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Exergy analysis for combined regenerative Brayton and inverse Brayton cycles

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Abstract
This paper presents the study of exergy analysis of combined regenerative Brayton and inverse Brayton cycles. The analytical formulae of exergy loss and exergy efficiency are derived. The largest exergy loss location is determined. By taking the maximum exergy efficiency as the objective, the choice of bottom cycle pressure ratio is optimized by detailed numerical examples, and the corresponding optimal exergy efficiency is obtained. The influences of various parameters on the exergy efficiency and other performances are analyzed by numerical calculations.

Keywords: Regenerative Brayton cycle; Inverse Brayton cycle; Exergy analysis; Exergy loss; Exergy efficiency; Optimization.

1. Introduction
Nowadays, in order to meet the demands of energy-saving and environmental protection, people want to construct new energy and power plants which could gain better performance. Because of their high efficiency and advances in the technologies of the individual components, combined-cycle power plants have been applied widely in recent years. Steam and gas turbine combined cycles are considered the most effective power plants [1]. The thermal efficiency of these cycle types exceeded 55 percent several years ago and is now at approximately 60 percent. Also these cycle types' application is becoming more and more common in mid and large scale power production due to their high efficiency and reliability. In order to increase the power output, a hybrid gas turbine cycle (Braysson cycle) was proposed based on a conventional Brayton cycle for the high temperature heat addition process and Ericsson cycle for the low temperature heat rejection process, and the first law analysis of the Braysson cycle was performed by Frost et al. [2] in 1997. Furthermore, the exergy analysis of the Braysson cycle based on exergy balance was performed by Zheng et al. [3] in 2001. Fujii et al. [4] studied a combined-cycle with a top cycle (Brayton cycle) and a bottom cycle consisting of an expander followed by an inter-cooled compressor in 2001. They found that when fixed the bottom cycle pressure ratio to 0.25 bar could avoid a rapid increase in gas flow axial velocity effectively. They also proposed the use of two parallel inverse Brayton cycles instead of one in order to reduce the size of the overall power plant. Bianchi et al. [5] examined a combined-cycle with a top cycle (Brayton cycle) and a bottom cycle (an inverted Brayton cycle in which compress to atmospheric pressure) in 2002. Agnew et al. [6] proposed combined Brayton and inverse Brayton cycles in 2003, and performed the first law analysis of the combined cycles. They indicated that the optimal expansion pressure of the inverse Brayton cycle is 0.5 bar for optimum performance. The exergy analysis and optimization of the combined Brayton and inverse Brayton cycles were performed

by Zhang et al. [7] in 2007. They indicated that exergy loss of combustion is the biggest in the cycle and followed by heat exchanger. Alabdoadaim et al. [8-10] studied the combined Brayton and inverse Brayton cycles (the base cycle) with their developed configurations. They revealed that using two parallel inverse Brayton cycles as bottom cycles can realize maximum energy utilization and reduce the physical sizes of the bottom cycle components. Furthermore, in order to use the evolved heat of the base cycle, Rankine cycle is added as one bottom cycle. Zhang et al. [11] performed exergy analysis of the combined Brayton and two parallel inverse Brayton cycles in 2009. Alabdoadaim et al. [10] also revealed that using regenerative Brayton cycle as top cycle can obtain higher thermal efficiency than the base cycle but smaller work output using the first law analysis method.

Analysis of energy and power systems based on the First Law is usually used when proposing new cycle configurations. In order to know more performance of new configurations, the exergy analysis should be carried out followed. The exergy analysis method [12-30] provides a more accurate measurement of the actual inefficiencies for the system and a more accurate measurement of the system efficiency for open cycle systems.

In this paper, the exergy analysis for combined regenerative Brayton and inverse Brayton cycles proposed in Ref. [10] is performed. The purposes of the study are to determine the largest exergy loss location and optimize the pressure ratio of the compressor of the regenerative Brayton cycle, which could obtain better exergy performance.

2. Cycle model [10]

The proposed system in Ref. [10] is shown in Figure 1. It is constructed from a top cycle (regenerative Brayton cycle) and a bottom cycle (inverse Brayton cycle). Figure 2 shows T-s diagrams of the system. Process 1-2 is an irreversible adiabatic compression process in the compressor 1. Process 2-3 is an absorbed heat process in the regenerator. Process 3-4 is an absorbed heat process in the chamber. Process 4-5 is an irreversible adiabatic expansion process in the turbine 1. Process 5-6 is an evolved heat process in the regenerator. Process 6-7 is an irreversible adiabatic expansion process in the turbine 2. Process 7-8 is an evolved heat process in the heat exchanger. Process 8-9 is an irreversible adiabatic compression process in the compressor 2.

The top cycle is used as a gas generator to power the bottom cycles. The purpose of the turbine in the top cycle is solely to power the compressor. The power output of the combined cycle is totally produced by the bottom cycle. The thermal efficiency of the system was analyzed in Ref. [10].

