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A numerical model for thermoelectric generator with the parallel-plate heat exchanger

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Abstract

This paper presents a numerical model to predict the performance of thermoelectric generator with the parallel-plate heat exchanger. The model is based on an elemental approach and exhibits its feature in analyzing the temperature change in a thermoelectric generator and concomitantly its performance under operation conditions. The numerical simulated examples are demonstrated for the thermoelectric generator of parallel flow type and counter flow type in this paper. Simulation results show that the variations in temperature of the fluids in the thermoelectric generator are linear. The numerical model developed in this paper may be also applied to further optimization study for thermoelectric generator.

Keywords: Thermoelectric generator; Power generation; Heat exchanger; Waste-heat recovery

1. Introduction

Thermoelectric generator is a useful device for direct energy conversion. Due to its relatively low conversion efficiency the applications in electrical power generation has been restricted in specialized fields, such as medical, military and space applications over the past. However, thermoelectric generators present potential application in the conversion of low-grade thermal energy into electrical power. Especially for waste-heat recovery, it is unnecessary to consider the cost of the thermal energy input, and thus the low conversion efficiency is also not an overriding consideration [1]. Therefore, in the past more efforts have been bestowed in thermoelectric generators to develop its applications for high power generation.

Wu [2] performed theoretical analysis on waste-heat thermoelectric power generators. In this study, a real waste-heat thermoelectric generator model was presented based on accounting for both internal and external irreversibility to predict realistic specific power and efficiency. The internal irreversibility is caused by the Joulean loss and conduction heat transfer, and the external irreversibility is caused by the temperature differences between the hot and cold junctions and the heat source and sink in the real waste-heat thermoelectric generator. Therefore, this approach gave a much more realistic generator specific power and efficiency prediction than does the ideal thermoelectric generator. Rowe and Gao [3] developed a procedure to assess the potential of thermoelectric modules when used for electrical power generation. The method was used to evaluate several commercially available modules and the results showed that a thermoelectric module is a promising device for low-temperature waste-heat recovery. Chen and Wu [4] used an irreversible model to study the performance of a thermoelectric generator with external and internal irreversibility. In this study, the optimal range of the parameter for device-design was determined and the problems relative to the maximum power output and maximum efficiency were discussed. Esarte et al. [5] applied a NTU–ε methodology to study the influence of fluid flow rate, heat exchanger geometry, fluid properties and inlet temperatures on the power supplied by the thermoelectric generator. The work could provide some guidelines for determining the optimum operating parameters of thermoelectric generator. Stevens [6] proposed an approximate procedure for the optimal coupled design of a thermoelectric generator with a small temperature difference. In this analysis, an approximate optimal design was derived that is applicable to systems with a small temperature difference between the reservoirs. For a fixed thermal resistance in the heat exchangers, the optimal configuration split the total temperature drop evenly between the thermoelectric

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Nomenclature

A	heat transfer area (m^2)
B	the width of fluid passage (m)
c	specific heat capacity at constant pressure ($J\,kg^{-1}\,K^{-1}$)
C_1, C_2	constants
d_e	hydraulic diameter (m)
f	fanning friction factor
G	mass flow rates ($kg\,s^{-1}$)
h	heat convection coefficient ($W\,m^{-2}\,K^{-1}$)
H	the height of fluid passage (m)
I	current (A)
k	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
K	thermal conductance of thermocouples ($W\,K^{-1}$)
l	the height of thermocouples (m)
L	the length of heat exchanger (m)
N_{TE}	the number of thermocouples
Nu	Nusselt number
P	power output (W)
Pr	Prandtl number
Q	the rate of heat flow (W)
R	electrical resistance (Ω)
Re	Reynolds number
R_L	the load electrical resistance (Ω)
R_t	thermal resistance ($m^2\,K\,W^{-1}$)
t	temperature (°C)
T	temperature (K)
ΔT	temperature difference of TE (°C)
u	velocity of flow ($m\,s^{-1}$)
w	the width of a single thermocouples leg (m)
x	axial distance along hot fluid tube flow axis (m)

