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New insight into regenerated air heat pump cycle

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ABSTRACT

Regenerated air (reverse Brayton) cycle has unique potentials in heat pump applications compared to conventional vapor-compression cycles. To better understand the regenerated air heat pump cycle characteristics, a thermodynamic model with new equivalent parameters was developed in this paper. Equivalent temperature ratio and equivalent isentropic efficiency of expander were introduced to represent the effect of regenerator, which made the regenerated air cycle in the same mathematical expressions as the basic air cycle and created an easy way to prove some important features that regenerated air cycle inherits from the basic one. Moreover, we proved in theory that the regenerator does not always improve the air cycle efficiency. Larger temperature ratio and lower effectiveness of regenerator could make the regenerated air cycle even worse than the basic air cycle. Lastly, we found that only under certain conditions the cycle could get remarkable benefits from a well-sized regenerator. These results would enable further study of the regenerated air cycle from a different perspective.

1. Introduction

Given the increasing environmental concerns, air is absolutely natural, safe and free working fluid for refrigeration or heat pump applications. Therefore, the reverse Brayton cycle using air as the refrigerant has been investigated for a long time. The air cycle machine was primarily served as the aircraft air conditioner [1,2] and was widely applied in cryogenic engineering [3–5]. In addition, the applicability of air cycle in transportation air conditioning [6–9], refrigerated storage [10–12], integrated air conditioning and desiccant system [13], domestic heating [14], heat pump clothes dryer [15], and heat pump water heater [16] was investigated as well. Nevertheless, the fairly low energy efficiency still constrains the air cycle from extensive applications in normal temperature range.

Limited by the commercial applications, more air cycle investigations were conducted with thermodynamic modeling not experiments. There were three main thermodynamic modeling approaches in the literature. One is the classical thermodynamics [17–21], the second is the finite-time thermodynamics [22–35], and the last is the recently developed entransy theory [36,37]. Although the assumptions or constraints taken in those investigations were more or less different, all studies reached consensus that air cycles should be thermodynamically optimized to get higher energy efficiency.

In the previous work [18], we found and proved that at optimal pressure ratio the basic air cycle can make the heating capacity in line with the heating load. It's an important feature for heat pump off-design operations, which vapor-compression heat pumps cannot do. Later on, we investigated the regenerated air cycle and did numerical simulation on the cycle performance at optimal pressure ratio [19]. Effected by the regenerator, the regenerated air cycle seemed more energy efficient than the basic one and could keep the optimal pressure ratio much more stable. Meanwhile, at the optimal pressure ratio the heating capacity is in line with the heating load. Therefore, the regenerated air cycle has higher potentials in heat pump applications. However, the existence of regenerator also introduced more complexity to the thermodynamic cycle model so that most conclusions were drawn by simulations not a solid theoretical proof.

In this paper, we develop a new theoretical approach to prove and reveal some important features of the regenerated air cycle. Two new parameters, so-called the equivalent temperature ratio and the equivalent expander efficiency, are introduced to represent the effect of regenerator so that the regenerated cycle can be in the same mathematical expressions as the basic one. Moreover, we can therefore have an in-depth look into what exactly the regenerator contributes to the cycle performance.

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Nomenclature

c_p	specific heat at constant pressure, J kg^{-1} K^{-1}
COP	coefficient of performance
h	enthalpy, J kg^{-1}
h_{fg}	latent heat of vaporization, J kg^{-1}
q_L	heat absorption per unit mass flow from low-temperature environment, J kg^{-1}
$q_{L,v}$	volumetric heat absorption, heat absorption per unit volume flow at compressor inlet, J m^{-3}
q_H	heating capacity per unit mass flow, J kg^{-1}
$q_{H,v}$	volumetric heating capacity, heating capacity per unit volume flow at compressor inlet, J m^{-3}
Q_H	heating capacity, W
p	pressure, Pa
p_r	pressure ratio
R	gas constant (= 8.314), J K^{-1} mol^{-1}
T	temperature,°C, K
$\dot{V}$	volume flow rate, m^3 s^{-1}
w_0	net power consumption per unit mass flow, J kg^{-1}
w_c	compressor power consumption per unit mass flow, J kg^{-1}
w_e	expander power recovery per unit mass flow, J kg^{-1}
X	intermediate variable defined in equation (58)
Y	intermediate variable defined in equation (59)