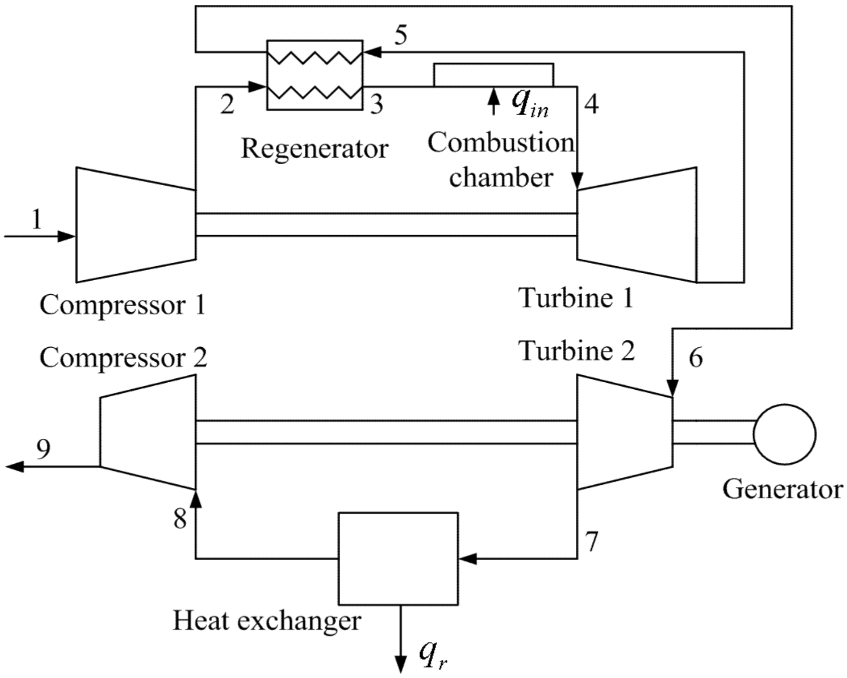


Figure 1. System layout of the combined cycle

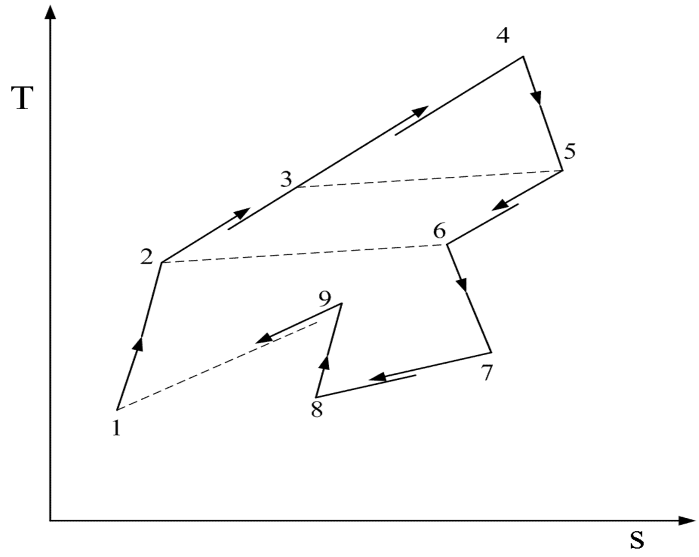


Figure 2. T-s diagram for the combined cycle

3. Exergy analysis and optimization

The following assumptions are made for simplicity and manipulating analytical expressions: The working fluid has constant specific heat ratio k ($k = c_P / c_V = 1.4$). The mass flow rate $\dot{m}$ is fixed as 1 kg/s.

For the system operating in a steady state, the general exergy balance equation is given in Refs. [12-16, 21]. After making an exergy balance equation, the expression of the exergy balance equation can be obtained for each component, respectively.

For the compressor 1, the following expression can be obtained:

$$w_{c1} = (e_2 - e_1) + e_{D.c1} \tag{1}$$

where $w_{c1} = c_p T_1 \psi_{c1} / \eta_{c1}$ is specific work consumed of the compressor 1, c_p is constant-pressure specific heat, T is temperature, $\psi_{c1} = \varphi_{c1}{}^m - 1$, $m = (k-1)/k$, $\varphi_{c1} = P_2 / P_1$ is pressure ratio of compressor 1, P is pressure, e is exergy, η_{c1} is the efficiency of the compressor 1, and $e_{D.c1} = c_p T_1 \left[\mathrm{In}(1 + \psi_{c1}/\eta_{c1}) - m \mathrm{In} \varphi_{c1} \right]$ is exergy loss of the compressor 1.

For the turbine 1, the following expression can be obtained:

$$w_{t1} + (e_5 - e_4) + e_{D.t1} = 0 \tag{2}$$

where $w_{t1} = c_p T_1 \tau_1 \psi_{t1} \eta_{t1}$ is specific work output of the turbine 1, $\psi_{t1} = 1 - 1/\varphi_{t1}{}^m$, $\varphi_{t1} = P_4 / P_5$ is pressure ratio of turbine 1, $e_{D.t1} = c_P T_1 \left[\mathrm{In}(1 - \eta_{t1}\psi_{t1}) - m \mathrm{In}(1/\varphi_{t1}) \right]$ is exergy loss of turbine 1, and η_{t1} is efficiency of the turbine 1.

For the turbine 2, the following expression can be obtained:

$$w_{t2} + (e_7 - e_6) + e_{D.t2} = 0 \tag{3}$$

where $w_{t2} = c_p T_1 \eta_{t2} \psi_{t2} (\tau_1 - \psi_{c1}/\eta_{c1})$ is specific work output of the turbine 2, $\psi_{t2} = 1 - 1/\varphi_{t2}{}^m$, $\varphi_{t2} = P_6 / P_7$ is pressure ratio of the turbine 2, $\tau_1 = T_4 / T_1$ is temperature ratio, η_{t2} is efficiency of the turbine 2, and $e_{D.t2} = c_p T_1 \left[\mathrm{In}(1 - \eta_{t2}\psi_{t2}) - m \mathrm{In}(1/\varphi_{t2}) \right]$ is exergy loss of the turbine 2.

For the combustion chamber, the following expression can be obtained:

$$e_f = (e_4 - e_3) + e_{D.f} \tag{4}$$

where $e_f = q_{in}/\eta_b$ is exergy of fuel, η_b is efficiency of combustion chamber, $q_{in} = h_4 - h_3 = c_p T_1 \left[\tau_1 - E_R \tau_1 (1-\eta_{t1}\psi_{t1}) - (1-E_R)(\psi_{c1}+\eta_{c1})/\eta_{c1}\right]$ is absorbed heat of the system, h is enthalpy, $e_{D.f} = c_p T_1 \left\{ \text{In}\left[\frac{\tau_1 \eta_{c1}}{(1-E_R)(\psi_{c1}+\eta_{c1}) + E_R \tau_1 (1-\eta_{t1}\psi_{t1})\eta_{c1}} \right] - m\text{In}D_2 \right\} + c_p T_1 (1/\eta_b - 1)\left[\tau_1 - (1-E_R)(\psi_{c1}+\eta_{c1})/\eta_{c1} + E_R \tau_1 (1-\eta_{t1}\psi_{t1})\right]$ is exergy loss of the combustion chamber, E_R is effectiveness of the regenerator, $D_2 = 1 - \Delta P_{3-4}/P_3$ is pressure recovery coefficient, and $\Delta P_{3-4} = P_3 - P_4$.