Greek symbols

α	Seebeck coefficient ($V\,K^{-1}$)
η	conversion efficiency
λ	thermal conductivity of fluids or TE material ($W\,(m\,K)^{-1}$)
ρ	electrical resistivity of TE material (Ω m)
ν	kinematic viscosity ($m^2\,s^{-1}$)

Subscripts

c	cold side
f	fluid
h	hot side
m	middle
TE	thermoelectric

module and the heat exchangers. For heat exchanger thermal resistances that vary with time, optimal configuration were also derived for linear, sawtooth function and sinusoidal variations. Chen et al. [7] investigated the characteristics of a multi-element thermoelectric generator with the irreversibility of finite-rate heat transfer. The effects of heat transfer and the number of elements on the performance were analyzed and concluded that the optimal current and the optimal number of thermocouples must be choose in the points of view of the compromise optimization between power output and efficiency in order to obtain the best performance. Pramanick and Das [8] conducted the study on constructal design of a thermoelectric device. Based on finite time thermodynamics, the model for a cascaded thermoelectric generator was developed, which incorporated all the essential features of a real heat engine. In the model, control volume formulation of cascaded thermocouples was carried out over a small but finite temperature gap to comply with the principles of irreversible thermodynamics. Several design considerations were also discussed for best possible device performance.

In the above literatures, those research works have mainly been focused on the analytical analysis to a single or multiple thermoelectric elements for thermoelectric generator. This analytical method provided some significant guidelines for the design of the thermoelectric generator in terms of the principle of thermodynamics. However, in practical situation a real thermoelectric generator always combine thermoelectric modules with the concrete heat exchangers in very large scale installations to produce useful amounts of electricity from low-grade heat sources. Therefore, research on the constructal design of the thermoelectric conversion systems and operating performance also has attracted researchers' wide attention in this area.

Suzuki and Tanaka [9] studied thermoelectric power generation with the multi-panels. Electric power was estimated in case of the large-scale flat thermoelectric panels exposed to two thermal fluids. The output powers of the proposed 15 systems were analytically deduced from heat transfer theory, and written by non-dimensional functions to reflect the characteristics of system design. Another study on the thermoelectric power generation with cylindrical multi-tubes was also carried out by Suzuki and Tanaka [10]. In the study, the mathematical expression of electric power was deduced in case of the large-scale cylindrical thermoelectric tubes exposed to two thermal fluids. Crane and Jackson [11] investigated thermoelectric generator with advanced heat exchangers for waste-heat recovery. Numerical heat exchanger models integrated with models for Bi_2Te_3 thermoelectric modules were created and validated. The results showed that heat exchangers with Bi_2Te_3 thermoelectrics could achieve net power densities over $40\,W\,l^{-1}$.

Generally, in the case of thermoelectric generator which involves different types of heat exchangers, the problem of the heat transfer through heat exchanger and the thermal energy conversion become very complex. The more precise analysis for thermoelectric generator with the concrete type of heat exchanger should be taken into consideration in order to estimate the device performance. The numerical analysis provides this effective approach for evaluating performance of thermoelectric generator. In particular, the numerical approaches are able to offer much more detailed prediction over analytical method for thermoelectric generator performance. Although, some work has been done in previous literatures, more information such as the temperature changes at heat exchanger and the temperature difference through thermoelectric elements in the thermoelectric generator with the concrete type of heat exchanger is required to identify aspects of further improvement in the generating performances of thermoelectric generator.

The thermoelectric generator with the parallel-plate heat exchanger is more appropriate for practical applications in waste-heat recovery because its configuration is simple and cost is low. Therefore, in this paper we present a more detailed numerical model of thermoelectric generator with the parallel-plate heat exchanger, which is developed based on an elemental approach for simulating the performance of this type of thermoelectric generator. This new study focuses on analyzing the fluid temperature changes along the fluid passage and the temperature difference through thermoelectric modules (TE). The effects of the inlet temperature and flow rate of hot fluid on the performance under operation conditions are also analyzed. This work is intended to give some guidelines for the further design and optimization of the heat exchangers as well as the matching of the heat exchanger and thermoelectric designs of thermoelectric generator for waste-heat recovery.

