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Optimum performance of the small-scale open and direct solar thermal Brayton cycle at various environmental conditions and constraints☆

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ABSTRACT

The Brayton cycle's heat source can be obtained from solar energy instead of the combustion of fuel. The irreversibilities of the open and direct solar thermal Brayton cycle with recuperator are mainly due to heat transfer across a finite temperature difference and fluid friction, which limit the net power output of such a system. In this work, the method of total entropy generation minimisation is applied to optimise the geometries of the receiver and recuperator at various steady-state weather conditions. For each steady-state weather condition, the optimum turbine operating point is also found. The authors specifically investigate the effect of wind and solar irradiance on the maximum net power output of the system. The effects of other conditions and constraints, on the maximum net power output, are also investigated. These include concentrator error, concentrator reflectivity and maximum allowable surface temperature of the receiver. Results show how changed solar beam irradiance and wind speed affect the system net power output and optimum operating point of the micro-turbine. A dish concentrator with fixed focal length, an off-the-shelf micro-turbine and a modified cavity receiver is considered.

1. Introduction

The solar thermal Brayton cycle uses the concentrated power of the sun as a heat source to generate mechanical power. Low operation and maintenance costs make the small-scale open and direct solar thermal Brayton cycle with recuperator attractive for power generation. The recuperator can increase the efficiency of the Brayton cycle and it allows the compressor to operate at lower pressure ratios. The Brayton cycle is definitely worth studying when comparing its efficiency [1] and cost [2] with those of other power cycles. A black solar receiver, mounted at the focus of a parabolic dish concentrator can be sized such that it absorbs the maximum amount of heat [3]. Sendhil Kumar and Reddy [4] compared different types of cavity receivers numerically and suggested that the modified cavity receiver may be preferred in a solar dish collector system. The total heat loss rate from the modified cavity receiver due to convection, radiation and conduction, is a function of the receiver geometry [5]. A numerical investigation of natural convection heat loss [6], an inclusion of the contribution of radiation losses [7] and an improved model for natural convection heat loss [8] was presented for the modified cavity receiver.

The irreversibilities of a small-scale solar thermal Brayton cycle with recuperator limit the net power output of such a system. These irreversibilities are mainly due to heat transfer across a finite temperature difference and fluid friction. To obtain the maximum net power output of a solar thermal Brayton cycle, a combined effort of heat transfer, fluid mechanics and thermodynamic thought is necessary. The method of total entropy generation minimisation combines these thoughts [9].

Optimisation using the second law of thermodynamics is commonly found in recent work. A second law analysis to study the effect of operating parameters on the optimum pressure ratio and component irreversibilities of the supercritical CO_2 recompression Brayton cycle [10], as well as an optimisation [11] have been performed. The optimal performance parameters for the maximum exergy delivery during the collection of solar energy in a flat-plate solar air heater were established by optimising the geometries of the plate [12]. Exergy analysis has also been applied in various power studies [13].

Various authors have emphasised the importance of the optimisation of the global performance of a system, by minimising the total irreversibility rate from all the different components or processes of such a system by sizing the components accordingly [14–19]. In recent work, a geometry optimisation method based on total entropy generation minimisation was developed and was applied to establish the maximum net power output of a small-scale open and direct solar thermal Brayton cycle with cavity

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receiver and recuperator at any steady-state condition and various micro-turbine operating points [20]. This was done for various concentrator diameters and micro-turbines. This method allows for the global performance of the system to be optimised, by minimising the total irreversibility rate by sizing the receiver and recuperator accordingly. This optimisation was done for multiple steady-state systems with no wind and a constant solar irradiance of 1000 W/m^2.

The effects of wind, receiver inclination, rim angle, atmospheric temperature and pressure, recuperator height, solar irradiance and concentration ratio on the optimum geometries and performance of the open and direct solar thermal Brayton cycle were also investigated [21]. For a specific weather condition, the geometries, operating conditions and irreversibilities of the optimised system were shown as a function of system mass flow rate. It was shown that for each specific environment and set of parameters an optimum receiver and recuperator geometry and system mass flow rate exist so that the system produces maximum net power output.

In this paper, the authors further investigate the effects that changed wind and solar irradiance have on the optimum turbine operating point of the micro-turbine. Other effects are also investigated, such as specular reflectivity and concentrator error.

2. Model

A micro-turbine from the Garrett range [22] and $D_{\text{conc}} = 5.2$ m is used in the analysis. The open and direct solar thermal Brayton cycle with recuperator is shown in Fig. 1. A parabolic dish supplies the solar heat for the cavity receiver.

2.1. The control volume

The rate of intercepted heat by the cavity receiver, $\dot{Q}*$, is a function of the cavity receiver geometry. For the analysis in this work, the apparent sun's temperature as an exergy source, T*, is assumed to be 2470 K [9] and at a point between the concentrator and receiver. $\dot{Q}*$ can be regarded as the intercepted power at the receiver, after the irreversibility rates due to scattering and the transformation of radiation have been deducted. $\dot{W}_{\text{net}}$ is the net power output of the system.

