Energy Conversion and Management 103 (2015) 616–622

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Thermo-economic modeling and optimization of an irreversible solar-driven heat engine

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ARTICLE INFO

Article history:
Received 4 March 2015
Accepted 30 June 2015
Available online 14 July 2015

Keywords:
Power output
Thermoeconomic performance
NSGA-II method
Solar-driven heat engine

ABSTRACT

The present paper illustrates a new thermo-economic performance analysis of an irreversible solar-driven heat engine. Moreover, aforementioned irreversible solar-driven heat engine is optimized by employing thermo-economic functions. With the help of the first and second laws of thermodynamics, an equivalent system is initially specified. To assess this goal, three objective functions that the normalized objective function associated to the power output (F_P) and Normalized ecological function (F_E) and thermal efficiency (η_{th}) are involved in optimization process simultaneously. Three objective functions are maximized at the same time. A multi objective evolutionary approaches (MOEAs) on the basis of NSGA-II method is employed in this work.

1. Introduction

Sahin [1] carried out a study on the thermodynamic performance analysis of an endoreversible solar-driven heat engine in the year 2000. The above mentioned scholar considered a radiative heat transfer between the working fluid and the hot reservoir; however, the Newtonian heat transfer law is employed for the mechanism of the heat transfer between the cold reservoir and the working fluid. Furthermore, he gained the optimum values for the working fluid temperature and the efficiency of the heat engine while it operates at the maximum power settings. Barranco-Jimenez et al. [2] performed a study in 2008 on the optimal operation settings for an endoreversible heat engine using various laws of heat transfer at the thermal connections while the thermodynamic cycle works under maximum ecological function conditions. In the aforementioned works, the economical approach was not adopted in the thermodynamic analysis. Sahin and Kodal [3] introduced the effect of the economic aspects to investigate the thermoeconomic performance of an endoreversible heat engine under the condition of maximum profit function. The profit function is identified as the values for the output power, the annual investment and the full consumption costs. However, De Vos already studied the thermoeconomic performance of a model for a power plant by employing the same kind of thermoeconomic analysis to investigate the performance of the Novikov type [4]. Sahin and colleagues [5] performed a thermo-economic performance analysis called the finite-time thermo-economic optimization method to investigate the thermoeconomic performance for an endoreversible solar driven-heat engine. In their research, an objective function was suggested by Sahin and colleagues [5] which defines in terms of the ratio of output power per unit total investment cost. Barranco-Jimenez et al. [6] has recently applied an investigation on the optimum thermoeconomic operation conditions for a solar-driven heat engine. In the aforementioned work, the scholars considered three systems for the performance: the maximum ecological function regime (MER) [7,8], the maximum efficient power [9] and the maximum power regime (MPR) [10–12]. By including the internal irreversibilities and heat leakage effects, in the year 2007, Ust [13] executed an analysis on the study implemented by Sahin and colleagues [5] for the thermo-economic performance analysis of an endoreversible solar-driven heat engine. In the aforesaid Finite-Time Thermodynamics and Finite-Time Thermoeconomics papers, the jointed heat transfer law was not used at maximum ecological function. In the current work, the thermoeconomics of an irreversible heat engine was studied including the heat leakage between thermal reservoirs, the thermal losses arise from the heat transfer across the finite temperature differences, and internal irreversibilities using a variable which is derived from the Clausius inequality. It is assumed that the heat transfer between the working fluid and the hot reservoir is the simultaneous combinations of radiation and conduction mechanisms and the heat flows to the cold reservoir through the conduction mechanism [13].

Multi-objective optimization is a robust approach to solve different engineering problems at different research fields [14–16].

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http://dx.doi.org/10.1016/j.enconman.2015.06.088
0196-8904/

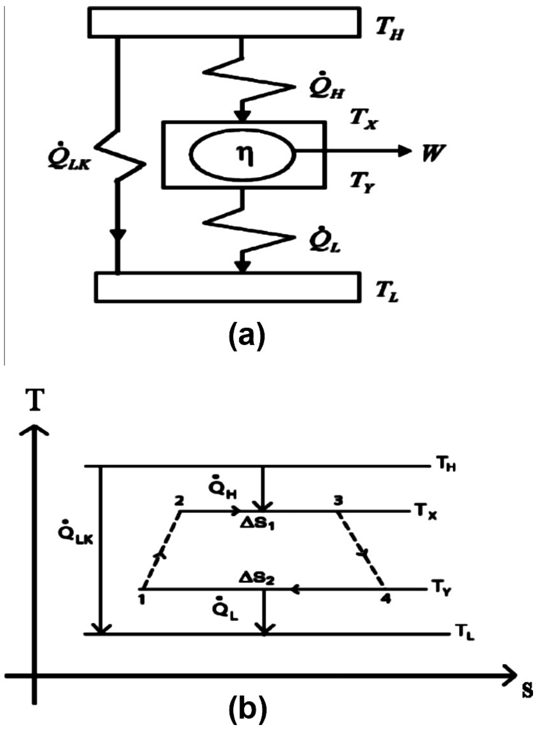

Fig. 1. Schematic diagram of the irreversible heat engine and its *T–S* diagram [5,38].