2. Model description

As it is known there are four types of heat exchangers for heat transfer between two fluids classified by the directions of hot and cold fluids. Generally, parallel flow type and counter flow type are often selected for heat exchangers. The parallel flow type is that the two fluids flow in the same direction, whereas the fluids flow in the opposite direction at the counter flow type. In literature [9], four types of thermoelectric power generation system with the multi-panels were presented. In present study, we also consider selecting the parallel flow type or counter flow type parallel-plate heat exchanger with the commercially available thermoelectric modules as the basic physical model of thermoelectric generator. In the basic physical model as shown in Fig. 1, it is assumed that the flat thermoelectric modules are sandwiched between the hot and cold fluids, which consist of multi-thermocouples with a single layer of p-type and n-type semiconductor thermocouples. The thermocouples along the directions of fluid path are connected electrically in series, but in parallel for the directions perpendicular to fluid path.

A numerical model in one-dimensional description was developed using control volume approach along with the method of average parameters. The control volume as shown in Fig. 1 is formulated for a set of thermocouples, which includes flow passages on both sides of the TE modules. The following

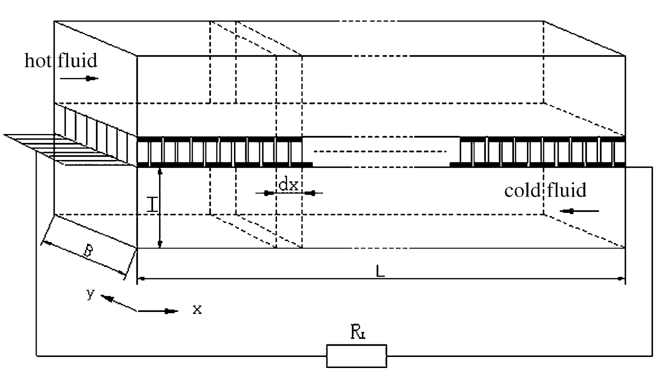

Fig. 1. Schematic layout of the model.

assumptions are made to simplify the complex problem of modeling:

(a) The axial heat conduction within the thermocouples is ignored, as transverse conduction along the thermocouples will be dominant.
(b) For simplicity, thermal resistance through the plates of parallel-plate heat exchanger, the ceramic plates and the metallic strips of TE modules is taken into account as an equivalent thermal resistance. All thermal contact resistance arises inside TE modules as well as between the TE modules and the parallel-plate heat exchanger plates are ignored.
(c) The heat losses between the parallel-plate heat exchanger and the thermoelectric modules are ignored.
(d) The control volume include a single thermocouples at least along x-coordinate direction, and a number of thermocouples along y-coordinate direction.
(e) The gap between thermocouples is ignored.

Following the above assumptions, the governing equations for the control volume can be written based on the energy balance.

Governing equations for the heat transfer on the side of hot fluid and cold fluid, respectively, are

$$\mathrm{d}Q_\mathrm{h} = -G_\mathrm{fh} c_\mathrm{fh}\,\mathrm{d}t_\mathrm{fh} \tag{1}$$

$$\mathrm{d}Q_\mathrm{h} = k_\mathrm{h}(t_\mathrm{fh} - t_\mathrm{mh})\,\mathrm{d}A \tag{2}$$

$$\mathrm{d}Q_\mathrm{c} = G_\mathrm{fc} c_\mathrm{fc}\,\mathrm{d}t_\mathrm{fc} \tag{3}$$

$$\mathrm{d}Q_\mathrm{c} = k_\mathrm{c}(t_\mathrm{mc} - t_\mathrm{fc})\,\mathrm{d}A \tag{4}$$

where heat transfer coefficient in the hot-side and cold-side heat exchanger is given by the following equations:

$$k_\mathrm{h} = \frac{1}{1/h_\mathrm{fh} + R_\mathrm{th}} \tag{5}$$

$$k_\mathrm{c} = \frac{1}{1/h_\mathrm{fc} + R_\mathrm{tc}} \tag{6}$$

where R_th and R_tc are the equivalent thermal resistance to the heat flow through the hot-side and cold-side heat exchanger.