For the regenerator, the following expression can be obtained:

$$e_{D.re} + (e_3 - e_2) + (e_6 - e_5) = 0 \tag{5}$$

where $e_{D.re} = c_p T_1 \text{In}\left[E_R \frac{\tau_1 (1-\eta_{t1}\psi_{t1})\eta_{c1}}{\psi_{c1}+\eta_{c1}} + 1 - E_R \right] \times \left[E_R \frac{\psi_{c1}+\eta_{c1}}{\tau_1 (1-\eta_{t1}\psi_{t1})\eta_{c1}} + 1 - E_R \right] - c_p T_1 m \text{In} D_1 D_3$ is exergy loss of the regenerator, $D_1 = 1 - \Delta P_{2-3}/P_2$ ($\Delta P_{2-3} = P_2 - P_3$) and $D_3 = 1 - \Delta P_{5-6}/P_5$ ($\Delta P_{5-6} = P_5 - P_6$) are pressure recovery coefficients.

For the heat exchanger, the following expression can be obtained:

$$(e_8 - e_7) + e_{HE} = 0 \tag{6}$$

where $e_{HE} = c_p T_1 \text{In}\left\{ (1-\varepsilon) + \frac{\varepsilon}{(1-\eta_{t2}\psi_{t2})\left[E_R(\psi_{c1}+\eta_{c1})/\eta_{c1} + \tau_1 (1-E_R)(1-\eta_{t1}\psi_{t1})\right]} \right\} - c_p T_1 m \text{In} D_4 - c_p T_1 \left\{ \varepsilon - \varepsilon (1-\eta_{t2}\psi_{t2})\left[E_R(\psi_{c1}+\eta_{c1})/\eta_{c1} + \tau_1 (1-E_R)(1-\eta_{t1}\psi_{t1})\right] \right\}$ is exergy loss of the heat exchanger, ε is effectiveness of the heat exchanger, and $D_4 = 1 - \Delta P_{7-8}/P_7$ ($\Delta P_{7-8} = P_7 - P_8$) is pressure-recovery coefficient.

For the compressor 2, the following expression can be obtained:

$$w_{c2} = (e_9 - e_8) + e_{D.c2} \tag{7}$$

where $w_{c2} = c_p T_1 \left\{ (1-\eta_{t2}\psi_{t2})(1-\varepsilon)\left[E_R(\psi_{c1}+\eta_{c1})/\eta_{c1} + \tau_1 (1-E_R)(1-\eta_{t1}\psi_{t1})\right] + \varepsilon \right\} \psi_{c2}/\eta_{c2}$ is specific work consumed of the compressor 2, η_{c2} is efficiency of the compressor 2, $e_{D.c2} = c_p T_1 \left[\text{In}(1 + \psi_{c2}/\eta_{c2}) - m\text{In}\varphi_{c2}\right]$ is exergy loss of the compressor 2, $\psi_{c2} = \varphi_{c2}{}^m - 1$ and $\varphi_{c2} = P_9/P_8$ is pressure ratio of the compressor 2.

For the exhaust gas of the inverse Brayton cycle, the following expression can be obtained:

$$e_9 - e_1 = e_{ex} \tag{8}$$

where $e_{ex} = c_p T_1 \{(1+\psi_{c2}/\eta_{c2})\{(1-\eta_{t2}\psi_{t2})(1-\varepsilon)[E_R(\psi_{c1}+\eta_{c1})/\eta_{c1} + \tau_1 (1-E_R)(1-\eta_{t1}\psi_{t1})] + \varepsilon\} - 1\} - c_p T_1 \text{In}(1+\psi_{c2}/\eta_{c2})\{(1-\eta_{t2}\psi_{t2}) \times (1-\varepsilon)[E_R(\psi_{c1}+\eta_{c1})/\eta_{c1} + \tau_1 (1-E_R)(1-\eta_{t1}\psi_{t1})] + \varepsilon\} D_0^m$ is exergy loss of the exhaust gas, and $D_0 = P_1/P_9$.

For the turbine 1 is solely used to power the compressor 1 ($w_{c1} = w_{t1}$), one can derive the following expression:

$$\varphi_{t1} = \left[\eta_{c1}\eta_{t1}\tau_1 / \left(\eta_{c1}\eta_{t1}\tau_1 - \varphi_{c1}{}^m + 1\right)\right]^{\frac{1}{m}} \tag{9}$$

For the total pressure ratios of expansion and compression are equal ($\varphi_{t2} = D\varphi_{c1}\varphi_{c2} / \varphi_{t1}$), one can derive the following expression:

$$\psi_{t2} = 1 - \frac{\eta_{c1}\eta_{t1}\tau_1}{D^m(\psi_{c1}+1)(\eta_{c1}\eta_{t1}\tau_1 - \psi_{c1})\varphi_{c2}^m} \tag{10}$$

where $D = D_0 D_1 D_2 D_3 D_4$ is total pressure-recovery coefficient.

The specific work and the exergy efficiency of the combined cycle are defined as:

$$\begin{aligned} w = w_{t2} - w_{c2} = c_p T_1 \eta_{t2}\left(1 - a/\varphi_{c2}^m\right)\left[E_R c + b\left(1-E_R\right)\right] - \\ c_p T_1 \left\{\left[1-\eta_{t2}\left(1-a/\varphi_{c2}^m\right)\right]\left[E_R c + b\left(1-E_R\right)\right]\left(1-\varepsilon\right)+\varepsilon\right\}(\varphi_{c2}^m - 1)/\eta_{c2} \end{aligned} \tag{11}$$

$$\eta_E = w/e_f = \frac{\begin{array}{c}\eta_{t2}(1-a/\varphi_{c2}^m)\left[E_R c + b(1-E_R)\right] - (\varphi_{c2}^m - 1)/\eta_{c2} \times \\ \{[1-\eta_{t2}(1-a/\varphi_{c2}^m)][E_R c + b(1-E_R)](1-\varepsilon)+\varepsilon\}\end{array}}{\tau_1 - E_R b - (1-E_R)c}\eta_b \tag{12}$$

where $a = \dfrac{\eta_{c1}\eta_{t1}\tau_1}{D_0{}^m(\psi_{c1}+1)(\eta_{c1}\eta_{t1}\tau_1 - \psi_{c1})}$, $b = \tau_1(1-\eta_{t1}\psi_{t1})$ and $c = 1 + \psi_{c1}/\eta_{c1}$.

