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Optimization of heat sink of thermoelectric cooler using entropy generation analysis

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A B S T R A C T

In the present study, an optimization model is developed for a thermoelectric cooler (TEC) based on the entropy generation minimization method. In the model, the total entropy generation rate, the entropy generation number and the exergetic efficiency are proposed as optimization objective functions, respectively, while the number of transfer units of the heat sink is considered as a constraint. The heat capacity rate of cooling fluid in the heat sink is optimized under other given conditions. The results show that a minimum total entropy generation rate and a minimum entropy generation number can be achieved by optimally selecting the heat capacity rate of the cooling fluid. In addition, the minimum total entropy generation rate and corresponding optimal heat capacity rate of the cooling fluid are both dominated by the number of transfer units and the electrical current. Furthermore, the effects of the heat capacity rate of cooling fluid and the electric current in terms of the exergetic efficiency are also evaluated.

1. Introduction

The increasing demand for electronic devices with small sizes and functionality enhancement has resulted in significant heat dissipation in those electronic devices. The high operating temperature caused by high heat flux could potentially lead to the failure of electronic component or the decrease in efficiency and reliability of electronic devices. Thus, an efficient cooling method is highly desirable to these electronic devices requiring precise control of the temperature. Among the various temperature control solutions for the electronic devices, the traditional passive cooling technologies are commonly used, including the forced-air cooling, liquid cooling, and heat pipe cooling. Compared to these convectional passive cooling technologies, thermoelectric coolers (TECs) can actively control the cooling capacity, and therefore has become dramatically popular as they have relatively small size and weight, noise free operation and absence of moving parts [1–3].

The performance of a TEC has been evaluated through both numerical models and experiments. Many efforts have been devoted to predict the performance of the TEC [4–6], to simulate the transient thermal behavior of the TEC [7,8] and to optimize the material properties and geometry of thermoelectric elements in the TEC [9,10]. Among these researches, the optimization of the TEC system with reasonable system constraints is a crucial problem, which needs to be solved in the design. Cheng and Lin [11] proposed a method of optimizing the dimensions of the TEC legs using the genetic algorithm to maximize the cooling capacity, and found that optimizing the dimensions of the TEC can increase its cooling capacity. Robert et al. proposed [12] an optimization methodology for the TEC used for electronic cooling, and found that a TE design candidate could both maximize the COP and minimize the junction temperature. In these studies, the first law of thermodynamics is the mostly widely utilized method. In addition, most of them mainly focused on the optimization of TEM characteristic and geometric variables. However, the real processes of TEC system present irreversibility and are associated with the entropy generation. In this case, the entropy generation optimization method could be a powerful tool to address the practical TEC optimization problem. Besides, this approach based on the combination of the first and second law of thermodynamics gives a more realistic view of the energy conversion and heat transfer processes, and therefore enables to distinguish the various mechanisms of irreversibility in a TEC system. Furthermore, the entropy generation minimization approach introduced by Bejan [13] has been widely used for various thermal systems and heat exchangers such as plate-fin exchanger reported in the previous literature [14–19]. Thus, the entropy

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Nomenclature

b_1	channel width (m)
c_p	specific heat (J kg^{-1}K^{-1})
COP	coefficient of performance
D	diameter (m)
f_{app}	friction coefficient
H	height (m)
h_{eff}	heat transfer coefficient of cooling fluid (WK^{-1}cm^{-2})
I	electrical current (A)
k	thermal conductivity (WK^{-1}cm^{-1})
K_c	dimensionless contraction coefficient
K_e	dimensionless expansion coefficient
L	length (m)
$\dot{m}$	mass flow rate (kg s^{-1})
N	number
Ns	entropy generation number
Nu	Nusselt number
NTU_h	number of transfer units of cooling fluid
p	perimeter (m)
P	pressure (kPa)
Pr	Prandtl number
Q	heating capacity (kW)
R	electric resistance/thermal resistance (Ωm)/(KW^{-1})
R_g	value of gas constant
Re	Reynolds number
S	entropy generation rate (WK^{-1})
T	temperature (°C)
U	thermal conductance (WK^{-1}cm^{-2})
V	velocity (m s^{-1})
ν	**kinematic** viscosity (m^2 s^{-1})
w	input power (W)
W	weight (m)