The convective heat transfer coefficient for the fluid in both the hot-side and cold-side is calculated by [11]:

$$\frac{h_\mathrm{f} d_\mathrm{e}}{\lambda_\mathrm{f}} = Nu_\mathrm{f} = \frac{(f_\mathrm{f}/2)(Re_\mathrm{f} - 1000)Pr_\mathrm{f}}{1 + 2.7(f_\mathrm{f}/2)^{1/2}(Pr_\mathrm{f}^{2/3} - 1)} \tag{7}$$

$$Re_\mathrm{f} = \frac{u_\mathrm{f} d_\mathrm{e}}{\nu_\mathrm{f}} \tag{8}$$

$$f_\mathrm{f} = C_1 + \frac{C_2}{Re_\mathrm{f}^{1/m}} \tag{9}$$

where for laminar flow ($Re_\mathrm{f} < 2100$), $C_1 = 0$, $C_2 = 16.0$ and $m = 1.0$, for transition flow ($2100 < Re_\mathrm{f} \le 4000$), $C_1 = 0.0054$, $C_2 = 2.3 \times 10^{-8}$ and $m = -0.6667$, and for turbulent flow ($Re_\mathrm{f} > 4000$), $C_1 = 0.00128$, $C_2 = 0.1143$ and $m = 3.215$. The Fanning friction factor f_f for the fluid flow is calculated for all flow regimes. The property values for calculating Nusselt number are evaluated at the local fluid temperature. In the following calculations, turbulent flow conditions are selected due to $Re_f > 4000$ at the selected operation conditions.

For the thermoelectric module in the control volume, the corresponding energy balance equations should be written as [4]:

$$\mathrm{d}Q_h = \left(\alpha T_{mh} I + K(T_{mh} - T_{mc}) - \frac{1}{2} I^2 R\right) N_{TE} \tag{10}$$

$$\mathrm{d}Q_c = \left(\alpha T_{mc} I + K(T_{mh} - T_{mc}) + \frac{1}{2} I^2 R\right) N_{TE} \tag{11}$$

where I is the current flow through a single thermocouples, N_{TE} the number of thermocouples, α, K and R are, respectively, the Seebeck coefficient, thermal conductance and electrical resistance of a single thermocouples.

The above differential equations are discretized along the axial direction of the hot fluid flow, which is captured with a first order forward difference scheme. Thus, the difference equations are written as

$$\frac{\Delta Q_{h(i)}}{\Delta x} = -G_{fh} c_{fh} \frac{t_{fh(i+1)} - t_{fh(i)}}{\Delta x} \tag{12}$$

$$\frac{\Delta Q_{h(i)}}{\Delta x} = k_h B \left(\frac{t_{fh(i)} + t_{fh(i+1)}}{2} - t_{mh(i)}\right) \tag{13}$$

$$\frac{\Delta Q_{c(i)}}{\Delta x} = G_{fc} c_{fc} \frac{t_{fc(i+1)} - t_{fc(i)}}{\Delta x} \tag{14}$$

$$\frac{\Delta Q_{c(i)}}{\Delta x} = k_c B \left(t_{mc(i)} - \frac{t_{fc(i)} + t_{fc(i+1)}}{2}\right) \tag{15}$$

where B is the width of fluid passage.

Accordingly, Eqs. (10) and (11) for the thermoelectric model can be also written as

$$\Delta Q_{h(i)} = \left(\alpha T_{mh(i)} I + K(T_{mh(i)} - T_{mc(i)}) - \frac{1}{2} I^2 R\right) N_{TE} \frac{\Delta x}{w} \tag{16}$$

$$\Delta Q_{c(i)} = \left(\alpha T_{mc(i)} I + K(T_{mh(i)} - T_{mc(i)}) + \frac{1}{2} I^2 R\right) N_{TE} \frac{\Delta x}{w} \tag{17}$$

where w is the width of a single thermocouple leg along x-coordinate direction.