To optimize the exergy efficiency, one can derive the following expression from the extremal condition of $\partial\eta_E / \partial\varphi_{c2} = 0$.

The optimal pressure ratio of the compressor 2 correcsponding to the optimal exergy efficiency is:

$$\varphi_{c2opt} = \left\{\frac{a[b(E_R - 1) - cE_R](\varepsilon - 1 + \eta_{c2})\eta_{t2}}{b(E_R-1)(\varepsilon-1)(\eta_{t2}-1) + cE_R(\eta_{t2}-1) + \varepsilon(cE_R - cE_R\eta_{t2} - 1)}\right\}^{1/m} \tag{13}$$

And the optimal exergy efficiency is:

$$\eta_{Eopt} = \eta_b \frac{\begin{array}{c}\eta_{t2}(1-a/\varphi_{c2opt}^m)\left[E_R c + b(1-E_R)\right] - (\varphi_{c2opt}^m - 1)/\eta_{c2} \times \\ \{[1-\eta_{t2}(1-a/\varphi_{c2opt}^m)][E_R c + b(1-E_R)](1-\varepsilon)+\varepsilon\}\end{array}}{\tau_1 - E_R b - (1-E_R)c} \tag{14}$$

The minimum dimensionless total exergy loss is:

$$\begin{aligned} (e_{loss}/(c_P T_1))_{\min} = \frac{\tau_1 - E_R b - \left(1-E_R\right)c}{\eta_b} - \eta_{t2}\left(1-a/\varphi_{c2opt}^m\right)\left[E_R c + \left(1-E_R\right)b\right] + \\ \left\{\left[1-\eta_{t2}\left(1-a/\varphi_{c2opt}^m\right)\right]\left[E_R c + b\left(1-E_R\right)\right]\left(1-\varepsilon\right)+\varepsilon\right\}\left(\varphi_{c2opt}^m - 1\right)/\eta_{c2} \end{aligned} \tag{15}$$

4. Numerical examples

In the calculations, it is set that $\eta_{c1} = \eta_{c2} = 0.9$, $\eta_{t1} = \eta_{t2} = 0.85$, $T_1 = 288.15K$, $P_1 = 0.1013MPa$, $P_9 = 0.104MPa$, $D_i = 0.98$ ($i = 1,2,3,4$), $\varepsilon = 0.9$ and $E_R = 0.9$. To see the effects of various parameters on exergy efficiency and other performances of the combined cycle, the results are presented graphically.

Figure 3 shows the influences of the effectiveness (E_R) of the regenerator on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss}/(C_P T_1))_{\min} - \varphi_{c1}$ characteristics, respectively. It shows that the optimal exergy efficiency $(\eta_E)_{opt}$ increases with the increase in E_R. The minimum exergy loss $(e_{loss}/(C_P T_1))_{\min}$ decreases with increase in E_R. It reveals that the base cycle with a regenerator can obtain better exergy performance.

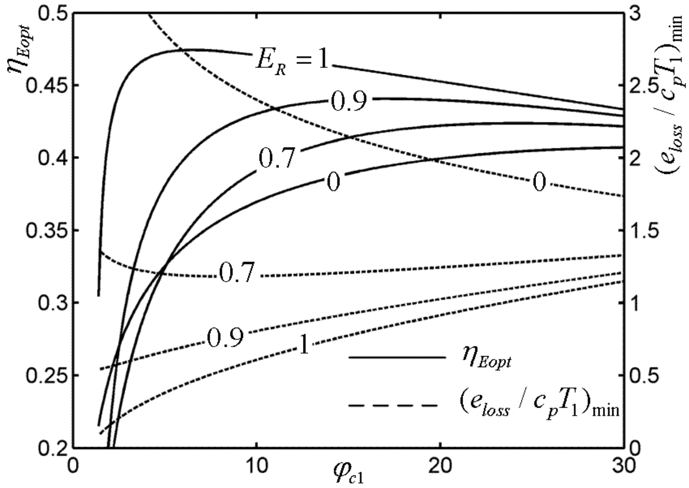


Figure 3. The influence of E_R on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss}/(C_PT_1))_{min} - \varphi_{c1}$ characteristics

Figures 4-7 show the influences of the temperature ratio (τ_1) of the Brayton cycle, the effectiveness (ε) of the heat exchanger, the total pressure-recovery coefficient (D), the compressor efficiencies (η_{c1} and η_{c2}), as well as the turbine efficiencies (η_{t1} and η_{t2}) on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss}/(C_PT_1))_{min} - \varphi_{c1}$ characteristics, respectively. They show that the optimal exergy efficiency $(\eta_E)_{opt}$ increases with the increases in τ_1, ε, D, η_{c1}, η_{c2}, η_{t1} and η_{t2}. The minimum exergy loss $(e_{loss}/(C_PT_1))_{min}$ decreases with increases in ε, D, η_{c1}, η_{c2}, η_{t1} and η_{t2} while increases with increase in τ_1 at low pressure ratio (φ_{c1}) of the compressor 1 and decreases with increase in τ_1 at high pressure ratio (φ_{c1}) of the compressor 1.

Figures 8-12 show the influences of the effectiveness (E_R) of the regenerator, the temperature ratio (τ_1) of the Brayton cycle, the effectiveness (ε) of the heat exchanger, the total pressure-recovery coefficient (D), the compressor efficiencies (η_{c1} and η_{c2}), as well as the turbine efficiencies (η_{t1} and η_{t2}) on the $\varphi_{c2opt} - \varphi_{c1}$ characteristic, respectively. They show that the optimal pressure ratio (φ_{c2opt}) of the compressor 2 increases with the increases in τ_1, ε, η_{c2}, η_{t2} and decreases in E_R, D, η_{c1}, and η_{t1}. They also show that the optimal pressure ratio of compressor 2 will equal to 1 when the effectiveness E_R of the regenerator is big enough or the efficiency η_{c2} of the compressor 2 is small enough. In other words, the compressor 2 should be canceled in these critical conditions.

