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A medium-temperature solar thermal power system and its efficiency optimisation

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Abstract

This paper firstly expounds that the reheat-regenerative Rankine power cycle is a suitable cycle for the parabolic trough collector, a popular kind of collector in the power industry. In a thermal power cycle, the higher the temperature at which heat is supplied, the higher the efficiency of the cycle. On the other hand, for a given kind of collector at the same exiting temperature, the higher the temperature of the fluid entering the collector, the lower the efficiency of the collector. With the same exiting temperature of the solar field and the same temperature differences at the hottest end of the superheater/reheater and at the pinch points in the heat exchangers (e.g., the boiler) in the cycle, the efficiencies of the system are subject to the temperature of the fluid entering the collector or the saturation temperature at the boiler. This paper also investigates the optimal thermal and exergetic efficiencies for the combined system of the power cycle and collector. To make most advantage of the collector, the exiting fluid is supposed to be at the maximum temperature the collector can harvest. Hence, the thermal and exergetic efficiencies of the system are related to the saturation temperature at the boiler here. © 2002 Elsevier Science Ltd. All rights reserved.

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1. Introduction

Energy and environment are two of the most concerning issues in the current world. Today most electricity is generated by consuming fossil fuels such as coal, oil and natural gas. Not

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only do fossil fuels have a limited life but also their combustion emissions have serious negative impacts on our environment such as adding to the greenhouse effect and causing acid rain. Reducing the amount of the fossil fuels consumed while supplying the sufficient power to meet the increasing demand is a tough task. As a clean, free, and non-depleting source, solar energy utilisations (e.g., solar power generation) are getting more and more attention [1–3]. The authors have proposed a solar aided regenerative Rankine power system for the places where there is a conventional regenerative Rankine power plant [4], and a modified regenerative-reheat Rankine power cycle for using low-temperature solar energy and other low-temperature heat sources as the main heat source [5]. However, the studies are mostly on the cycles. In terms of solar thermal power system, not only the performance of the power cycles, but also that of the collectors, and thus that of the whole system, should be evaluated. This paper investigated the optimal efficiency for a combined system of a regenerative-reheat Rankine power cycle and the parabolic trough collector, a popular kind of collector in the power industry.

2. The model of the solar thermal power system

The solar thermal electric technologies usually concentrate large amounts of sunlight onto a small area to permit the buildup of relatively high-temperature heat energy which can be converted into electricity in a conventional heat engine [1]. Parabolic trough collectors are the most developed and deployed type of solar concentrators [2]. The temperature of the heat carrier is usually less than 400 °C for parabolic trough collectors [3]. There have been some practical examples using parabolic trough collectors and the reheat Rankine cycle [1] for power generating. Therefore, the parabolic trough collector is selected as the collector in the analysis, and the upper temperature limit of the solar field is the temperature the fluid can achieve in the parabolic trough collector. To investigate the effects of the heating temperature on the efficiency of the system, both the temperature that the process heat is supplied in the power cycle and the temperature of the heat carrier in the collector should be changeable. The regenerative Rankine cycle is suitable here (the temperature of the water fed to the boiler can be changed by bleeding the steam at different pressure). The temperature at which the process heat is supplied in the power cycle can be changed by changing the boiling pressure/temperature. With the parabolic trough collector to collect and supply the heat energy, water/steam can be, and here is, employed as the working fluid in the power cycle. Owing to the change of the boiling pressure, the vapour exhausted from the turbine in the regenerative Rankine cycle at the condensing pressure may be in the wet saturated, dry saturated, or superheated states. In order to extend the heating temperature range while maintaining the quality of the wet vapour at no less than 0.9, and improve the efficiency of the cycle furthermore, the reheat model is needed. Therefore, the power cycle used here is the reheat-regenerative Rankine cycle. A simple reheat-regenerative arrangement is schematically shown in Fig. 1. Some typical properties used in the following illustration are also shown in it. The *T–s* diagram of the arrangement for the case that the reheat pressure is greater than that of the regenerative is schematically shown in Fig. 2.