It should be noted that for heat transfer analysis of the model, the parallel-plate heat exchanger is divided into small segments and calculations are carried over each segment, which is denoted i. The elemental length Δx of each segment is specified as a multiple of the distance between adjacent thermocouple, which is set at a constant length.

The current flow through the thermocouples aligned in series for a given thermoelectric generator is determined by the number of thermocouples aligned in series and the electrical resistance of circuit load. The total voltage is the sum of the voltages generated in each segment, which are wired in series along x-coordinate direction. Therefore, the current can be written as

$$I = \frac{\sum_{i=1}^{n} \alpha(T_{mh(i)} - T_{mc(i)}) N}{R_L N_{TE} + \sum_{i=1}^{n} (RN)_i} \tag{18}$$

where $n = L/\Delta x$ is the number of segment, L the length of parallel-plate heat exchanger and $N = \Delta x / w$ is the number of thermocouples in each segment.

The total current flow through the thermoelectric generator is the summation of the current along y-coordinate direction, which can be written as

$$I_y = \frac{B}{w/2} I_x = \frac{2 \sum_{i=1}^{n} \alpha(T_{mh(i)} - T_{mc(i)}) NB}{\left(R_L N_{TE} + \sum_{i=1}^{n} (RN)_i\right) w} \tag{19}$$

The output power of the thermoelectric generator is

$$P = \sum_{i=1}^{n} \Delta Q_{h(i)} - \Delta Q_{c(i)} \tag{20}$$

The efficiency of thermoelectric generator is thus defined as

$$\eta = \frac{P}{\sum_{i=1}^{n} \Delta Q_{h(i)}} \tag{21}$$

The solution methodology of the numerical model is implemented by using an iterative method. Input parameters include the thermophysical properties of the thermocouples, configuration parameters of thermocouples and parallel-plate heat exchanger, circuit load as well as the inlet fluids temperatures and mass flow rate to the parallel-plate heat exchanger.

3. Simulation results and discussion

In the following numerical simulation, the working fluid will be water and the thermoelectric modules are commercially available the Bi_2Te_3 semiconductor. For the exchanger geometry, a unit passage of hot and cold fluid has a length of around 1.14 m and a cross-sectional area of about 1.2×10^{-2} m × 6.0×10^{-3} m. The single thermocouple has a height of around 2.0×10^{-3} m and a cross-sectional area of about 1.5×10^{-3} m × 1.5×10^{-3} m These dimensions are selected based on taking into account the available thermocouples size of the commercial thermoelectric module and constraining the heat transfer process of heat exchanger to turbulent flow conditions for having better heat transfer coefficient. The thermophysical values of Bi_2Te_3 semiconductors will be used for the thermocouples. The relevant parameters are $\alpha = 2.2 \times 10^{-4}$ V K^{-1}, $\lambda = 1.5$ W K^{-1} m^{-1}, and $\rho = 1.0 \times 10^{-5}$ Ω m [12]. The copper materials are used for the plates of parallel-plate heat exchanger, and the metallic strips of thermoelectric module. The thickness of the plates and the metallic strips are 1.0×10^{-3} and 1.0×10^{-3} m, respectively. The ceramic plates are made of Al_2O_3 materials, and the thickness of the ceramic plate is 7.0×10^{-4} m. The thermophysical properties of these materials are determined using the inlet temperatures of working fluids, and are considered as constant during whole heat transfer process.

The inlet temperature and the flow rate of cold fluid are kept at a constant, but the inlet temperature and the flow rate of hot fluid are selected as variable parameter for different analytical cases. Due to selecting water as the working fluid in current simulation examples, the inlet temperature of cold water could be selected at the normal atmospheric pressure. For the inlet temperature of hot water, it is selected over 100 °C by assuming the corresponding pressure over the normal atmospheric pressure in order to facilitate the simulation, although selecting hot water/glycol mixture fluids could be more appropriate for the temperature over 100 °C in practical applications. In current simulation examples, the relevant thermophysical property of hot water values are evaluated at the corresponding water temperature and pressure. The above correlation on the heat transfer coefficient in the model could be still chosen because the Reynolds number and Prandtl number are within the extent of its validation.

