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Parametric analysis of a reheat carbon dioxide transcritical power cycle using a low temperature heat source

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Abstract. CO_2 transcritical rankine power cycle has been widely investigated recently, because of its better temperature glide matching between sensible waste heat source and working fluid in vapor generator, and its desirable qualities, such as moderate critical point, little environment impact and low cost. A reheat CO_2 transcritical power cycle with two stage expansion is presented to improve baseline cycle performance in this paper. First law analysis is carried out to investigate parametric effects on reheat cycle performance. The main results show that reheat cycle performance is sensitive to the variation of medium pressures and the optimum medium pressures exist for maximizing work output and thermal efficiency, respectively. Reheat cycle is compared to baseline cycle under the same conditions. More significant performance improvements by reheat are obtained at lower turbine inlet temperatures and higher maximum cycle pressure. Work output improvement is much higher than thermal efficiency improvement, because extra waste heat is required to reheat CO_2, which reduces the thermal efficiency. It is found that reheat cycle has great potential to improve thermal efficiency and especially work output of a CO_2 transcritical power cycle using a low-grade heat source.

Keywords: Transcritical cycle, Carbon dioxide, Reheat, Low-grade heat source, System optimization

1. Introduction

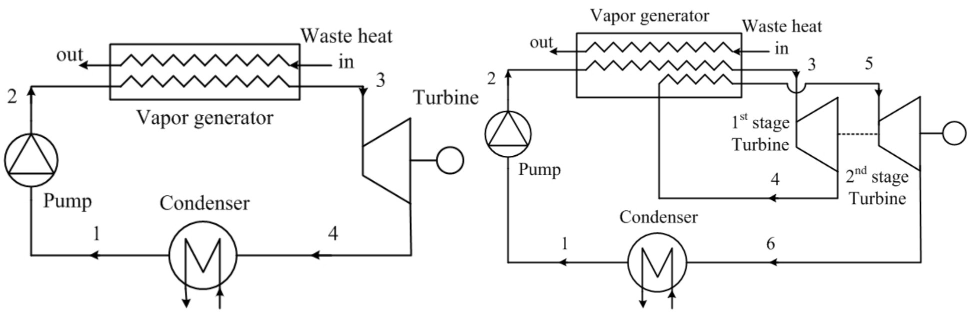
The utilization of low-grade industrial waste heat becomes more attractive due to the world increasing electricity demand. Supercritical power cycles show high potentials to recover such low grade heat, because working fluid temperature glide above the critical points provides a better temperature profile match in the vapor generator with less irreversibility.

Most previous studies focus on thermodynamic analysis and optimization of conventional carbon dioxide transcritical power cycles. Chen et al. [1] found that the carbon dioxide transcritical power cycle had a slightly higher work output than did an organic rankine cycle (ORC) with equal mean thermodynamic heat rejection temperature. Zhang et al. [2] studied a similar cycle powered by solar energy for both power and heat generation. They also conducted experimental study to validate the feasibility of this cycle [3]. Cayer et al. [4] optimized specific work output of a transcritical cycle by investigating the turbine inlet temperature on cycle performance. Two transcritical cycles were optimized by Baik et al. [5] for power output through examining turbine inlet temperatures and pressures. Wang et al. [6] used a generic algorithm and an artificial neural network to optimize a few important parameters with exergy efficiency as an objective function.

Previous studies indicated that higher turbine inlet temperatures produced more work output as well as better thermal efficiency. This is because CO_2 exhibits a steeper slope for the isobaric curve of the high-pressure region than in the low pressure region, and thus specific work output is increased when carbon dioxide is expanded at a fixed expansion ratio but a higher inlet temperature [7]. Thus, this indicates the performance of transcritical cycle can be improved by reheat cycle where CO_2 is only firstly expanded to medium pressure and then reheated to maximum temperature before 2nd stage expansion. Dostal [8] investigated a reheat CO2 brayton cycles and analysed its improvement over baseline cycle. However, none of the published studies on transcritical CO2 power cycles have investigated effects of reheat on cycle performances and compares to baseline cycles without reheat. Thus, in this paper, a reheat transcritical carbon dioxide cycle is introduced and optimized.