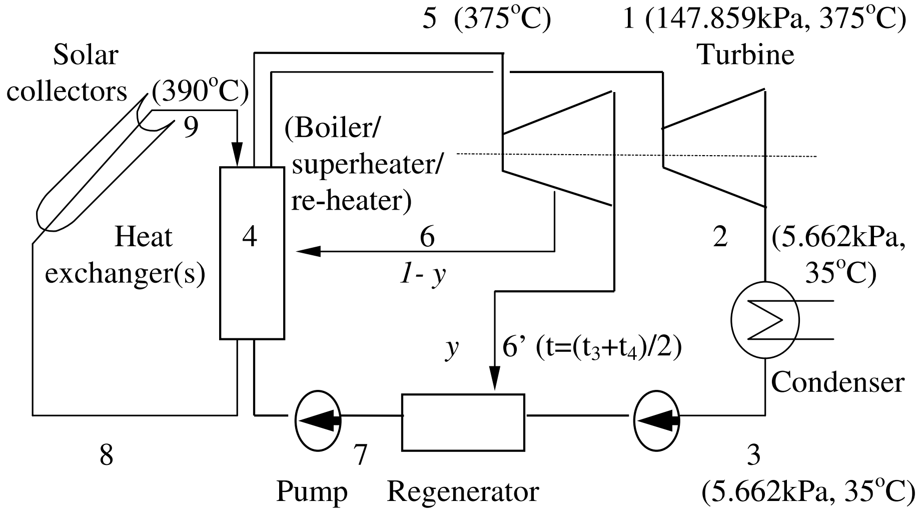

Fig. 1. The system of one-stage reheat-regenerative power cycle and the parabolic trough collector.

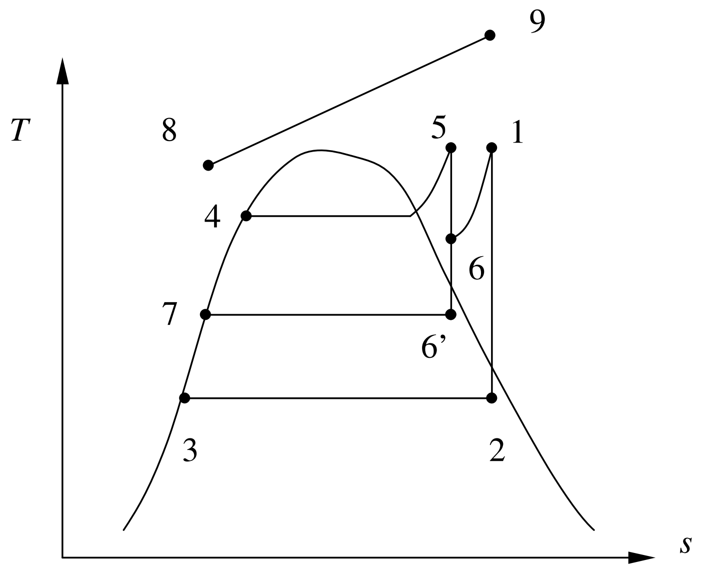

Fig. 2. The power cycles expressed in the T–s diagram.

3. The analysis of the solar power system

3.1. The power cycle

The fraction of the total flow used for regeneration

$$y = \frac{h_7 - h_3}{h_{6'} - h_3} \tag{1}$$

The turbine work output (neglecting the pump work)

$$W = h_5 - h_6 + y(h_6 - h_{6'}) + (1 - y)(h_1 - h_2) \tag{2}$$

The heat added

$$Q = h_5 - h_7 + (1 - y)(h_1 - h_6) \tag{3}$$

The thermal (first law) efficiency

$$\eta_{\mathrm{I,cycle}} = \frac{W}{Q} \tag{4}$$

The exergy (availability [6]) increase of the working fluid in the heating process in the cycle

$$\mathrm{Ex} = h_5 - h_7 - T_0(s_5 - s_7) + (1 - y)[h_1 - h_6 - T_0(s_1 - s_6)] \tag{5}$$

where T_0 is the ambient temperature.