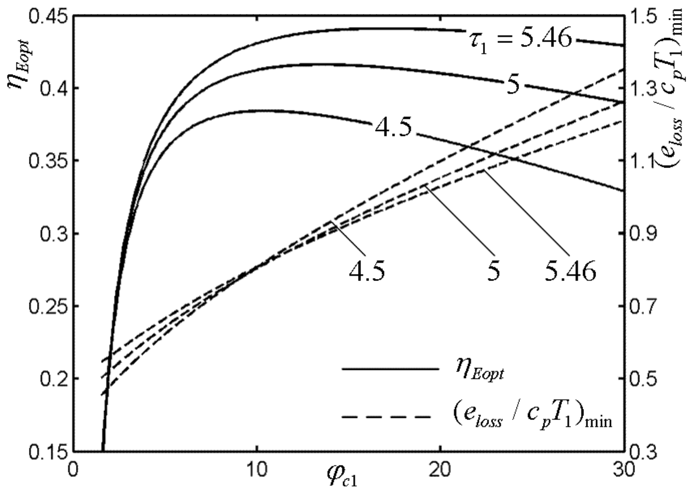


Figure 4. The influence of τ_1 on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss}/(C_PT_1))_{min} - \varphi_{c1}$ characteristics

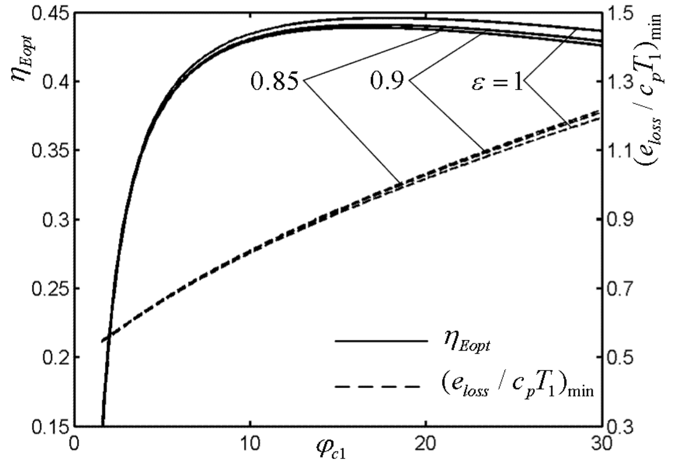


Figure 5. The influence of ε on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss} / (C_P T_1))_{\min} - \varphi_{c1}$ characteristics

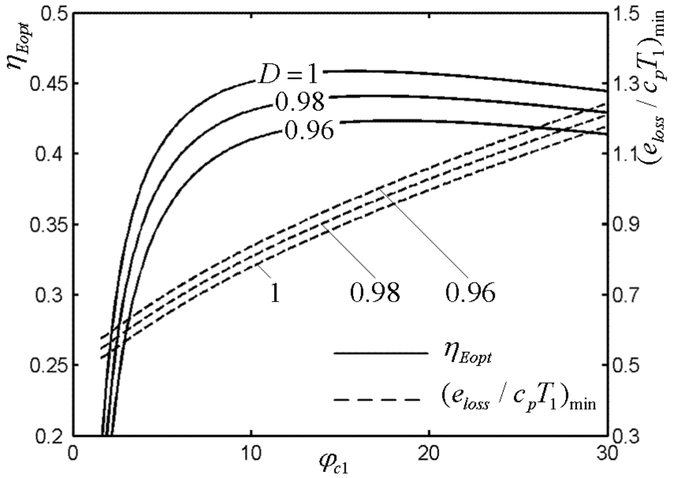


Figure 6. The influence of D on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss} / (C_P T_1))_{\min} - \varphi_{c1}$ characteristics

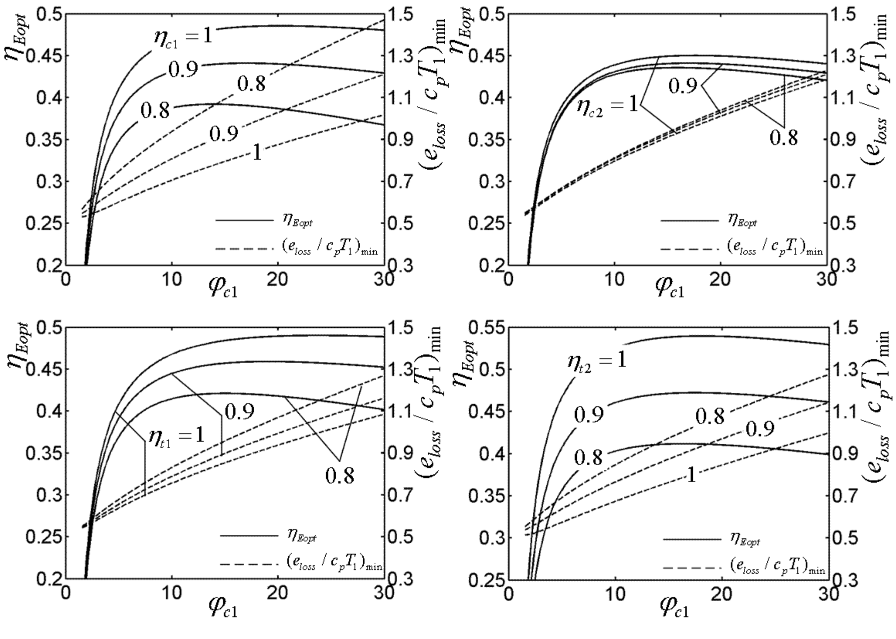
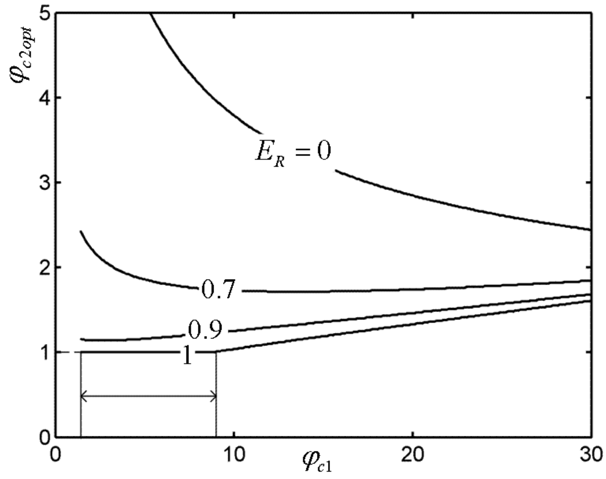


