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Exergy Analysis and Parametric Study of Recuperated Supercritical Carbon Dioxide Rankine Cycle

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Abstract

Supercritical carbon dioxide shows significant advantages as a working fluid used in waste heat recovery applications. In this study, a recuperated supercritical carbon dioxide Rankine cycle is designed for waste heat recovery purpose. Component wise and overall exergy analysis, effect of various thermodynamic parameters on exergy destruction and overall exergy efficiency are performed. It is shown that key thermodynamic parameters such as heat source temperature, turbine inlet temperature and pressure, pump inlet temperature and terminal temperature difference have significant effect on overall exergy efficiency of the cycle.

Keywords: Supercritical carbon dioxide, waste heat recovery, recuperated Rankine cycle, exergy analysis.

1. Introduction

Waste heat recovery is gaining interest in recent years as it can increase a thermal system's power output and reduce environmental pollution caused by the system. Research indicates a 20 percent fulfillment of domestic electricity demand and simultaneously 20 percent reduction in greenhouse gas emission by waste heat recovery in the U.S. The recovery process can be performed directly without using heat exchanger or indirectly with the use of a heat exchanger where suitable fluids are used to recover the heat. Indirect methods are more popular due to their location, contamination and treatment consideration.
Supercritical carbon dioxide cycle is similar to a steam Rankine cycle in layout and similar to a gas turbine system in component wise design where carbon dioxide is used above its critical point (7.38 MPa pressure and 31.1℃ temperature) so that it doesn't change its phase but rather undergoes drastic density change over a wide range of temperature and pressure. Supercritical carbon dioxide cycle shows better performance than most other steam and organic Rankine cycle as *pinch point* phenomenon doesn't occur here. The turbomachines used in the cycle are compact because of the higher density of the working fluid. The availability and low cost, non-toxic, non-corrosive and non-flammable nature of carbon dioxide also make it an appropriate working fluid to be used in power cycles. The cycle shows higher thermal efficiency and less emission characteristics than organic Rankine cycle in heat recovery application.
Dostal [1] designed a recompression supercritical carbon dioxide cycle with turbine inlet temperature of 550-750 °C. Wang [2] used a simple Rankine cycle model to use supercritical carbon dioxide to extract waste heat from low temperature source. He also optimized the thermodynamic parameters to get maximum efficiency using genetic algorithm. D bella [3] showed the potentiality of supercritical carbon dioxide cycle to replace steam Rankine cycle to extract heat from a source of over 450 °C. Kimzey [4] compared performance of different supercritical carbon dioxide bottoming cycle layouts which can potentially maximize the power output from exhaust gas of current gas turbines. Y.Ahn [5] presented a performance comparison of 12 different supercritical carbon dioxide cycle layouts where the recompression cycle had shown the best efficiency but it required the largest recuperator size. The layout with preheating showed the minimum efficiency but required minimum recuperator size. Crespi [6] analyzed twelve different supercritical carbon dioxide cycles at four different turbine inlet temperature and pressure ratios. The results had shown that most cycles reached the peak thermal efficiency at a pressure higher than 40 MPa when the turbine inlet temperature was higher than 550 °C.
Up to the present none of the published studies analyzed the effect of recuperator on the performance of a supercritical carbon dioxide Rankine cycle in waste heat recovery application. So the aim of this study is exergy analysis of a recuperated supercritical carbon dioxide Rankine cycle. The effects of various thermodynamic parameters on overall exergy efficiency of the cycle have also been shown.