The exergetic (second law) efficiency of the cycle

$$\eta_{\mathrm{II,cycle}} = \frac{W}{\mathrm{Ex}} \tag{6a}$$

The exergy transfer accompanying the heat of the solar heat carrier [6]

$$\mathrm{Ex}_Q = Q\left\{1 - \frac{T_0}{T_{\mathrm{h1}} - T_{\mathrm{h2}}} \ln\left[\frac{T_{\mathrm{h1}}}{T_{\mathrm{h2}}}\right]\right\} \tag{7}$$

where T_{h1}, T_{h2} is the temperature of the solar heat carrier entering and exiting the heat exchanger(s), respectively.

The exergetic (second law) efficiency of the cycle (including the exergy loss in the heating process)

$$\eta'_{\mathrm{II,cycle}} = \frac{W}{\mathrm{Ex}_{,Q}} = \frac{\eta_{\mathrm{I,cycle}}}{1 - \frac{T_0}{T_{\mathrm{h1}} - T_{\mathrm{h2}}} \ln \frac{T_{\mathrm{h1}}}{T_{\mathrm{h2}}}} \tag{6b}$$

3.2. The collector

The thermal efficiency of solar collectors [3]

$$\eta_{\mathrm{I,collector}} = \frac{Q}{{}^S E_b} = (\alpha\tau)F + (\alpha\varepsilon)F\frac{\sigma T_{\mathrm{c}}^4}{C\,{}^S E_b} - (\varepsilon\bar{\rho})F\frac{\sigma T^4}{C\,{}^S E_b} - U_{\mathrm{L}}F\frac{T - T_{\mathrm{a}}}{C\,{}^S E_b} \tag{8}$$

where $\alpha\tau = \alpha_{\mathrm{a}}\tau_{\mathrm{c}}/1 - \rho_{\mathrm{c}}(1 - \alpha_{\mathrm{a}})$ and α_{a} is the absorptivity of the absorber, τ_{c} is the transmissivity of the cover, and ρ_{c} is the fraction backscattered by the cover, $\alpha\varepsilon = \alpha_{\mathrm{a}}\varepsilon_{\mathrm{c}}/1 - \rho_{\mathrm{c}}(1 - \alpha_{\mathrm{a}})$ and ε_{c} is the emissivity of the cover, $\varepsilon\bar{\rho} = \varepsilon_{\mathrm{a}}(1 - \rho_{\mathrm{c}})/1 - \rho_{\mathrm{c}}(1 - \alpha_{\mathrm{a}})$ and ε_{a} is the emissivity of the absorber, σ is the Stefan–Boltzmann constant, 5.67×10^{-8} W/m^2 K^4, $\bar{\rho}$ is the average reflectivity, U_{L} is the heat loss coefficient, F is termed as fin factor which is close to 1 for a well designed receiver or collector, T is the fluid temperature, T_{c} is the cover temperature, T_{a} is the ambient temperature, C is the concentration ratio, ${}^S E_b$ is the direct radiation, ${}^S E_b = {}^S E - {}^S E_{\mathrm{d}}$ where ${}^S E_{\mathrm{d}}$ is the diffusion radiation and ${}^S E$ is the global irradiance, ${}^S E = f\sigma T_{\mathrm{s}}^4$ where f is the dilution factor.

The exergy released by the solar irradiance [3]

$$\mathrm{Ex}_{\mathrm{solar}} \approx {}^{S}E\left(1 - \frac{4}{3}\frac{T_{\mathrm{a}}}{T_{\mathrm{s}}}(1 - 0.28\ln f)\right) \tag{9}$$

The exergetic (second law) efficiency of the collector

$$\eta_{\mathrm{II,collector}} = \frac{\mathrm{Ex}_{,Q}}{\mathrm{Ex}_{,\mathrm{solar}}} \approx \eta_{\mathrm{I,collector}} \frac{1 - \frac{T_0}{T_{\mathrm{h1}} - T_{\mathrm{h2}}} \ln\left[\frac{T_{\mathrm{h1}}}{T_{\mathrm{h2}}}\right]}{1 - \frac{4}{3}\frac{T_{\mathrm{a}}}{T_{\mathrm{s}}}(1 - 0.28\ln f)} \tag{10}$$

3.3. The combined system of the collector and the power cycle

For the combined system of the collector and the power cycle, the thermal efficiency and the exergetic efficiency are evaluated by the following expressions.