Figure 7. The influence of η_{c2}, η_{c1}, η_{t1} and η_{t2} on the $(\eta_E)_{opt} - \varphi_{c1}$ and $(e_{loss} / (C_P T_1))_{\min} - \varphi_{c1}$ characteristics



Figures 8. The influence of E_R on the $\varphi_{c2opt} - \varphi_{c1}$ characteristic

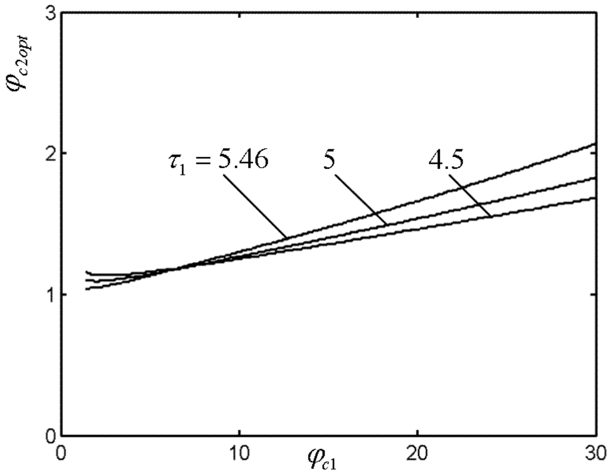


Figure 9. The influence of τ_1 on the $\varphi_{c2opt} - \varphi_{c1}$ characteristic

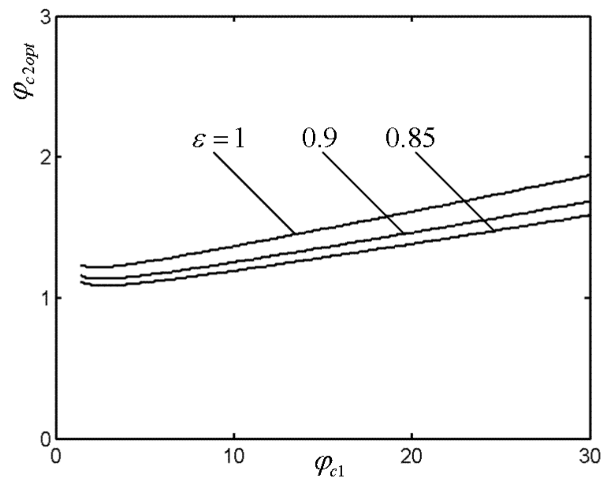


Figure 10. The influence of ε on the $\varphi_{c2opt} - \varphi_{c1}$ characteristic

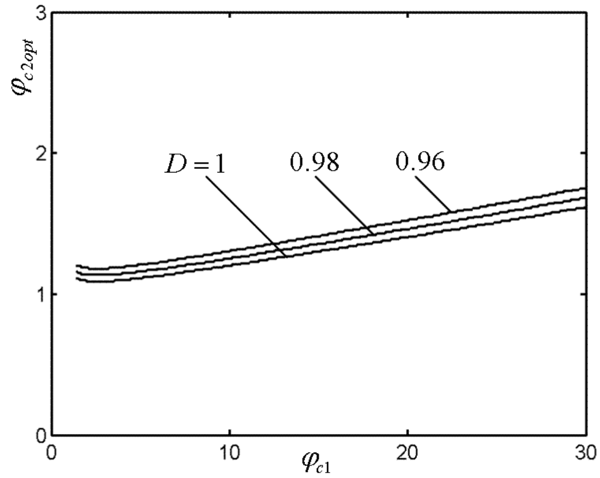


Figure 11. The influence of D on the $\varphi_{c2opt}-\varphi_{c1}$ characteristic

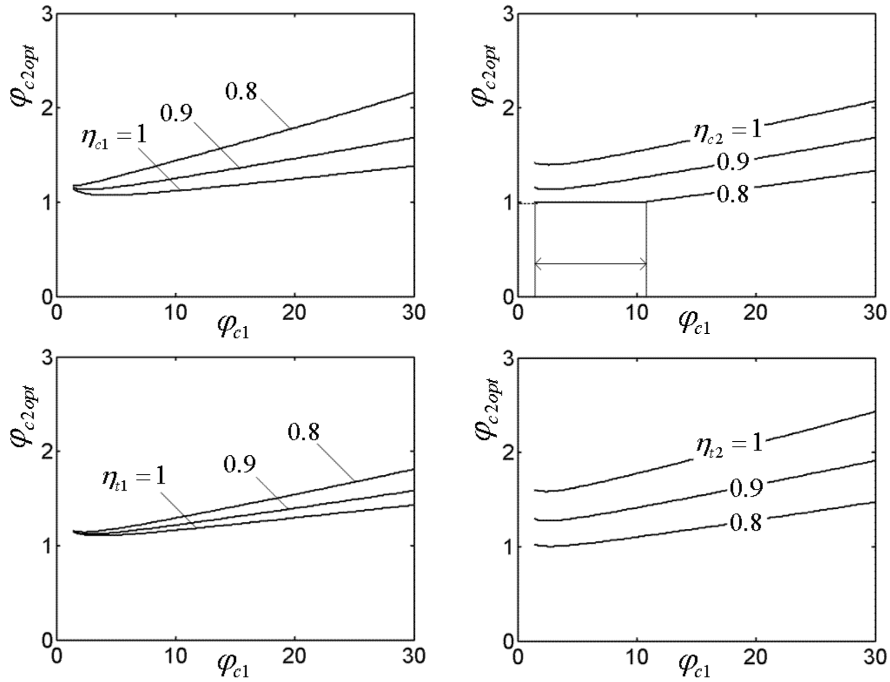


Figure 12. The influences of η_{c1}, η_{c2}, η_{t1} and η_{t2} on the $\varphi_{c2opt}-\varphi_{c1}$ characteristic

Figures 13-21 show the influences of the pressure ratio (φ_{c1}) of the compressor 1, the effectiveness (E_R) of the regenerator, the temperature ratio (τ_1) of the Brayton cycle, the effectiveness (ε) of the heat exchanger, the total pressure-recovery coefficient (D), the compressor efficiencies (η_{c1} and η_{c2}), as well as the turbine efficiencies (η_{t1} and η_{t2}) on the component irreversibilities for the combined cycle, respectively. They show that the exergy loss of the combustion is the largest, and followed by the exergy loss of the heat exchanger.