In addition, the load resistance is selected to equal the effective internal resistance of the thermoelectric modules so that the maximum power output of a thermoelectric module could be achieved.

The simulation was carried out for both parallel flow type and counter flow type of exchanger at the same geometrical dimension of exchanger and thermophysical values of Bi_2Te_3 semiconductors. The following results are obtained when the inlet temperature of cold fluid is $t_{fc(1)} = 32\,°C$ and the flow rate is kept at a constant. The inlet temperature and the flow rate of hot fluid are shown in relevant figures for different analytical cases.

Fig. 2 shows the temperature variation of fluid flow for parallel flow type and counter flow type. As it can be seen in Fig. 2, the temperature variation tendency of hot and cold fluid flow along the axis of a thermoelectric generator in this model is similar to that in the ordinary heat exchanger. Nevertheless, the variations in temperature are almost linear case, which is different from logarithmic variation occurred in the ordinary heat exchanger. This special feature in the parallel-plate heat exchanger of thermoelectric power generation is due to Seebeck effect in the thermoelectric modules. The Seebeck effect make certain amount of energy from high-temperature fluid be taken away as electricity, and thus the heat flow from the high-temperature fluid is not equal to the heat flow to the low-temperature fluid. This could lead to the linear variation in the temperature of fluids.

Changes in fluid temperature would result in corresponding changes in temperature difference ΔT_{hc} between hot side and cold side of thermoelectric modules. Fig. 3 shows the variation of the temperature difference of thermoelectric modules for parallel flow type and counter flow type. As shown in Fig. 3, for a given inlet temperature of the fluids, there is a significant variation in temperature difference ΔT_{hc} for parallel flow type, but relatively smaller variation for counter flow type. However, the average temperature difference in parallel flow type is basically equal to the average temperature difference in counter flow type. This also means the maximum power output follows the same trend to this two flow types.

In general, the power output and the conversion efficiency depend mainly on the inlet temperature of hot fluid for a given

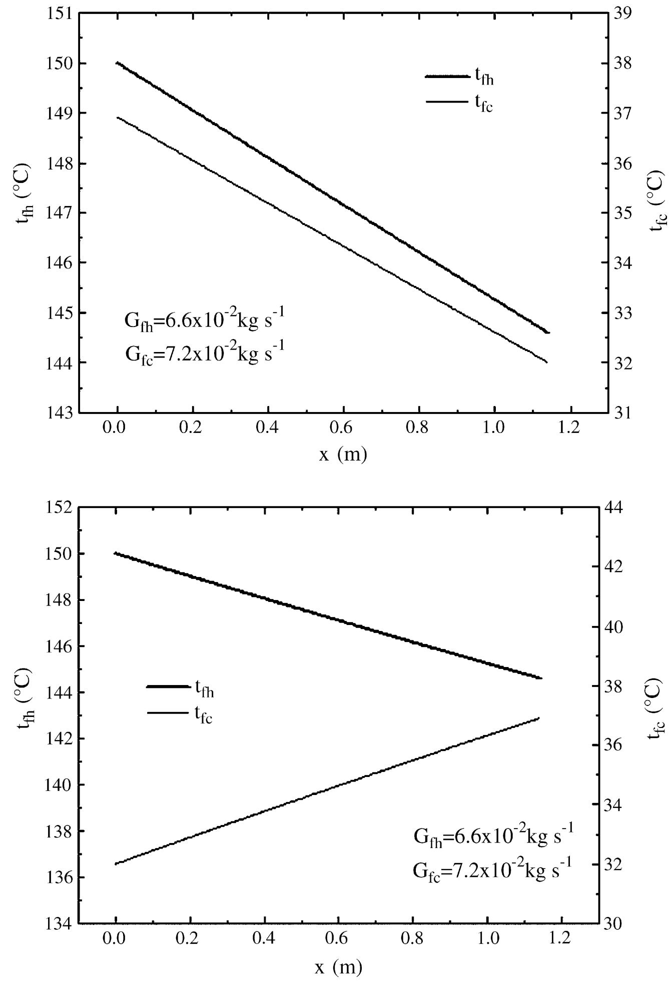

Fig. 2. The temperature variation of hot-fluid flow for two types of parallel-plate heat exchanger: (a) counter flow type and (b) parallel flow type.