2. Thermodynamic Modelling of the cycle

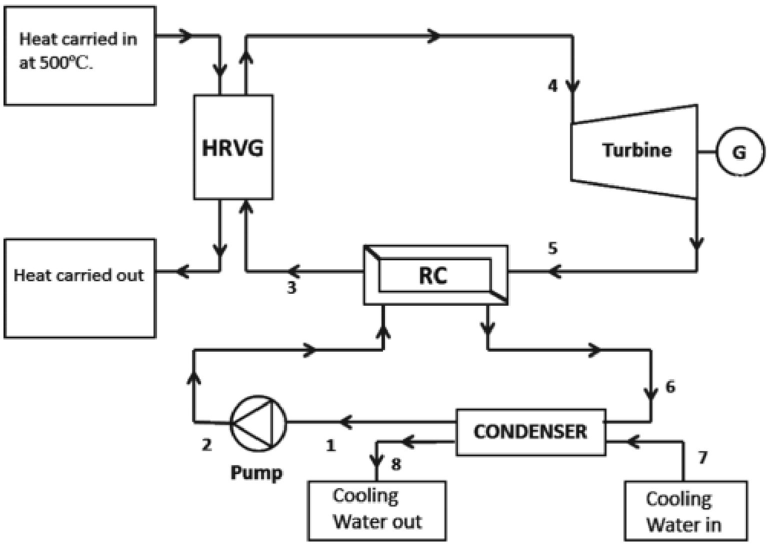


Fig. 1. Schematic diagram of the cycle of present study.

The schematic diagram of the cycle of the present study has been shown in **Fig. 1**. The cycle is modeled like a Rankine cycle with a recuperator assuming the heat source as the waste heat to be recovered from a power plant (not shown in the figure). The cycle consists of the following process:
Process 3-4: A constant pressure heat absorption process in heat recovery vapor generator (HRVG).
Process 4-5; A non-isentropic expansion process in the turbine.
Process 5-6: A constant pressure heat rejection process in the recuperator.
Process 6-1; A constant pressure heat rejection process in the condenser.
Process 1-2; A non-isentropic compression process in the pump
Process 2-3; A constant pressure heat absorption process in the recuperator.
For the cycle's thermodynamic analysis following assumptions are made:
1. The system is in steady state and heat loss from the components to the environment and pressure loss at pipes of the system are negligible.
2. The condenser outlet state is saturated liquid.
3. In recuperator terminal temperature difference (minimum temperature difference between hot and cold gases) occurs at point 2.
Turbine inlet state or HRVG outlet state can be obtained when turbine inlet temperature and pressure are given. From the isentropic efficiency of the turbine and pump and from terminal temperature difference of the recuperator, inlet and outlet conditions of the condenser, pump and recuperator can be obtained.
In HRVG, from energy balance the heat absorbed by the working fluid is,

$$Q_{in} = m_g \left(h_{in,g} - h_{out,g} \right) = m \left(h_3 - h_4 \right) \tag{1}$$

Where g indicates characteristics of exhaust gas, h is enthalpy and m indicates mass flow rate. The heat transfer process is isobaric and occurs at a pressure which is same as turbine inlet pressure.
The isentropic efficiency of the turbine can be expressed as,

$$\eta_{TBN} = \frac{h_4 - h_5}{h_4 - h_{5s}} \tag{2}$$

Where s denotes the property if the state is at isentropic condition.
Considering the turbine to be adiabatic, the work output from the turbine can be expressed as,

$$W_{TBN} = m \left(h_4 - h_5 \right) \tag{3}$$

In condenser, heat rejection of the working fluid to the cooling water is,

$$Q_{out} = m \left(h_6 - h_1 \right) = m_w \left(h_8 - h_7 \right) \tag{4}$$

Where w indicates the characteristics of cooling water. The process is isobaric and occurs at a pressure which is same as turbine outlet pressure.
The isentropic efficiency of the turbine can be expressed as,

$$\eta_{PUMP} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{5}$$

The pump work can be expressed as,

$$W_{PUMP} = m \left(h_2 - h_1 \right) \tag{6}$$

In the recuperator unused heat from the turbine output is recovered by heating the working fluid before it enters the HRVG. From the energy balance concept it can be shown that,

$$h_3 - h_2 = h_5 - h_6 \quad (7)$$

The heat exchange process occurs at two different pressure.

The state point of the recuperator can be obtained from terminal temperature difference which is assumed to occur between point 2 and 6, which is similar to the study of A.Kadse [7]. From this concept it can be said that,

$$T_6 = T_2 + T_{tt} \quad (8)$$

Where T_{tt} indicates terminal temperature difference.