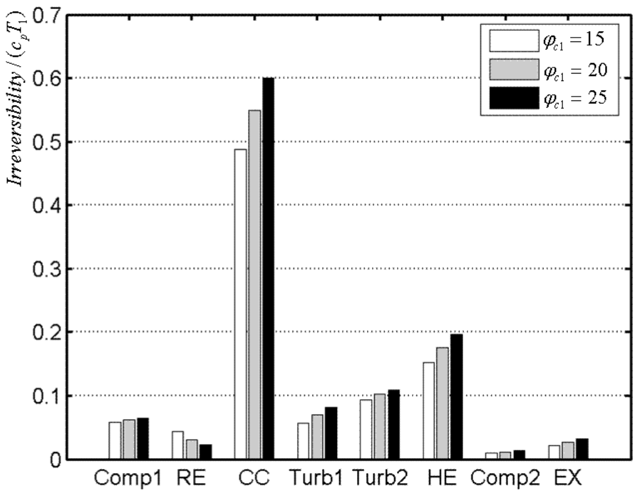


Figure 13. The influence of φ_{c1} on the component irreversibility for the combined cycle

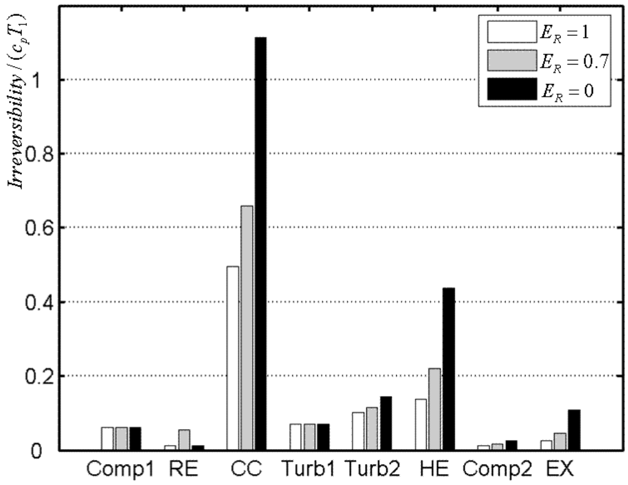


Figure 14. The influence of E_R on the component irreversibility for the combined cycle

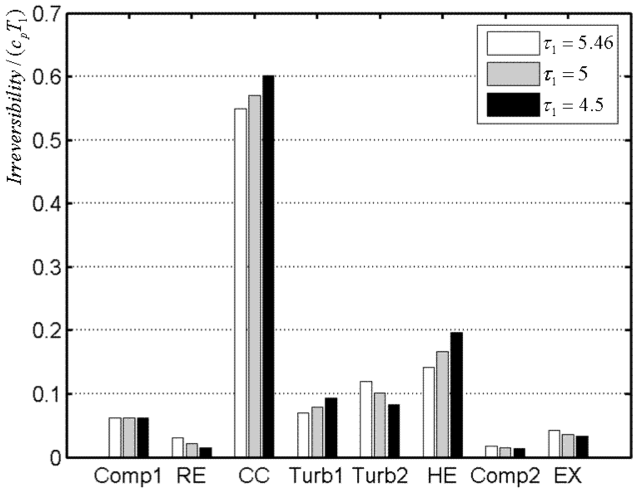


Figure 15. The influence of τ_1 on the component irreversibility for the combined cycle

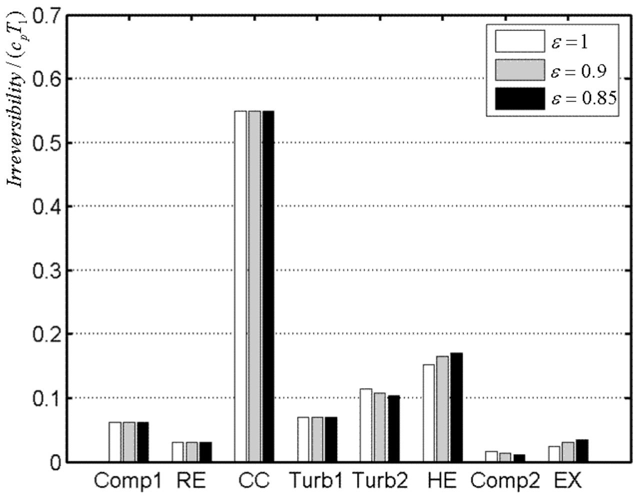


Figure 16. The influence of ε on the component irreversibility for the combined cycle

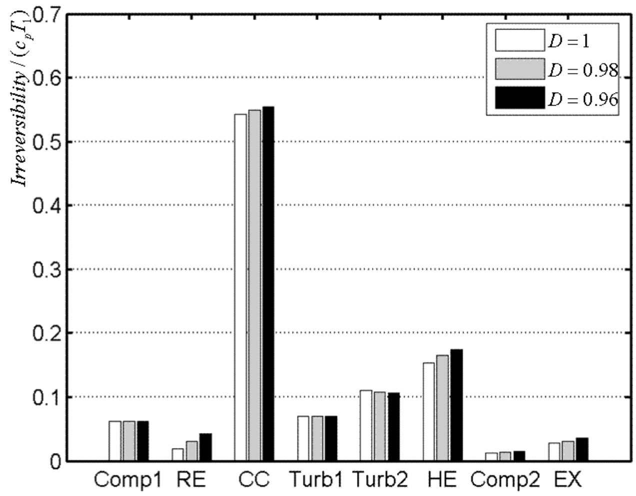


Figure 17. The influence of D on the component irreversibility for the combined cycle