2.2. Solar receiver model

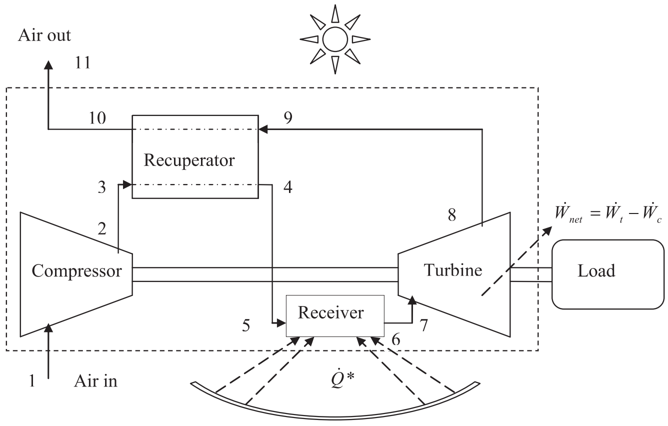

Fig. 1. The open and direct solar thermal Brayton cycle with recuperator.

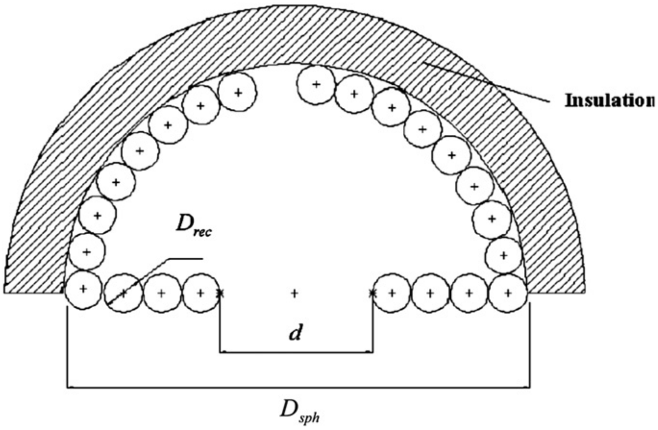

Fig. 2. Modified solar cavity receiver.

A section view of the modified cavity receiver suggested by Reddy and Sendhil Kumar [8] is shown in Fig. 2. The receiver inner surface is made up of a closely wound copper tube with diameter, D_{rec}, through which the working fluid travels. The receiver tube with length, L_{rec}, constructs the half spherical cavity receiver and its aperture. Note that the tube is concentrically wound. An area ratio of $A_w/A_a = 8$ is recommended [8] as it was found to be the ratio that gives the minimum heat loss rate from the cavity receiver. The diameter of the receiver can be calculated [20,21] as

$$D_{\text{sph}} = 2\sqrt{(A_w + A_a)/3\pi} \tag{1}$$

Due to the area ratio constraint, the receiver diameter is a function of the receiver aperture diameter,

$$D_{\text{sph}} = \sqrt{3}d \tag{2}$$

The receiver aperture diameter can be calculated using Eq. (3) since $A_w = D_{\text{rec}}L_{\text{rec}}$.

$$d = \sqrt{D_{\text{rec}}L_{\text{rec}}/2\pi} \tag{3}$$

For $A_w/A_a = 8$, the Nusselt number, $\text{Nu}_D = (h_{\text{nconv}}D_{\text{sph}})/k$, for natural convection heat loss rate based on receiver diameter for a 3-D receiver model can be calculated as a function of the inclination angle of the receiver [8],

$$\text{Nu}_D = 0.698\text{Gr}_D^{0.209}(1+\cos\beta)^{0.968}(T_w/T_0)^{-0.317}\left(d/D_{\text{sph}}\right)^{0.425} \tag{4}$$

For $A_w/A_a = 8$, the ratio of radiation heat loss to convection heat loss is a function of receiver inclination and varies between approximately 0.92 and 1.46 [7]. It is assumed that $\dot{Q}_{\text{loss,nrad}} = U\dot{Q}_{\text{loss,nconv}}$ for the modified cavity receiver, where U is a function of the inclination of the receiver and varies between 1.92 and 2.46. The rate of heat loss due to natural convection and radiation is therefore

$$\begin{aligned}\dot{Q}_{\text{loss,nrad}} &= 0.698\text{Gr}_D^{0.209}UC(1+\cos\beta)^{0.968}(T_w/T_0)^{-0.317} \\ &\quad\times\left(d/D_{\text{sph}}\right)^{0.425}\end{aligned} \tag{5}$$

where $C = (kA_a/D_{\text{sph}})(T_w - T_0)$.

With an assumed insulation thickness of $D/2$, the rate of heat loss due to conduction [5] is

$$\dot{Q}_{\text{loss,cond}} = (T_{s,in} - T_0)\Big/\left(1/2\pi Dk_{\text{ins}} + 1/2\pi h_{\text{conv}}D^2\right) \tag{6}$$

where h_{conv} is the external heat transfer coefficient on the insulation surface.