Solution to a multi-objective optimization quandary needs concurrent fulfilment of various objectives. Therefore, through the 20th century, the evolutionary algorithms (EA) were introduced and developed in an effort to solve multi-objective problems using random approaches [17]. An appropriate method to solve a multi-objective dilemma is to inquiry a collection of routes, each of them convinces the objectives at a satisfactory degree away being overcome by another route [18]. Multi-objective optimization problems typically stand for a feasibly countless group of routes called frontier of Pareto, whose assessed vectors illustrate the foremost feasible exchanges throughout the region of the objective function. Recent researches showed that multi-objective optimizations for various thermodynamic and energy systems are employed in various field of engineering [19–35].

In this work, multi-objective optimization algorithms are employed to maximize the normalized objective function with regards to the power output (F_P), the normalized ecological function (F_E) and the thermal efficiency (η_{th}). Error analysis is implemented to obtain the precision of the result gained from three well-known decision makers.

2. Theoretical model

As Fig. 1a depicts, the considered process for an irreversible solar-driven heat engine works between the collector (the heat source) at temperature T_H and the cooling water (heat sink) at temperature T_L. The working fluid temperatures which transfers heat with the reservoirs at temperatures T_H and T_L are respectively denoted by T_X and T_Y. Fig. 1b illustrates a *T–S* diagram for the process approach which includes the finite-time heat transfer, heat leakage, and internal irreversibilities. The heat exchanging process from the hot reservoir is considered a concurrent combinations of conduction and radiation mechanisms. The net rate for the heat flow $\dot{Q}_H$ between the heat engine and the hot reservoir is expressed by [13],

$$\dot{Q}_H = \dot{Q}_{HC} + \dot{Q}_{HR} = U_{HC}A_H(T_H - T_X) + U_{HR}A_H(T_H^4 - T_X^4) \quad (1)$$

In Eq. (1), U_{HC} and U_{HR} denote the heat transfer coefficients for convection and radiation mechanisms, correspondingly, and A_H stand for the area of heat transfer for the hot-side heat exchanger. Furthermore, conductive heat transfer is considered to be the dominant mechanism for the heat transfer to the low-temperature reservoir. Hence, the rate of heat flow $\dot{Q}_L$ from the heat engine to the cold reservoir is expressed by,

$$\dot{Q}_L = U_{LC}A_L(T_Y - T_L) \quad (2)$$

In Eq. (2), U_{LC} is the heat transfer coefficient and A_L stands for the heat transfer area for the heat exchanger both calculated at the cold sides. Assuming the internal conductance of the heat engine to be γ, the heat leakage flow rate $\dot{Q}_{LK}$ between the cold reservoir with temperature T_L and the hot reservoir with temperature T_H is expressed by,

$$\dot{Q}_{LK} = \gamma(T_H - T_L) = \xi U_H A_H(T_H - T_L) \quad (3)$$

In Eq. (3), ξ stands for the percentage of the internal conductance with regards to the heat transfer area and hot-side conduction heat transfer coefficient calculated by $\xi = \left(\frac{\gamma}{U_H A_H}\right)$ [13]. Therefore, the total rate of heat exchange $\dot{Q}_{HT}$ transferred from the hot reservoir is given by,

$$\dot{Q}_{HT} = \dot{Q}_H + \dot{Q}_{LK} \quad (4)$$

and the total rate of heat exchange $\dot{Q}_{LT}$ transferred to the cold reservoir is expressed by,

$$\dot{Q}_{LT} = \dot{Q}_L + \dot{Q}_{LK} \quad (5)$$

The first law of thermodynamics can be applied to give the output power as,

$$\dot{W} = \dot{Q}_{HT} - \dot{Q}_{LT} = \dot{Q}_H - \dot{Q}_L \quad (6)$$

Using Eqs. (1), (2) and (6) yields the normalized expression for the output power $\bar{W} = \left(\frac{\dot{W}}{U_{HC}A_H}\right)$, expressed as,

$$\bar{W} = (T_H - T_X) + \beta\frac{(T_H^4 - T_X^4)}{T_H^4} - \psi A_R(T_Y - T_L) \quad (7)$$

where $\beta = \left(\frac{U_{HR}}{U_{HC}}\right)T_H^3$, $\psi = \left(\frac{U_{LC}}{U_{HC}}\right)$ and $A_R = \left(\frac{A_L}{A_H}\right)$. The second law of thermodynamics is utilized to obtain the irreversible part for the model as,

$$\oint \frac{dQ}{T} = \frac{\dot{Q}_H}{T_X} - \frac{\dot{Q}_L}{T_Y} < 0 \quad (8)$$

The inequality in Eq. (8) may be rewritten using so-called non-endoreversibility parameter R as [36–39],

$$\frac{\dot{Q}_H}{T_X} = R\frac{\dot{Q}_L}{T_Y} \quad (9)$$

The non-endoreversibility parameter which is within the interval $0 < R \leqslant 1$ in principle ($R = 1$ used for the endoreversible case), can be seen as a measure of the difference from the endoreversible regime [36–39]. Introducing Eqs. (1) and (2) into Eq. (9), a relationship between T_Y and T_X is given by,

$$\frac{T_Y}{T_L} = \frac{R\psi A_R}{R\psi A_R - \frac{(1-\theta)}{\theta} - \beta\frac{(1-\theta^4)}{\theta}} \quad (10)$$

where $\theta = \left(\frac{T_X}{T_H}\right)$. Furthermore, the thermal efficiency of the irreversible heat engine is formulated as following as,