The net work output from the cycle can be described as,

$$W_{net} = W_{TBN} - W_{PUMP} \quad (9)$$

So from the first law of thermodynamics, the thermal efficiency or first law efficiency of the cycle is,

$$\eta_t = \frac{W_{net}}{Q_{in}} \quad (10)$$

To analyze the efficiency or performance of a thermodynamic cycle, second law efficiency or exergy efficiency is important as it includes the concept of irreversibility.

If chemical, kinetic and potential exergies are neglected, the exergy of a state point can be expressed as,

$$E_i = m\,[(\,h_i - h_0\,) - T_0(s_i - s_0)] \quad (11)$$

Where '0' subscript denotes environmental condition.

The exergy destruction for a thermodynamic cycle is,

$$I = \sum_{in} E - \sum_{out} E \quad (12)$$

So, in HRVG the exergy destruction is,

$$I_{HRVG} = E_{in} + E_3 - E_4 - E_{out} \quad (13)$$

In turbine the exergy destruction is,

$$I_{TBN} = E_4 - W_{TBN} - E_5 \quad (14)$$

In condenser the exergy destruction is,

$$I_{CND} = E_6 + E_7 - E_1 - E_8 \quad (15)$$

In pump the exergy destruction is,

$$I_{PUMP} = W_{PUMP} - E_1 - E_2 \quad (16)$$

In recuperator the exergy destruction is,

$$I_{REC} = E_5 + E_2 - E_6 - E_3 \quad (17)$$

So, the overall exergy efficiency or the second law efficiency is,

$$\eta_{exg} = \frac{E_{in} - \sum I - E_{loss}}{E_{in}} \quad (18)$$

Where E_{loss} denotes exergy loss carried by exhaust gas and cooling water.

For simulation purpose several parameters have been assumed. Waste heat source in the HRVG has been considered as 100% dry air having a temperature of 500 ℃ and mass flow rate of 640 kg/s which is similar to the exhaust of gas turbine of NWPGCL, Khulna. Pump inlet or condenser outlet state is assumed to be saturated liquid. Other parameters which are assumed for simulation purpose are listed in **Table 1**.

