desired effect to required input and II law efficiency is ratio of exergy recovered to exergy supplied. The COP and II law efficiency contours are shown in Figure 9. It is exhibited that for a given evaporator temperature as gas cooler exit temperature increases the system COP decreases and it increases with increase in evaporator temperature for a given gas cooler exit temperature as reported in previous studies. However, effect of gas cooler temperature on COP is more pronounced than that of effect of evaporator temperature. The similar trend is observed in all chosen configurations (Figure 9). Further, it is observed that II law efficiency decreases with increase in both gas cooler temperature and evaporator temperature. However, the total exergy destruction increases with increase in gas cooler exit temperature and decrease in evaporator temperature.

In WRC, there is significant reduction in compressor work due to availability of the turbine work. Further, isentropic expansion in turbine brings the gain in specific refrigeration effect. Eventually, the cycle exhibit highest COP. Further, the exergy loss is relatively less in expansion compared to throttling that result higher exergetic efficiency.

The percentage improvement in COP of the chosen configurations with respect to BC is shown in Figure 10. It is observed that percentage improvement in COP is not very appreciable with the modifications: DEC, IHEC and VTEC, 1%, 1% and 3%, respectively. However, there is significant gain with WRC and combined with WRCIHE as 25%, 16%, respectively. It can be seen that the percentage improvement in COP is maximum with WRC followed by WRCIHE and then VTEC. Decrease in throttling losses and work recovery in turbine both contribute to maximize the gain in COP with configuration WRC, whereas presence of suction superheat with the use of IHX in WRCIHE, the relative gain in COP is comparatively less. The percentage gain in II law efficiency is highest with WRC configuration as like COP in comparison to other chosen configurations as shown in Figure 11.

The exergy loss in the respective components of the chosen modified cycles is shown in Figures 12–16. The exergy loss in gas cooler for all the chosen cycles is invariable highest while exergy loss is lowest in evaporator, $<10\%$, at all operating conditions. Exergy loss in compressor and throttle valve is almost same for all configurations. As expected, exergy loss in IHX is quite low.

The exergy loss in the work recovery turbine is significantly lower that results relatively better exergetic efficiency of the cycles WRC and WRCIHE in comparison to other cycles. Further, there is no work recovery in VT and the exergy loss is relatively higher than the throttle valve; however, the II law efficiency of VTEC is better than the cycles with throttles valve but lower than WRC and WRCIHE cycles where work recovery turbine is used as an expansion device.

5. CONCLUSIONS

Steady-state energetic and exergetic analysis and performance simulation of the five modified CO_2 transcritical cycles: DEC, IHEC, WRC, WRCIHE and VTEC are carried out in the present study. All the chosen cycles are simulated employing the EES. Results are compared with the BC. Optimum gas cooler pressure, discharge temperature, optimum COP and II law efficiency contours are developed that facilitate to determine the performance parameters on the chosen gas cooler exit and evaporator temperature. The outcome of the present study is as follows.

- WRC exhibits highest COP and exergetic efficiency among all the five chosen modified cycles.
- All the chosen cycles show the relatively higher COP and lower optimum gas cooler pressure at low gas cooler exit temperature and high evaporator temperature.
- Gas cooler exit temperature is relatively more influential than the evaporator temperature on the system COP, optimum discharge pressure and compressor exit temperature.
- The exergy loss in gas cooler for all the chosen cycles is invariable highest while exergy loss is lowest in evaporator.
- Modifications with work recovery turbine and work recovery turbine with IHX are relatively more effective than that of other chosen modifications. However, use of vortex tube is also moderately effective to enhance the performance of the CO_2 transcritical cycle.

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