The thermal efficiency

$$\eta_{\mathrm{I}} = \eta_{\mathrm{I,collector}}\eta_{\mathrm{I,cycle}} \tag{11}$$

The exergetic efficiency

$$\eta_{\mathrm{II}} = \eta_{\mathrm{II,collector}}\eta'_{\mathrm{II,cycle}} \tag{12}$$

4. Example

To illustrate the principle of the approach, let us investigate the following case. Assuming that the temperature of the heat source (outlet temperature of the solar field) is 390 °C. Water is chosen as the working fluid in the power cycle. The heat transfer temperature difference at the hottest end of the superheater/reheater is 15 °C and is 10 °C at the pinch points in the heat exchangers (e.g., the boiler), and the saturated temperature of the water in the condenser is 35 °C. The steam is heated to the same temperature (375 °C) in the superheater and the reheater to take utmost use of the heat source. The system with a simple reheat-regenerative arrangement schematically shown in Fig. 1 is considered. The vapour exhausted from the second-stage turbine is assumed to be at the dry saturated state. The main parameters are listed in Table 1. The ambient temperature for exergy evaluation is assumed to be 298.15 K and neglecting the loss in the turbines (both heat and work) and the losses of heat in the boiler and other heat exchangers. The results of the analysis are shown in Figs. 3 and 4 (the bled temperature is taken as $(t_3 + t_4)/2$, $(\alpha\tau)F = 0.8$, $(\varepsilon\rho)F = 0.8$, $(\alpha\varepsilon)F = 0.8$, $U_{\mathrm{L}} = 20\ \mathrm{W\,m^{-2}\,K^{-1}}$, $T_{\mathrm{a}} = 298.15$ K, $T_{\mathrm{c}} = 300$ K, the fluid temperature in the collector T is taken as the average temperature of the heat carrier, $C = 40$, standard spectrum with $f = 1.3 \times 10^{-5}$, $T_{\mathrm{s}} = 5777$ K, and ${}^{S}E_b \approx 821\ \mathrm{W\,m^{-2}}$).

The analyses show that when raising the saturation temperature in the boiler, i.e., when raising the temperature of the heat carrier entering the collector while keeping the temperature exiting the collector constant, the thermal efficiency of the cycle increases, but that of the collector, decreases. The efficiency of the combined system grows and then drops down, and there is an optimum

Table 1
The key parameters of the example system

Point	P (kPa, absolute)	t (°C)	h (kJ/kg)	s (kJ/kg K)
1. The steam entering the second-stage turbine	147.8588	375	3226.134	8.3543
2. The stream leaving the second-stage turbine	5.622	35	2565.4	8.3543
3. The liquid leaving the condenser (saturated)	5.622	35	146.6	0.5049
4. The saturated liquid in the boiler	(changeable)	(changeable)		
5. The steam entering the first-stage turbine		375		
6. The stream leaving the first-stage turbine to the superheater	147.8588			
6′. The stream leaving the first-stage turbine to the regenerator		$(t_3 + t_4)/2$		
7. The liquid leaving the regenerator (saturated)	$(\approx P_{6'})$			
8. The fluid entering the solar collector		$t_4 + 10 - (h_4 - h_7)/(mc)_h$		
9. The fluid leaving the solar field		390		

Note: $(mc)_h$ is the heat capacity of the solar heat carrier corresponding to 1 kg water circulated in the boiler.

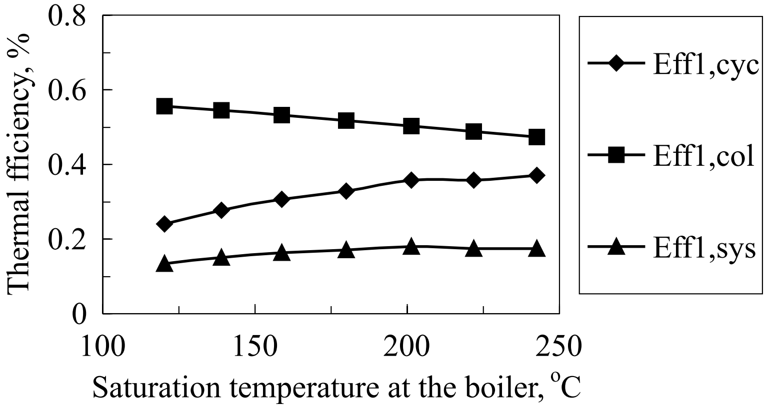

Fig. 3. The thermal efficiencies vs. the saturation temperature in the boiler (Eff1,cyc, Eff1,col, and Eff1,sys are the thermal efficiencies of the cycle, the collector, and the combined system, respectively).