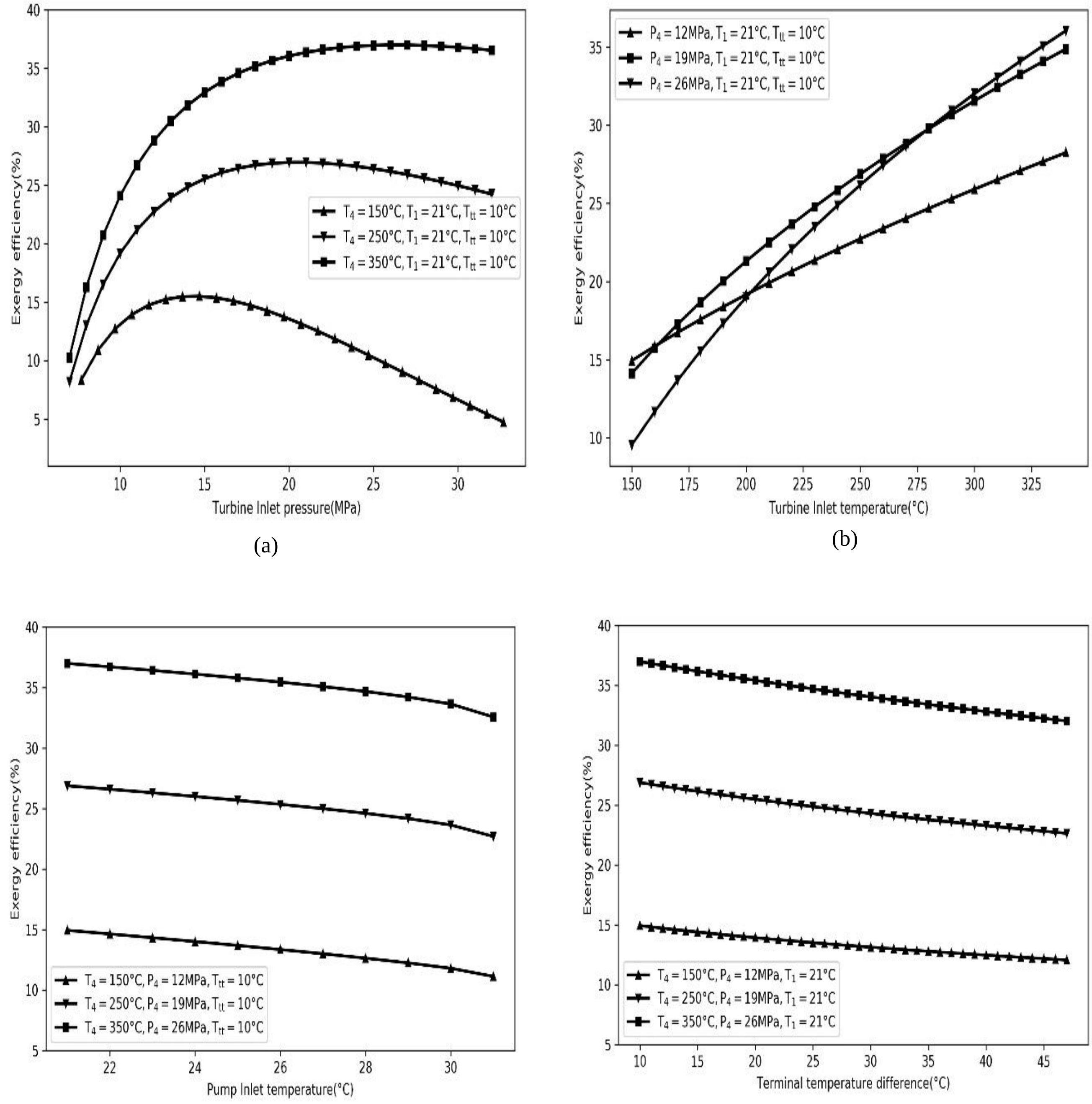
Table 1. Initial parameters for simulating the cycle

Environment temperature (℃)	15
Environment pressure (kPa)	101.325
Isentropic efficiency of turbine (%)	75
Isentropic efficiency of pump (%)	70
Cooling water inlet temperature (℃)	25
Cooling water mass flow rate (kg/s)	2000

3. Result and discussion

The simulation process is performed by a program written in Python programming language. All thermodynamic properties were calculated using CoolProp 6.3.0[8] which is an open source thermodynamic and transport properties database.

According to the first law of thermodynamics work and heat are equivalent and this idea doesn't include the concept of irreversibility. But in second law of thermodynamics heat and work can be differentiated by irreversibility.



(a) (b) (c) (d)

Fig. 2. (a) Variation of exergy efficiency with turbine inlet pressure, (b) Variation of exergy efficiency with turbine inlet temperature, (c) Variation of exergy efficiency with pump inlet temperature, (d) Variation of exergy efficiency with terminal temperature difference.

So, in this present study second law efficiency or exergy efficiency is emphasized. Effects of key thermodynamic parameters such as turbine inlet temperature and pressure, pump inlet temperature and terminal temperature difference on overall exergy efficiency are calculated. The calculated results are shown from **Fig. 2(a)** to **Fig. 2(d)**.