Greek symbols

ε_r	effectiveness of regenerator
η_c	isentropic efficiency of compressor
η_e	isentropic efficiency of expander
η'_e	equivalent isentropic efficiency of expander defined in equation (22)
π	derived pressure ratio defined in equation (17)
θ	temperature ratio defined in equation (18)
Θ	equivalent temperature ratio defined in equation (19)

Subscripts

in	inlet
H	high-pressure side; hot side; heating
L	low-pressure side; cold side
opt	optimal
out	outlet
v	volumetric

2. Thermodynamic model of regenerated air heat pump cycle

A typical regenerated air cycle heat pump is as shown in Fig. 1. Compared to the basic air cycle, a regenerator is added to transfer heat from hot air after the hot side heat exchanger to the compressor suction line. To reduce the heat transfer loss and eliminate the frosting/defrosting issue in heat pump applications, a semi-open air cycle without the cold side heat exchanger is preferred and investigated in this work.

Fig. 2 illustrates the thermodynamic processes which air undergoes in the regenerated air heat pump cycle. Curves 2–3 and 5–6 represent compression and expansion processes, respectively. Curve 3–4 demonstrates the isobaric heat rejection process in the hot side heat exchanger. Process 6-1 is open for the semi-open cycle. Curves 4–5 and 1–2 are the processes in hot and cold sides of the regenerator, respectively.

The assumptions and irreversibilities taken into account for model development of the regenerated cycle are as follows.

1) The air is ideal gas.
2) All processes in heat exchangers are isobaric.
3) Constant isentropic efficiencies are specified for compressor and expander.
4) The effectiveness of regenerator is given.
5) Temperature difference between the hot side inlet of regenerator and high-temperature heat sink (so-called exit temperature difference) is given.
6) Temperature difference between the cold side inlet of regenerator and low-temperature heat source is zero for the semi-open cycle.

Accordingly, we can develop the thermodynamic model hereinbelow.

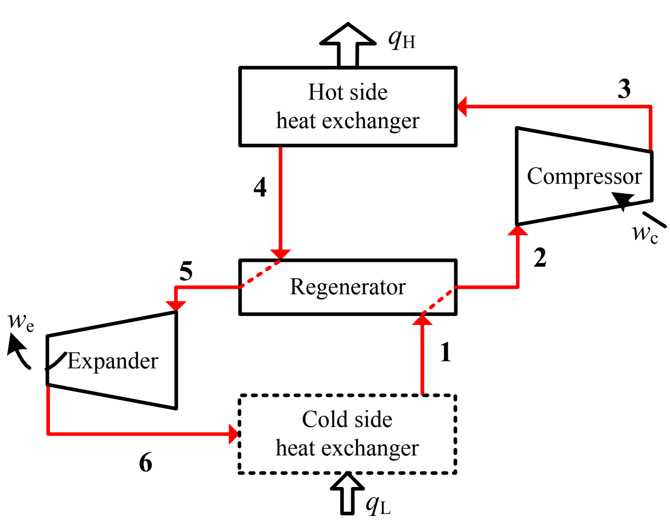

Fig. 1. Schematic of regenerated air cycle heat pump.

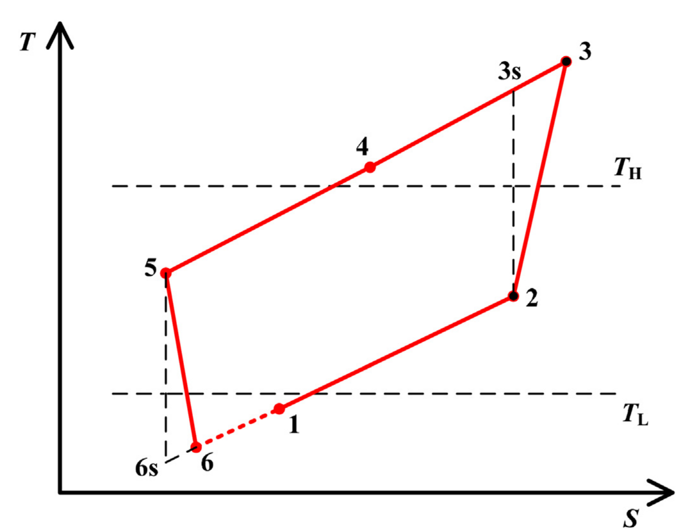

Fig. 2. T-s diagram of regenerated air cycle.