2. System description and modeling

Fig. 1(a) and (b) show schematics of a baseline cycle and a reheat cycle, respectively. In the reheat cycle, superheated vapor from the vapor generator (3) is expanded through 1st stage turbine to a medium pressure (P_m). Then, it is reheated again in the vapor generator and expanded through 2nd stage turbine to a low pressure (P_c), and finally the vapor is completely condensed to saturated liquid. A regenerator is not included since recent studies [9, 10] have shown that a regenerator only produces small improvement of thermal and exergetic efficiency but has little influence on the net power output. For the present study, the heat source is industrial waste water at a temperature of 100 ºC. The cooling water is at 10 ºC. The following general assumptions are formulated for this study without losing generosity: each component is considered as a steady-state steady-flow system; kinetic and potential energies as well as the heat and friction losses are neglected; pump and the turbine isentropic efficiencies are both set to be 0.8; saturated liquid is assumed at the condenser exit.



(a) Baseline cycle (b) Reheat cycle

Fig. 1. Schematic diagrams of supercritical CO_2 power cycle

The first law of thermodynamics is used for energy analysis. The thermal efficiency η and specific work output W_{net} are final results for comparison. For the reheat cycle, the model and equations for the different components are the following:

For the pump:

$$\eta_P = \frac{h_{2,s} - h_1}{h_2 - h_1} \tag{1}$$

$$W_P = \dot{m}_c (h_2 - h_1) \tag{2}$$

For the two turbines:

$$\eta_{t,1} = \frac{h_3 - h_4}{h_3 - h_{4,s}} \tag{3}$$

$$\eta_{t,2} = \frac{h_5 - h_6}{h_5 - h_{6,s}} \tag{4}$$

$$W_{t,1} = \dot{m}_c (h_3 - h_4) \tag{5}$$

$$W_{t,2} = \dot{m}_c (h_5 - h_6) \tag{6}$$

For the vapor generator:

$$Q_{in} = \dot{m}_c \left[(h_3 - h_2) + (h_5 - h_4) \right] \tag{7}$$

For the condenser:

$$Q_{out} = \dot{m}_c \left(h_6 - h_1 \right) \tag{8}$$

For the specific work output:

$$W_{net} = \frac{W_{t,1} + W_{t,2} - W_P}{\dot{m}_c} \tag{9}$$

For the thermal efficiency of the reheat cycle:

$$\eta = \frac{W_{t,1} + W_{t,1} - W_p}{Q_{in}} \qquad (10)$$

3. Results and Discussion

Fig. 2 shows the T-s diagram for reheat cycle at a high pressure P_h=12 MPa. Fig. 3 shows the effect of medium pressure P_m on the specific work output and thermal efficiency at a turbine inlet temperature of 90 ºC. Especially, for the case of P_m=P_h, this indicates only 2nd turbine produces work and no reheat is reqired from vapor generator. Thus, the reheat cycle works as the baseline cycle with the same system performance. It can be seen that the specific work output and thermal efficiency become higher as increase of P_m, and exibit maximum values at different optimal P_m. In addition, the optimal P_m at maximum work output is lower than that at maximum efficiency. This can be explained that produced work from 1st turbine dercreases but from 2nd turbine increases as increase of P_m, thus maximum work output is about 17.77 kJ/kg at the optimal P_m=8.49 MPa. However, higher P_m reduces the reheat input, because the temperature after the 1st stage expansion is increased due to the lower expansion ratio and less amount of waste heat is required to reheat CO_2. The required total heat input decreases still relatively faster than work output does, so maximum thermal efficiency occurrs at a slight higher P_m.

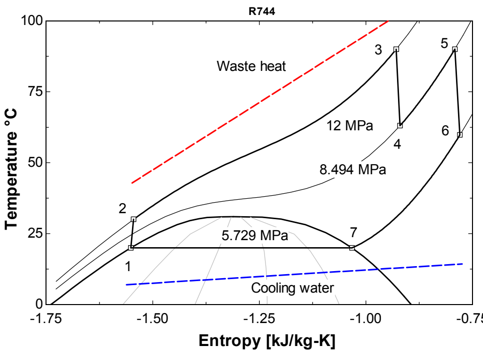


Fig. 2. T-s diagram for the reheat cycle

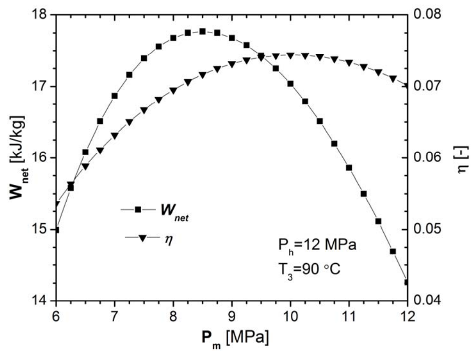
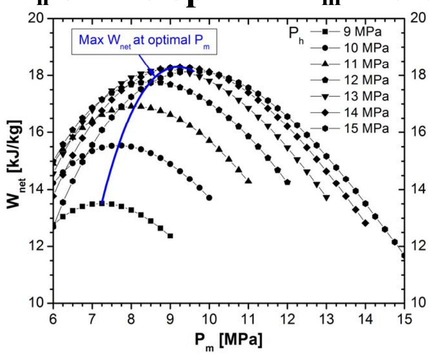