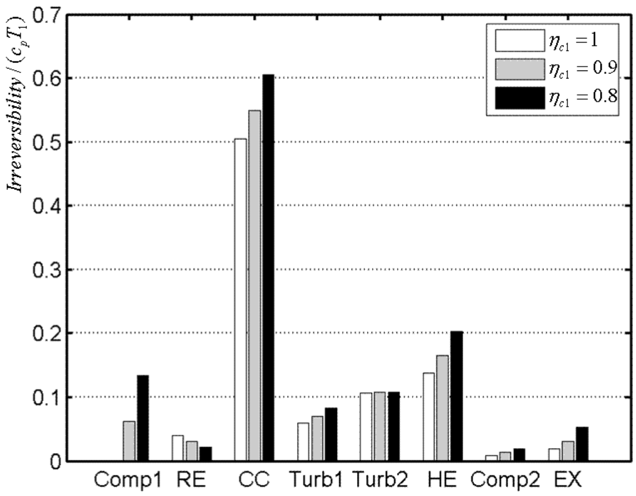


Figure 18. The influence of η_{c1} on the component irreversibility for the combined cycle

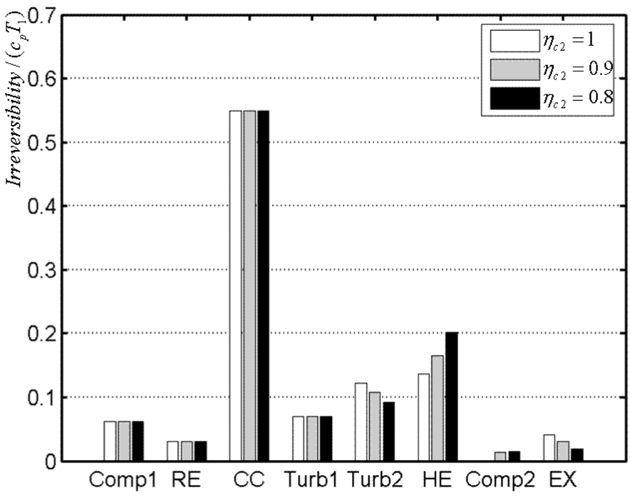


Figure 19. The influence of η_{c2} on the component irreversibility for the combined cycle

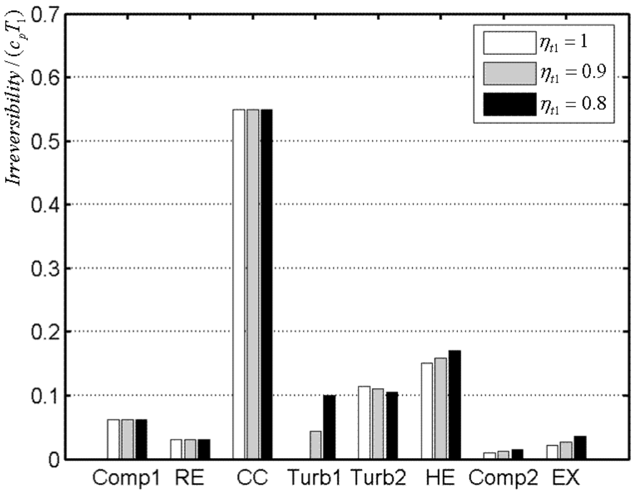


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Figure 20. The influence of η_{t1} on the component irreversibility for the combined cycle

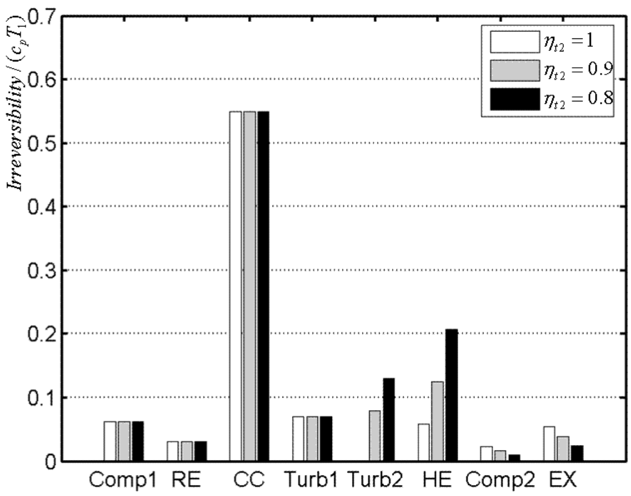

Figure 21. The influence of η_{t2} on the component irreversibility for the combined cycle

5. Conclusion

An exergy analysis of the combined regenerative Brayton and inverse Brayton cycles proposed in Ref. [10] has been performed in this paper. The effects of the effectiveness of the regenerator and other parameters on the exergy performances of the combined cycle are analyzed, and the exergy performances are optimized. One can see that the base cycle with regenerator can obtain better exergy performance than that of the base cycle. It presents facilitates the design and optimization of complex cycles by pinpointing the exergy losses. The exergy loss of combustion chamber is the largest in the combined cycle and followed by heat exchanger.

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Nomenclature

c	specific heat (kJ/(kgK))
CC	the combustion chamber
$Comp\,1$	the compressor of the regenerative Brayton cycle
$Comp\,2$	the compressor of the inverse Brayton cycle
e	exergy/exergy loss (kJ/kg)
E	the effectiveness of the regenerator/exergy
EX	the exhaust gas of the inverse Brayton cycle
h	enthalpy (kJ/kg)
HE	the heat exchanger of the inverse Brayton cycle
$Irreversibility$	the irreversibility of the component of the combined cycle
k	ratio of the specific heats
P	pressure(MPa)
q	heat (kJ/kg)
T	temperature (K)
$Turb\,1$	the turbine of the regenerative Brayton cycle
$Turb\,2$	the turbine of the inverse Brayton cycle
w	specific work output (kJ/kg)

Greek symbols

ε	the effectiveness of the heat exchanger
φ	pressure ratio
η	efficiency
τ	temperature ratio

Subscripts

b	burning
c	compressor
D	loss
E	exergy
ex	exhaust
f	fuel
g	generator
HE	heat exchanical
in	input
$loss$	total exergy loss
$\min$	mechanical
opt	optimal
out	output
p	pressure
r	rejected
R	regenerator

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Q	heat
t	turbine
V	volume
W	work
0	ambient
$1-10$	state points/sequence number

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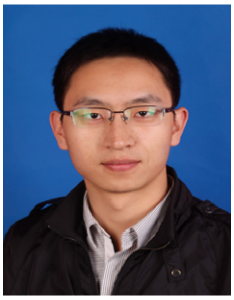

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