efficiency for the combined system. In the case we discussed, the optimum efficiency, 17.9%, is reached when the saturation temperature in the boiler is about 201 °C (Fig. 3).

The exergetic efficiency of the cycle itself and the cycle including the exergy loss in the heating process also reaches the highest value of 95.17% and 76.94% respectively when the saturation temperature in the boiler is at about 201 °C. The exergetic efficiency of the collector varies a little from 159 to 201 °C of the saturation temperature in the boiler, and reaches the maximum value of 32.73% at about 180 °C. Owing to the low-exergetic efficiency of the collector, the exergetic efficiency of the combined system is not high and the maximum value is 25.12% when the saturation temperature in the boiler is about 201 °C again (Fig. 4).

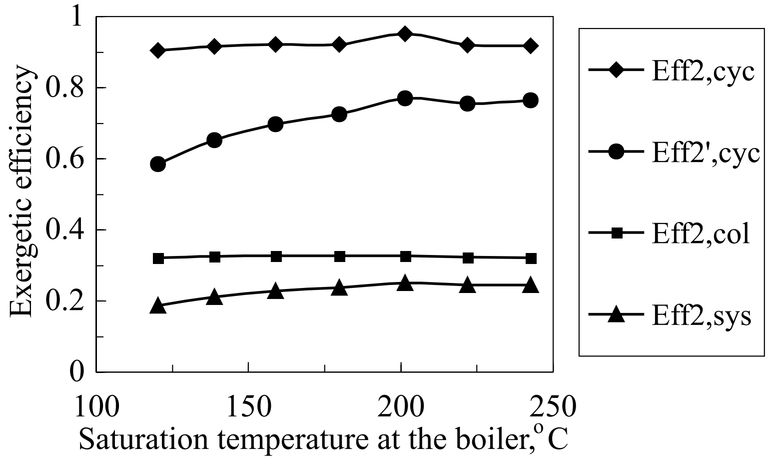

Fig. 4. The exergetic efficiencies vs. the saturation temperature in the boiler (Eff2,cyc, Eff2,col, and Eff2,sys are the exergetic efficiencies of the cycle, the collector, and the combined system, respectively).

From the analyses it can be seen that there is an optimum saturation temperature in the boiler, i.e., an optimum temperature of the fluid entering the collector (with the same exiting temperature) at which both the thermal efficiency and the exergetic efficiency of the combined system of the reheat-regenerative cycle and the solar collector reaches the maximum values simultaneously. Hence it is also important to optimise the temperature of the solar collecting or the saturation temperature in the boiler to obtain the high efficiencies.

It should also be pointed out that the reheat-regenerative arrangement presented here is just a simple pattern which is used here to illustrate the principle of the investigation of the optimum efficiencies for such systems. If the number of the stages of reheat regenerative and the operating parameters are optimised to a particular case, high efficiencies can be expected.

5. Conclusions

The analyses show that the reheat-regenerative arrangement is suitable for the medium-temperature solar thermal power generation purpose. There is an optimum saturation temperature in the boiler or an optimum temperature the fluid entering the solar field (with the same exiting temperature) at which both the thermal efficiency and the exergetic efficiency of the combined system of the reheat-regenerative cycle and the solar collector hits the maximum values simultaneously. In the case we discussed, the optimum saturation temperature in the boiler is about 201 °C, and the thermal efficiency and the exergetic efficiency of the system is 17.9% and 25.12%, respectively. Hence it is also important to optimise the temperature of the solar collecting or the saturation temperature in the boiler to obtain the high efficiencies. If the number of the stages of reheat regenerative and the operating parameters are optimised to a particular case, high efficiencies can be expected.

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