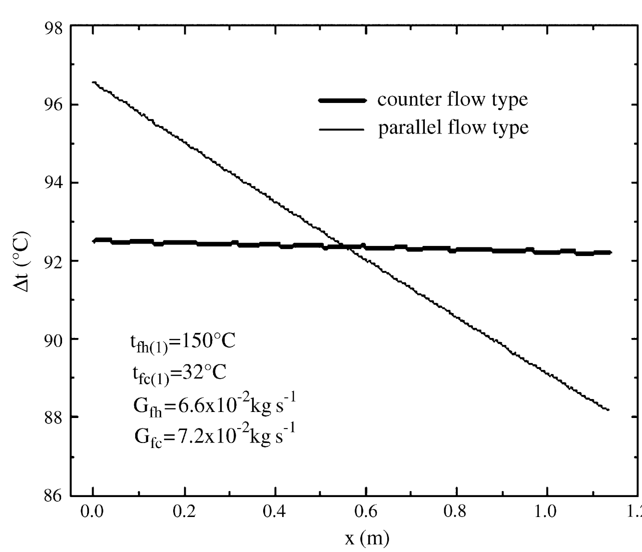

Fig. 3. The comparison of temperature difference of thermoelectric module for two types.

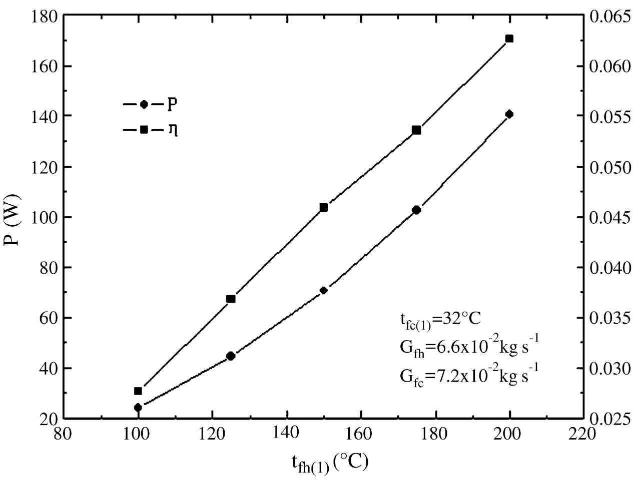

Fig. 4. The variation of the power output and the conversion efficiency with the inlet temperature of hot fluid for counter flow type.

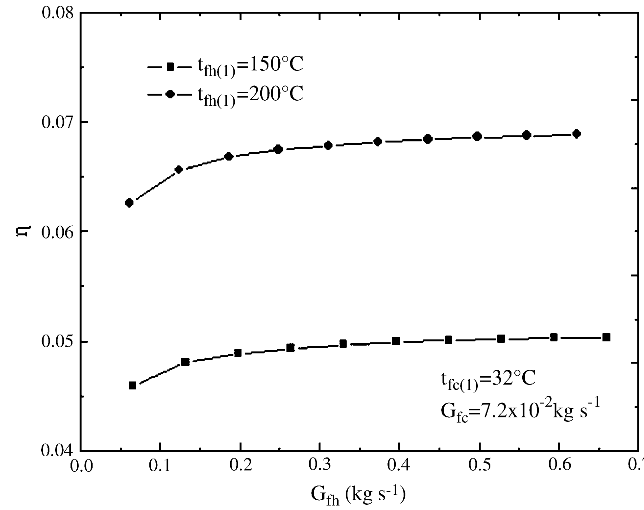

Fig. 6. The variation of conversion efficiency with hot flow rate.

thermoelectric generator. Fig. 4 illustrates the variation of the power output and the conversion efficiency with the inlet temperature of hot fluid for counter flow type when cold fluid temperature is 32 °C. It can be observed that, the power output and the conversion efficiency increases with increasing the inlet temperature of hot fluid for a fixed flow rate. This result from that the increases of inlet temperature lead to the increase of average temperature difference ΔT_{hc} through the modules, and thus the power output and the efficiency can be increased. By the calculation, the efficiency of the model can reach 6.26% when the inlet temperature of hot fluid is $t_{fh(1)}$ = 200 °C.