Greek symbols

α	Seebeck coefficient (V K^{-1})
δ_b	thickness of the base plate (m)
η	exergetic efficiency
ε_h	heat exchanger effectiveness
ρ	density (kg m^{-3})

Subscripts

av	average
c	cold/reverse Carnot cycle
cr	cross-sectional
f	fluid
fin	fin
gt	total
h	hot/hydraulic
TEM	thermoelectric module
m	module

generation minimization methodology can be good alternatives for optimization problems in TEC.

In this study, by combing the first and second law of thermodynamics, the present study provides a mathematical model to explore the optimum heat sink configuration for TEC system based on the entropy generation analysis method. Except the internal irreversibility of TEC system, i.e. thermoelectric module (TEM), the external irreversibilities of TEC system are also considered to offer a more comprehensive and practical optimization method for TEC system, including the heat source (an electronic chip) and the heat sink (a plate-fin heat exchanger). Since the number of transfer units of the cooling fluid (NTU_h) is a crucial parameter with respect to the heat capacity rate of cooling fluid for a TEC system, it becomes necessary to establish a relationship between the entropy generation and the NTU_h. For this purpose, the total entropy generation rate and the number of transfer units of the cooling fluid are considered as the objective functions and an equality constraint, respectively, to perform the optimization of a TEC. Moreover, a dimensionless entropy generation number based on the heat capacity rate of cooling fluid in the heat sink and TEC exergetic efficiency are also discussed in the paper. Based on the model, a theoretical study is conducted to explore the optimal heat sink configuration for the TEC system. The purpose of this paper is to provide optimum design candidates for TEC systems.

2. Mathematical modeling

Fig. 1(a) shows the configuration of a single stage TEC system, in which TEM is sandwiched between an electronic chip and a plate-pin heat exchanger. It should be noted that we do not select a real electronic chip for the modeling, which is only represented by a schematic cooled object with a certain heat load or temperature. In the plate-fin heat exchanger, air as heat transfer fluid is used to dissipate heat from the TEM. Fig. 1(b) gives a schematic diagram of the plate-fin heat exchanger, in which the main geometries are also provided. As shown in Fig. 1 (b), the plate-fin heat exchanger runs in a flow-through mode. The TEC system is modeled by using the steady-state one-dimensional method, which has been carried out in the majority of previous literature. The following assumptions are made in the model:

1) Thermal and electrical properties of the thermoelectric material are constant;
2) Thermal and electrical contact resistances are neglected;
3) Thomson effect is not taken into account;
4) The air flow through the plate-fin heat exchanger is steady and can be assumed to be one-dimensional;
5) The axial heat conduction inside the heat exchanger is ignored.

For a given TEC module, the rejected heat at the hot side Q_h, and heat absorbed at the cold side, Q_c, can be written, respectively, as

$$Q_c = N_t\left[\alpha I T_c - K(T_h - T_c) - \frac{1}{2}RI^2\right] \quad (1)$$

$$Q_h = N_t\left[\alpha I T_h - K(T_h - T_c) + \frac{1}{2}RI^2\right] \quad (2)$$

Where N_t is the number of thermocouples in the TEM. I is the electric current. α, K and R are the Seebeck coefficient, thermal conductance and the electric resistance of a thermocouple, respectively. T_c and T_h are the cold and hot junction temperatures of the TEM.

Base on the heat balance of the TEM, the Q_c and Q_h are the heat transfer rates of the cold side and the hot side heat exchanger, respectively. For the cold side heat exchanger (conduction plate), the absorbed heat Q_c can be expressed as

$$Q_c = U_c A_c (T_c' - T_c) \quad (3)$$

where T_c' is the temperature of the cooled object, $U_c A_c$ is the overall heat transfer coefficient of the cold side heat exchanger from the