1   **Performance analysis on a solar concentrating thermoelectric generator using**
2   **the micro-channel heat pipe array**

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14   **Abstract:**

15   Micro-channel heat pipe can convert the low heat flux to the high heat flux by
16   changing the ratio of the evaporator area to the condenser area and has a higher heat
17   transfer performance than the common heat pipe. Combining the solar concentrating
18   thermoelectric generation with micro-channel heat pipe can save the quantity of
19   thermoelectric generation and reduce the cost significantly. In this paper, a solar
20   concentrating thermoelectric generator using the micro-channel heat pipe array was
21   designed, and the mathematical model was built. Furthermore, the comparison of the
22   experiment and the simulation between the solar concentrating thermoelectric

generator using the micro-channel heat pipe array and the thermoelectric generations in series was made. In addition, the performance on the different areas of selective absorbing coating, different concentration ratios, different ambient temperatures, different wind speed all were analyzed. The outcomes showed the overall performance of the solar concentrating thermoelectric generator using the micro-channel heat pipe array system.

Keywords: *thermoelectric generation, Micro-channel heat pipe, mathematical model, strip-focus Fresnel lens*

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Nomenclature

A	Area of selective absorbing coating	R_{ct1}	Thermal contact resistance between selective absorbing coating and heat pipe KW^{-1}
a_{teg}	Cross-sectional area of a P or N leg m^2	R_{ct2}	Thermal contact resistance between heat pipe and the TEG KW^{-1}
C	Concentration ratio	R_{ct3}	Thermal contact resistance between TEG and heat sink KW^{-1}
C_{fx}	Local friction coefficient	R_{fin}	Thermal resistance of the heat sink KW^{-1}
C_p	Specific heat of air $\text{kJ}/(\text{kgK})$	r_{teg}	Electrical resistivity of P or N junction $\Omega \cdot \text{m}$
E_{con}	Heat flux via conduction W	Re	Reynolds number
E_{cov}	Heat flux via convection W	St	Stanton number
E_{in}	Solar energy absorbed by selective absorbing coating W	S_1	Cross section area of the heat sink m^2
E_{rad}	Heat flux via radiation W	S_2	Total area of the heat sink m^2

G	Solar radiation W/m^2	T_a	Temperature of ambient air K
h_{cov}	Coefficient of convection heat transfer $\text{W}/(\text{m}^2\text{K})$	T_h	Temperature of the evaporating side of the heat pipe K
h_{rad}	Coefficient of radiation heat transfer $\text{W}/(\text{m}^2\text{K})$	T_{tegl}	The temperature of cold side of TEG K
I	Current A	T_{tegh}	Temperature of the hot side of TEG K
K	Thermal conductivity of heat pipe $\text{Wm}^{-1}\text{K}^{-1}$	T_1	Temperature of the condensate side of the heat pipe K
k_{air}	Thermal conductivity of air $\text{Wm}^{-1}\text{K}^{-1}$	T_p	Temperature of absorbing coating K
k_{teg}	Thermal conductivity of TEG $\text{Wm}^{-1}\text{K}^{-1}$	T_{sky}	Temperature of sky K
L	Length of the heat sink m	u	Wind speed m/s
L_{hp}	Length of heat pipe m	Greek Letters	
l_{teg}	Height of TEG m	α	Heat diffusivity m^2/s
n_{teg}	Numbers of PN junction	α_b	Absorptivity of absorbing coating
Nu	Nuselt number	α_{teg}	Seebeck coefficient VK^{-1}
Pr	Prandtl number	δ	Stefan-Boltzmann constant $\text{Wm}^{-2}\cdot\text{K}^{-4}$
Q_{tegh}	Energy that passed in hot side of the TEG W	ε	Reflectivity of absorbing coating
Q_{tegl}	Energy that passed out of cold side of the TEG W	μ	Viscosity (dynamic) ² , Ns/m^2
R_{conf}	Resistance of the heat conduction in heat sink KW^{-1}	ν	Kinematic viscosity, m^2/s
R_{covf}	Thermal resistance of convection Heat transfer between heat sink and ambient air KW^{-1}	ρ	Density of air kg/m^3

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36 1. Introduction

37 With ever-increasing consumption of fossil energy and growing concern on
38 environment problem, there is a primary topic on how to effectively utilize green
39 energy sources and recover waste heat [1]. Compared to conventional electrical power

generator systems, thermoelectric generators (TEG) theoretically offer many advantages such as being simple and highly reliable, having no moving parts, and being environmentally friendly [2]. Thus it is a good choice to utilize the Seebeck effect to convert heat from solar energy directly into electrical energy through the movement of charge carriers induced by a temperature span across the TEG.

Ali Ekber Özdemir et al presented an experimental design of a new solar based thermoelectric generator with wind chimney, which mainly consists of four parts: a heat pipe with solar collector tube for solar heating, a wind chimney for cooling [3].

Wei He et al described a solar heat pipe thermoelectric generator (SHP-TEG) unit comprising an evacuated double-skin glass tube, a finned heat pipe and a TEG module [4]. Bin-Juine Huang et al studied a thermoelectric generator using loop heat pipe [5].

