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Thermal performance of two heat exchangers for thermoelectric generators

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ABSTRACT

The thermal performance of a heat exchanger is important for the potential application in an integrated solar cell/module and thermoelectric generator (TEG) system. Usually, the thermal performance of a heat exchanger for TEGs is analysed by using 1D heat conduction theory which ignores the detailed phenomena associated with thermo-hydraulics. In this paper, thermal and momentum transports in two different heat exchangers are simulated by means of a steady-state, 3D turbulent flow k-ε model with a heat conduction module under various flow rates. In order to simulate the actual working conditions of the heat exchangers, a hot block with an electric heater is included in the model. The TEG module is simplified by using a 1D heat conduction theory, so its thermal performance is equivalent to a real TEG. Natural convection effects on the outside surfaces of the computational domains are considered. Computational models and methods used are validated under transient thermal and electrical experimental conditions of a TEG. The two heat exchangers designed in this paper have better thermal performance than an existing heat exchanger for TEGs. More importantly, the fin heat exchanger is more compact and efficient than the tube heat exchanger.

1. Introduction

The thermal performance of a heat exchanger, which is important for an integrated solar cell/module and thermoelectric generator (TEG) system, has been paid little attention to date. From a comprehensive literature review, it is found that the performance analysis of a heat exchanger for TEG application is usually done by using a simple 1D heat conduction theory. For example, free convection heat transfer and heat conduction in a flat-plate heat exchanger of TEG for an ocean thermal energy conversion (OTEC) system was analysed by Henderson [1] with a 1D heat transfer theory. Consequently, the heat exchanger-TEG system performance was investigated and a relationship between the TEG element area-to-length ratio and the temperature difference across the TEG and resulting electric power generation was obtained. Likewise, the thermal performance of a parallel-plate heat exchanger was calculated by Yu and Zhao [2]. As a result of this, the TEG performance

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associated with the exchanger was predicted analytically, and the heat exchanger-TEG system performance was explored further by means of a parametric study. Again by employing a basic 1D heat transfer model, three types of heat exchanger such as spiral, zig-zag and straight fin geometries for the TEG's cold and hot sides were investigated by Esarte et al. [3]. The overall heat transfer coefficient for the heat exchanger-TEG system was found to be 2.07 W/°C, 2.094 W/°C and 1.72 W/°C, respectively. Particularly, for the spiral heat exchanger, experiments were carried out to confirm the analytical temperature difference across the TEG module at varying water flow rate. As evidenced, this 1D simulation method may provide useful information but clearly ignores the effects associated with combined thermal and hydraulic occurrences within systems.

Recently, computational fluid dynamics (CFD) methodology has been applied to characterise the thermal performance of various heat exchangers such as plate, shell and tube, vertical mantle, compact and printed circuit board exchangers and also to optimize their design configurations (e.g. see Bhutta et al. [4]). It was pointed out that CFD has emerged as a cost effective tool to speed up the investigation of various heat exchanger designs as well as their hydraulic and thermal performances. It has also been noticed that the k-ε turbulence model has been extensively employed and an accuracy of 2–10% has generally been achieved against experimental data [4]. For example, a plate heat exchanger with undulated surfaces was optimized with a CFD package, ANSYS CFX 10.0, as well as with a response surface method conducted by Kanaris et al. [5]. Their chosen computational model was a 3D narrowed channel with inclined triangular undulations in herringbone pattern. The design variables included a blockage ratio, channel aspect ratio, corrugation aspect ratio, angle of attack and Reynolds number. The objective function was a linear combination function of heat transfer augmentation and pressure drop.

A method was also developed by Karmo et al. [6] to design an effective finned heat exchanger. The tube cross-sectional and longitudinal shapes were changed so that the angle between the fins and the tubes was on longer orthogonal. Various configurations were analysed using the CFD code, Fluent, to identify an optimal design. Recently, elliptical finned-tube heat exchanger thermal-hydraulic performance was optimized by using Fluent and response surface method based on seven design variables proposed by Sun and Zhang [7]. It was found out that a slightly increased axis ratio of elliptical tube section can improve the thermal-hydraulic performance at a higher air velocity outside the tubes or lower water flow rate in the tubes.

However, the literature also suggests that the CFD analysis leading to design and performance optimization of heat exchanger for TEGs is very limited. A hybrid TEG/thermal system with a radiation concentrator was proposed recently by Urbiola and Vorobiev [8]. An optimal configuration of the flat plate heat exchanger placed in the cold side of TEG was identified by using COMSOL Multiphysics software, and results showing the thermal energy stored in cooling water around 50 °C at midday. Most recently, TEGs and heat exchangers have been simulated by Sarhadi et al. [9] with the same software (COMSOL) and particularly the TEG arrangements layouts were investigated. The TEGs are considered to be a solid body, and the oil flows in the exchanges are considered to be laminar. Thermal contact resistance is also included in the interfaces between the TEGs and exchangers. However, the cooling oil flow rate is assumed to be distributed uniformly from one finned cannel to another.

In this contribution, the thermal performance of two heat exchangers on the cold side of a TEG is studied by means of ANSYS 15.0 CFX to identify their optimal configuration. The computational model includes a hot block-electrical heater, a TEG, a cold block-heat exchanger which maintains the TEG cold side to be in lower temperature. Since CFX is unable to deal with electrical modelling, the real TEG is represented by a thermal equivalent TEG with the same geometrical size. The heater is subject to the highest temperature, and cooling water at the exchanger inlet is kept at 19 °C. Natural convection effects on the outside surfaces of the computational domains due to environmental effects are considered. Water flow in the heat exchangers is turbulent, so turbulent eddy shear stresses are modelled by means of a k-ε turbulence model.

2. Computational models and methods

2.1. Computational models

Two heat exchangers are designed, one is a tube exchanger and the other is a fin exchanger, as shown in Fig. 1. Obviously, the fin heat exchanger is more compact than the tube exchanger. The heat exchanger is firmly attached to the top surface of a TEG by a pressure load. The water flow in the exchanger absorbs the heat discharged by the TEG when it generates electric power. The TEG is heated underneath through its substrate by an electrical heater inside the hot block. As a result, the interface between the exchanger and the TEG is kept at a lower temperature, but the interface between the hot block and the TEG is at a higher temperature. Eventually, a temperature difference is established between both sides of the TEG and electrical power is generated. To deal with an uneven distribution of temperature on the interfaces between the TEG and both the hot block and heat exchanger, three components must be considered simultaneously in a heat transfer simulation.

The TEG structure is simplified to a solid structure so as to make the heat transfer analysis easy, see Fig. 2. This simplified structure keeps its thermal behaviour being equivalent to the TEG based on a 1D heat conduction theory illustrated by Incropera et al. [10]. In the simplification, a few assumptions are made: (1) the materials of each component of the TEG are homogenous and their thermal resistance, heat specific capacity and density are constant and independent of temperature; (2) the thermal contact effect inside the TEG and on the interfaces between both the heater and heat exchanger and the TEG are negligible; (3) the effects of the air inside the TEG on the free heat convection and the equivalent density of the TEG are not taken into account. The heat conductivity of the copper straps is two-orders of magnitude larger than those of the legs and substrates, thus the heat resistance in the copper straps is also ignored. Additionally, the copper straps are as thin as 0.2 mm, so their contribution to the equivalent density of the TEG is excluded.