Fig. 3. Effect of P_m on reheat cycle performance

3.1 Effects of P_h on the optimal P_m and cycle performance



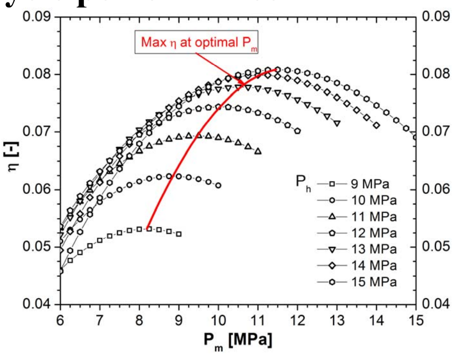


Fig. 4. Optimal P_m for maximum W_{net} Fig. 5. Optimal for P_m for maximum η

Fig. 4 and 5 show effects of varied high pressures P_h on the cycle performance and optimal medium pressure P_m for maximum work output and thermal efficiency, respectively. As expected, optimal medium pressures P_m are higher as increase of high pressure P_h. Maximum work output W_{net} rises dramatically when high pressure P_h increases from 9 MPa to about 12 MPa, and then a maximum W_{net} about 18.3 kJ/kg is obtained at P_h=13.87 MPa and P_m=9.208 MPa. Thermal efficiency increases monotonically with P_h at the given range in this study. Efficiency reaches the maximum value of 0.08. For comparison of baseline and reheat cycles, the improvement of W_{net} and η can be illustrated in Fig. 6 and Fig. 7, repectively. Larger enhancement by reheat for W_{net} and η is achieved at higher P_h. This is because high-pressure isobari curve is steeper than at low pressure. For the case of P_h=15 MPa, W_{net} and η are increased by approximately 50% and 15%, respectively. Additionally, it can be noticed that thermal efficiency improvement is relatively low compared to work output, because extra reheat input counteracts increased work output.

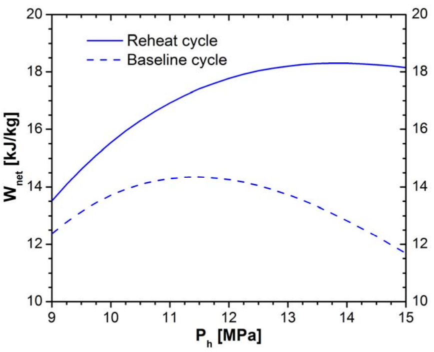


Fig. 6. W_{net} maximum improvement by reheat

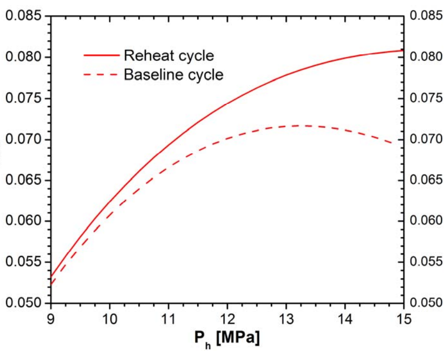


Fig. 7. η maximum improvement by reheat

3.2 Effects of turbine inlet temperature T_3 on the cycle performance

Fig. 8 and Fig. 9 illustrate effects of turbine inlet temperatures on the performance of the reheat cycle. Obviously, for the fixed P_h, higher turbine inlet temperature leads to larger W_{net} and higher η. Compared to others' studies for baseline cycles, dependence of reheat cycle performnace on turbine inlet temperature is similar. In order to quanlify the improvement by means of reheat, improvement factors for W_{net} and η are defined as:

$$\varepsilon_W = \frac{\left(W_{net}\right)_r - \left(W_{net}\right)_b}{\left(W_{net}\right)_b} \tag{11}$$

$$\varepsilon_\eta = \frac{(\eta)_r - (\eta)_s}{(\eta)_s} \tag{12}$$

Where subcripts r and b denote reheat cycle and baseline cycle, respectively.