The effect of turbine inlet pressure on the overall exergy efficiency of the cycle for three different turbine inlet temperatures is shown in **Fig. 2(a).** The other parameters are kept constant as described in **Table 1**, pump inlet temperature is assumed 6℃ higher than environmental temperature while terminal temperature difference is kept at 10℃. From the figure it can be seen that for all three turbine inlet temperature values exergy efficiency first increases with increasing turbine inlet pressure. For a certain turbine inlet pressure it reaches a maximum value. With the increase of turbine inlet temperature the value of turbine inlet pressure for maximum efficiency increases. It is known that the net exergy output of the cycle is the difference between turbine work output and pump work input. With the increase of turbine inlet pressure the pressure ratio across the turbine increases which results in increased enthalpy drop. As a result turbine work output and overall exergy efficiency increase at first with increasing pressure. But increasing pressure ratio is also responsible for decreasing mass flow rate in HRVG. After a certain value of turbine inlet pressure, enthalpy decreased by decreasing mass flow rate exceeds the enthalpy gain by increased pressure ratio. So total exergy output and exergy efficiency decrease afterward. It is also seen that with the increase of turbine inlet temperature overall exergy efficiency increases and the curves become comparatively flatter. Increased enthalpy for all state point at an increased temperature is the reason behind this phenomena. It also indicates that with the increase of turbine inlet temperature the effect of decreased mass flow rate is shifted to the right.

The effect of turbine inlet temperature on overall exergy efficiency of the cycle is shown in **Fig. 2(b)** keeping other variables constant as stated in **Fig. 2(a)** for three different turbine inlet pressures. It is seen from the figure that with the increase of turbine inlet temperature overall exergy efficiency of the cycle also increases which is explained before. Unlike in **Fig. 2(a)** the effect of turbine inlet pressure is different for different values of turbine Inlet temperature. The exergy efficiency is lower for higher turbine inlet pressure and lower turbine inlet temperature. With the increase of turbine inlet temperature, higher inlet pressure gains higher efficiency. Due to this non-linear nature optimization of the parameters are necessary for optimal exergy gain.

The effect of pump inlet temperature on overall exergy efficiency of the cycle is shown in **Fig. 2(c)** keeping other variables constant as stated in **Fig. 2(a)** for three different turbine inlet pressure and temperature. It can be seen from the figure that for a fixed turbine inlet temperature and pressure increasing pump inlet temperature has little effect on overall exergy efficiency which decreases with the increase of pump inlet temperature. So lowering pump inlet temperature produces higher efficiency. But it shouldn't be lowered beyond a minimum value as condensation occurs beyond that value and exergy is lost for condensation which is not desirable.

The effect of terminal temperature difference on overall exergy efficiency of the cycle is shown in **Fig. 2(d)** keeping other variables constant as stated in **Fig. 2(a)** for three different turbine inlet temperature and pressure. From the figure it can be seen that increasing terminal temperature difference has little effect on overall exergy efficiency and efficiency decreases with the increase of terminal temperature difference. So the cycle should be operated at minimal terminal temperature difference to get maximum efficiency.

4. Conclusion

In this present study a recuperated supercritical carbon dioxide Rankine cycle has been proposed for waste heat recovery purpose. As first law efficiency doesn't include the concept of irreversibility, second law efficiency or exergy efficiency has been emphasized to analyze the cycle. Exergy analysis of the cycle and effects of various thermodynamic parameter on overall exergy efficiency have been discussed. From the discussion above, it can be concluded that key thermodynamic parameters such as turbine inlet temperature, turbine inlet pressure, pump inlet temperature and terminal temperature difference have significant effects on the overall exergy efficiency of the cycle. Among them increasing pump inlet temperature or terminal temperature difference has a reduction effect on overall exergy efficiency. Though variation of them create little variation of exergy efficiency, a minimum value is required for efficient operation of the cycle. Unlike them, variation of turbine inlet temperature and pressure create significant variation of the overall exergy efficiency. With the increase of turbine inlet temperature exergy efficiency increases but it reaches maximum value and then decreases with the increase of turbine inlet pressure.

5. Recommendation for future extension

In this study a recuperator has been used for better cycle performance. Suitable additional components can be added for higher efficiency. Design of the heat exchanger can also be discussed for withstanding high pressure. More compact design of the pump, heat exchanger etc. can be discussed and analyzed. Optimization of the cycle to gain optimal exergy efficiency isn't discussed in this study. This can be performed keeping focus on turbine inlet temperature and pressure. Turbine blade design for significant output can also be discussed.

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