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# Operating conditions of an open and direct solar thermal Brayton cycle with optimised cavity receiver and recuperator

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The small-scale open and direct solar thermal Brayton cycle with recuperator has several advantages, including low cost, low operation and maintenance costs and it is highly recommended. The main disadvantages of this cycle are the pressure losses in the recuperator and receiver, turbomachine efficiencies and recuperator effectiveness, which limit the net power output of such a system. The irreversibilities of the solar thermal Brayton cycle are mainly due to heat transfer across a finite temperature difference and fluid friction. In this paper, thermodynamic optimisation is applied to concentrate on these disadvantages in order to optimise the receiver and recuperator and to maximise the net power output of the system at various steady-state conditions, limited to various constraints. 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 were investigated. The dynamic trajectory optimisation method was applied. Operating points of a standard micro-turbine operating at its highest compressor efficiency and a parabolic dish concentrator diameter of 16 m were considered. The optimum geometries, minimum irreversibility rates and maximum receiver surface temperatures of the optimised systems are shown. For an environment with specific conditions and constraints, there exists an optimum receiver and recuperator geometry so that the system produces maximum net power output.



## 1. Introduction

Concentrated solar power systems use the concentrated power of the sun as a heat source to generate mechanical power. The Brayton cycle is definitely worth studying when comparing its efficiency with that of other power cycles [1]. Emphasis may shortly shift to solarised Brayton micro-turbines from Dish-Stirling technology due to high Stirling engine costs [2]. When a recuperator is used, the Brayton cycle has very high efficiency at low pressure ratios. The main disadvantages of a solar thermal Brayton cycle are the pressure losses in the recuperator and receiver, turbomachine efficiencies and recuperator effectiveness [3], which limit the net power output of such a system. To obtain the maximum net power output, a combined effort of heat transfer, fluid mechanics and thermodynamic thought is required. The method of entropy generation minimisation combines these thoughts [4].

The irreversibilities of the recuperative solar thermal Brayton cycle are mainly due to heat transfer across a finite temperature difference and fluid friction. 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 the system by sizing the components accordingly [5–11]. The geometries of the receiver and recuperator can be optimised in such a way that the total entropy generation rate is minimised to allow maximum net power output at any steady-state condition.

Entropy generation minimisation has been used in various internal flow optimisation studies such as: the optimum tube diameter for a tube [5,7]; the optimal aspect ratio for single-phase, fully developed, laminar and turbulent flow with constant heat flux [12]; and the optimum channel geometries with constant wall temperature or constant heat flux [10,11]. Entropy generation and its minimisation have been expressed for numerous heat exchangers and heat transfer surfaces: counterflow and nearly-ideal heat exchanger neglecting fluid friction [13], tubular heat exchangers [14,15], heat exchangers restricted to perfect gas flows [16], balanced cross-flow recuperative plate-type heat exchangers with unmixed fluids [17]; and a parallel-plate ideal gas counterflow heat exchanger [8]. The $\varepsilon$-*NTU* method, based on the second law of thermodynamics, can be used to get the outlet temperatures and the total heat transfer from the hot fluid to the cold fluid in a heat exchanger [8,16,17].

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When mounting a black solar receiver at the focus of a parabolic dish concentrator, it can be sized such that it absorbs the maximum heat [3]. Convection losses can be drastically reduced with the use of a cavity receiver. Different types of cavity receivers have been compared [18–20]. The modified cavity receiver is suggested by Sendhil Kumar and Reddy [20] since it experiences lower convection heat losses. For the modified cavity receiver, a numerical investigation of natural convection heat loss is available [21], the contribution of radiation losses is considered [22] and an improved model for natural convection heat loss is available [23].

Exergy analysis has been applied in various power studies [24–26]. Exergetic analysis for a regenerative Brayton cycle with isothermal heat addition and isentropic compressor and turbine [27] is available. Thermodynamic analyses and optimisation of a recompression $N_2O$ Brayton power cycle have been done [28]. The performance of a regenerative Brayton heat engine has been studied by focusing on the minimisation of irreversibilities [29].

With an exergy analysis of the open and direct solar thermal Brayton cycle, the geometries of a modified cavity receiver [20–23] and counterflow plate-type recuperator [30] have been optimised [31] for various configurations of micro-turbines and concentrator diameters, so that the system produces maximum net power output. In this paper, the operating conditions of a single optimised configuration are given at various steady-state conditions. The effects of environmental parameters (geometry and environment conditions) on the optimum operating conditions are investigated.