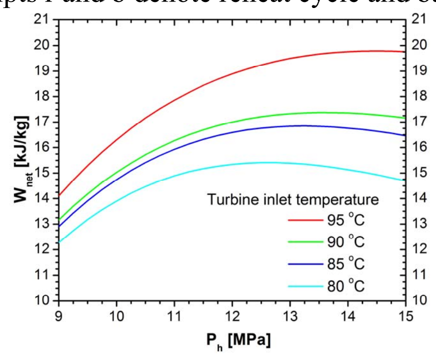


Fig. 8. work output versus P_h at different T_3

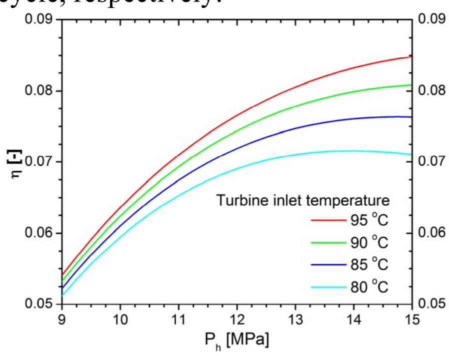


Fig. 9. Thermal efficiency versus P_h at different T_3

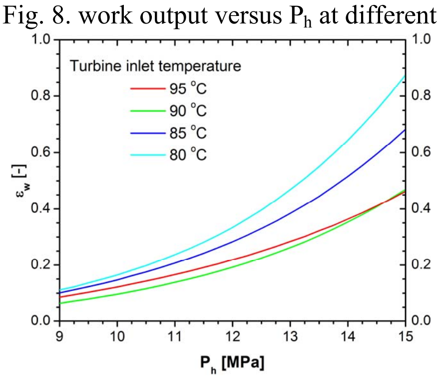


Fig. 10. W_{net} improvement factor versus P_h

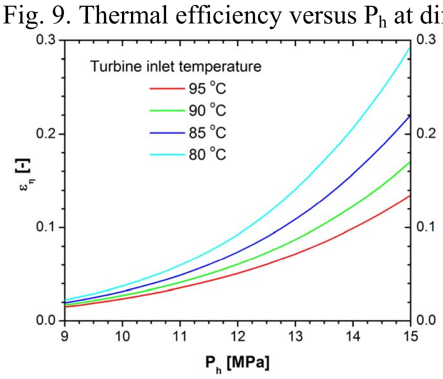


Fig. 11. η improvement factor versus P_h

Results are shown in Fig. 10 and Fig. 11, respectively. It can be seen that higher improvements of both W_{net} and η are obtained at a lower turbine inlet temperature and a higher P_h. Under present conditions, the maximum improvements of W_{net} and η are approximately 90% and 30%, respectively. Additionally, the improvement of W_{net} is more significantly than that of η. This may indicate some potential applications of reheat cycle. Firstly, resource of waste heat is relatively cheap and even free like waste streams, thus achieveing high thermal efficiency is less important than genrating the maximum specific net work output. Secondly, the maximum temperature of waste heat is relatively low, therefore limiting the turbine inlet temperature, since both improvement factors are higher at lower turbine inlet temperature.

4. Conclusions

In this paper, a reheat transcritical CO_2 power cycle has been investigated and compared to a non-reheat baseline cycle with respect to net power output, thermal efficiency and exergy efficiency. It is found that medium pressure before 2nd stage expansion is an important parameter to determine reheat cycle performance. For the given cycle high pressure and turbine inlet temperature, two different optimal medium pressures exist corresponding to maximum work output and thermal efficiency, respectively. The optimal pressure for maximum work output is higher than that for maximum thermal efficiency under the same conditions. Compared to baseline cycle, larger improvements of work output and thermal efficiency by reheat cycle are obtained at a lower turbine inlet temperature and/or larger high cycle pressure. The maximum improvements for work and thermal efficiency are approximately 90% and 30%, respectively. The overall results show that work output is increased by reheat more significantly than thermal efficiency. This indicates it is proper to use the reheat cycle under which condition maximizing work output is most concerned.

In summary, the reheat with two stage expansion has great potential to improve work output and thermal efficiency of a CO_2 transcritical power cycle using a low-grade heat source.

Nomenclature

h	enthalpy (kJ/kg)	*Subscripts*	
P	pressure (MPa)	b	baseline cycle
Q	heat input (kW)	C	condenser
T	temperature (ºC)	h	high pressure
W	power (kW)	m	medium pressure
Greek letters		net	net work output
ε	improvement factor	p	pump
η	thermal efficiency	r	reheat cycle
η_p	Pump isentropic efficiency	t	turbine
η_t	Turbine isentropic efficiency	VG	vapor generator

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