It is known the convective heat transfer coefficient is changed when varying the fluid flow rate. This change also brings a change in thermal resistances of heat exchanger, results in an increase of the average temperature difference ΔT_{hc} through the modules. Therefore, an increase of the power output and the efficiency can be obtained.

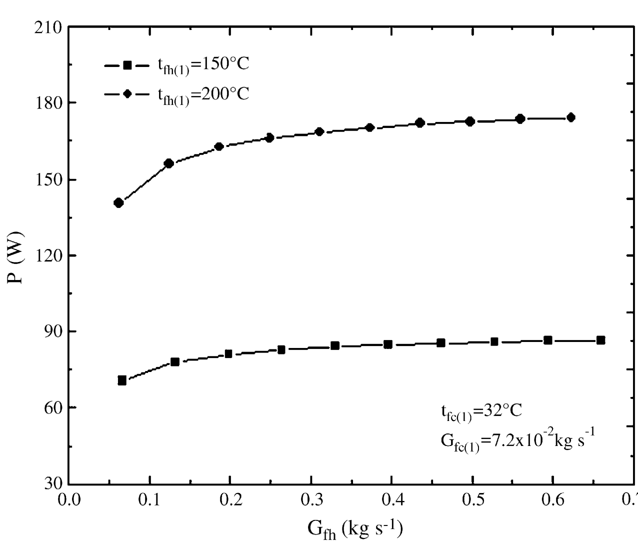

Fig. 5. The variation of power output with hot flow rate.

Fig. 5 shows the variation of power output with the hot flow rate. As can be seen from Fig. 5, for a given inlet temperatures, the power output increases with increasing the flow rate. It is also found that the increase of the power output is significant at the beginning, but the increase trend becomes smaller as the flow rate further increases, especially at lower inlet temperature case. Fig. 6 further displays the effects of the hot flow rate on the conversion efficiency of thermoelectric generator. It is also seen that as the flow rate increases, the conversion efficiency becomes larger.

Obviously, the fluid pressure drop through the passage of parallel-plate heat exchanger also increases when the flow rate is increased. This causes an increase of the pump power for driving the fluid through heat exchanger. When the pump consumption is considered, the net power output of thermoelectric generator could be the total power output minus the pump power. Therefore, the appropriate the flow rate should be determined to meet the optimal matching of operating conditions for real thermoelectric generators.

4. Conclusion

A numerical model for thermoelectric generator with the parallel-plate heat exchanger was developed based on one-dimensional differential equations representing energy conservation of heat exchanger. These equations are then restructured and linked to the formulations of thermoelectric modules. This elemental approach gives a much more detailed prediction for the fluid temperature changes and the temperature difference through thermoelectric modules along the channels. The simulation results revealed the linear variations in temperature of the fluids in the thermoelectric generator, which is different from logarithmic variation case in the ordinary parallel-plate heat exchanger. It is apparent that the numerical model developed in this study may be also applied to further optimization study for thermoelectric generator. This numerical model has incorporated the thermocouples with both the contact layers (ceramic plates and the metallic strips) and the systems level heat exchanger. Therefore, the effective temperature difference between the hot and the cold junctions can be estimated, and the optimization of both the heat exchanger geometry as well as the thermoelectric module geometry for optimal waste-heat recovery performance can be simultaneously performed. In addition, this numerical model also provides a basis for further analysis on net power output as the fluid side pressure across the heat exchanger and associated pumping work can be calculated by using the above Fanning friction factor.

Further work is needed to validate the accuracy of the model by corresponding experiment, and thus the next step is to construct and operate a scaled-up thermoelectric generator with the parallel-plate channel.

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