## 2. Model

### 2.1. The control volume

The open and direct solar thermal Brayton cycle with recuperator is shown schematically in Fig. 1. A parabolic concentrator provides the solar heat for the cavity receiver. For a specific concentrator with constant diameter, focal length and rim angle, the rate of steady-state intercepted heat by the cavity receiver, $\dot{Q}^*$, depends on the cavity receiver aperture (which depends on the geometries of the cavity receiver). $\dot{Q}^*$ can be regarded as the intercepted heat rate at the receiver, after the irreversibility rates due to scattering from the concentrator and the transformation of radiation at the receiver have been deducted. $\dot{W}_{\text{net}}$ is the net power output of the system.

### 2.2. Solar receiver model

The modified cavity receiver suggested by Reddy and Sendhil Kumar [23] is considered in the analysis and is shown in Fig. 2. The receiver inner surface is made up of a closely wound copper tube

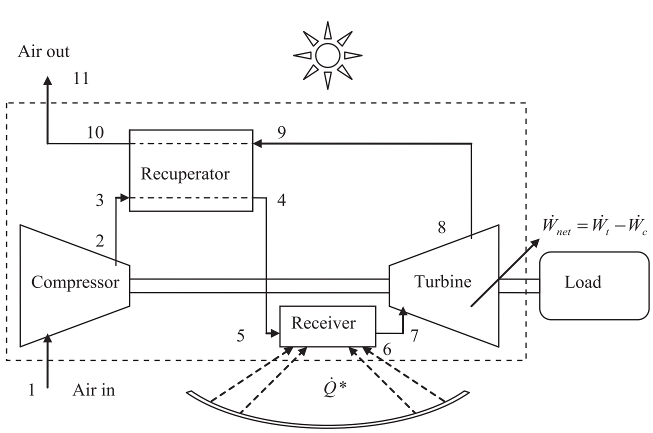


**Fig. 1.** The open and direct solar thermal Brayton cycle with recuperator.

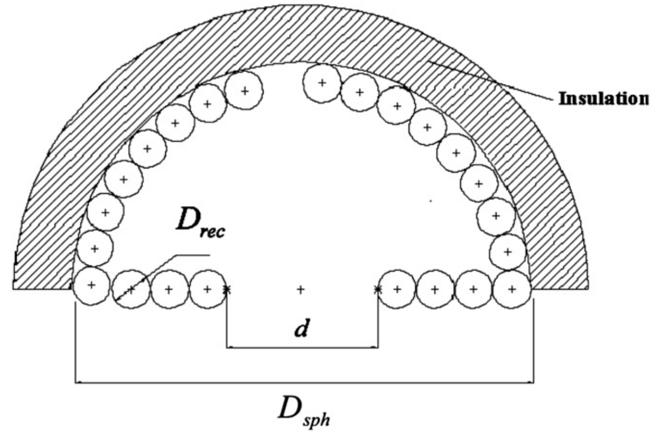


**Fig. 2.** Modified cavity receiver.

with diameter, $D_{\text{rec}}$, through which the working fluid travels. The receiver tube with length, $L_{\text{rec}}$, constructs the cavity receiver and its aperture. The receiver diameter, $D_{\text{sph}}$, is a multiple of the aperture diameter of the receiver, $d$. In this analysis, this multiple is fixed. An area ratio of $A_w/A_a = 8$ is recommended [23] as it was found to be the ratio that gives the minimum heat loss from the cavity receiver. The convection heat loss takes place through the receiver aperture. Since the surface area of a sphere is $\pi D^2$, the diameter of the spherical receiver can be calculated 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 $A_w = D_{\text{rec}}L_{\text{rec}}$.

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

According to Reddy and Sendhil Kumar [23], for $A_w/A_a = 8$, the Nusselt number for natural convection heat loss based on receiver diameter for a 3-D receiver model can be calculated as a function of the inclination angle of the receiver,

$$Nu_D = h_{\text{conv}}D_{\text{sph}}/k = 0.698\, Gr_D^{0.209}(1+\cos\beta)^{0.968} \times (T_w/T_0)^{-0.317}(d/D_{\text{sph}})^{0.425} \tag{4}$$

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 0.9 and 1.45 [22]. For the heat loss rate from the cavity receiver, it is therefore assumed that $\dot{Q}_{\text{loss,rad}} \approx \dot{Q}_{\text{loss,conv}}$ or $\dot{Q}_{\text{loss}} \approx 2\dot{Q}_{\text{loss,conv}}$. The total rate of heat loss due to convection and radiation, is approximated as

$$\dot{Q}_{\text{loss}} \approx 1.396\, Gr_D^{0.209}(1+\cos\beta)^{0.968}(T_w/T_0)^{-0.317} \times (d/D_{\text{sph}})^{0.425}(kA_a/D_{\text{sph}})(T_w - T_0) \tag{5}$$

### 2.3. Determination of net absorbed power

In practice, reflected rays from a solar concentrator form an image of finite size centred around its focal point. This is due to the sun's rays not being truly parallel and due to concentrator errors. The larger the receiver aperture diameter, the larger the rate of heat