Ashwin Date et al used the heat pipe to cool thermoelectric generators for the water heating [6]. Wei-Hsin Chen et al made the modeling and simulation for the design of thermal-concentrated solar thermoelectric generator [7]. Li et al. designed a prototype optical-concentrated solar TEG and used a numerical method to evaluate its performance [8]. Xiao et al. analyzed the performance of a multi-stage solar TEG by means of a numerical method [9]. Peng et al. performed the detailed experimental and theoretical analysis of solar concentration system using TEG, with specific recommendations about the necessary concentration degree and the different materials for TEG [10].

E.A. Cha'vez-Urbiola et al investigated of performance of several hybrid systems producing electric and thermal energy with application of TEGs and working

in non-concentrated as well as concentrated solar radiation [11]. Ying-Ying Wu et al studied the effect of the glass cover on the photovoltaic–thermoelectric hybrid system [12]. Tianjun Liao et al analyzed the performance characteristics of a photovoltaic-thermoelectric hybrid power generation device with a low concentrator [13]. Xing Ju et al made the numerical analysis and optimization of a spectrum splitting concentration photovoltaic–thermoelectric hybrid system [14]. D.N. Kossyvakis et al also carried on the computational analysis and performance optimization of a solar thermoelectric generator [15]. M. Hasan Nia et al investigated an electricity and preheated water cogeneration solar system by thermoelectric [16].

However, a mass of TEGs in solar system appears to be economically unfeasible, owing to the significantly higher cost and lower power output of the TE module [17]. To tackle this disadvantage, a heat pipe was introduced to combine the solar system. In fact, converting low heat flux to high heat flux is the one of main functions of a heat-pipe by changing the ratio of evaporator area to condenser area at an almost constant temperature, which can save the amount of the TEG modules. The common heat pipe solar TEG systems have been studied [2,3,4], But the flat-plate micro-channel heat pipe (MCHP) has a higher heat transfer performance than the common heat pipe, and can be closer to the surface of the TEG, so the MCHP would have a larger advantage than the common heat pipe. In addition, comparison with other solar TEGs without MCHP, the STEG-MCHP system can save the number of TEGs and reduce the cost significantly. Therefore, in this paper, we present a solar concentrating TEG with MCHP system (STEG-MCHP). Considering the fact that

increasing the electricity output of TEG can lead to the cost reduction, a strip-focus Fresnel lens will be combined with the system.

2. Mathematical model

2.1 System description

Solar selective absorbing coating is attached to the upper surface of the evaporator of the MCHP while the TE module is attached to the lower surface of the condenser (Fig.1). During operation, the solar heat imposed on the upper surface of the MCHP's evaporator is conveyed to its condenser, by means of the evaporation of the working fluid within the MCHP. In the condenser, the heat is released via condensation of the MCHP working fluid and is then transferred to the attached TE modules. This creates a temperature gradient across the TE module, thus resulting in a thermal-to-electrical conversion by means of the 'Peltier effect'. The cooling structure of TEG is air cooling, whose performance is decided by the cooling structure, ambient temperature and wind speed.

Fig.1 Schematic diagram of the STEG-MCHP system

This study is intended to analyze the steady state performance of the STEG-MCHP system, and the following assumptions are adopted to simplify the problem.

(1) All the energy balance equations are for the steady state conditions, so the thermal capacity of any parts of the STEG-MCH system is neglected.

(2) The rest section of MCHP is covered by the insulating materials, so the heat loss of this part is neglected.

(3) The square of external electrical resistance is assumed to equal the product of internal electrical resistance and external electrical resistance, i.e., $R_{load}^2 = R_{te} R_{load}$.

2.2 Model

The heat transfer process of STEG-MCHP system is shown in Fig.2. According to heat balance, the energy conducted by the heat pipe can be expressed as below,

$$E_{in} = E_{con} + E_{rad} + E_{cov} \quad (1)$$

Where E_{in} is the solar energy absorbed by the selective absorbing coating. It can be expressed as below,

$$E_{in} = \alpha_b C A G \quad (2)$$

Where α_b , C and A are the absorptivity, concentration ratio, area of the selective absorbing coating and respectively. G is the solar radiation.

Fig.2 Schematic of the heat transfer network of the STEG-MCHP

E_{rad} is the energy transferred to the space via the radiation, which is expressed

as below,

$$E_{rad} = h_{rad} A (T_p - T_{sky}) \quad (3)$$

h_{rad} can be defined as

$$h_{rad} = \epsilon \sigma (T_p^2 + T_{sky}^2) (T_p + T_{sky}) \quad (4)$$

T_{sky} can be defined as

$$T_{sky} = 0.0552 T_a^{1.5} \quad (5)$$

T_p is temperature of the selective absorbing coating and T_a is the temperature of ambient air.

E_{cov} is the energy convection between the selective absorbing coating and the ambient air, which is expressed as

$$E_{cov} = h_{cov} A (T_p - T_a) \quad (6)$$

Where h_{cov} can be defined as [18]

$$h_{cov} = 6.5 + 3.3u \quad (7)$$

The energy conducted via the heat pipe can also be express in several ways. It can be the heat flux pass through the thermal contact resistance between selective absorbing coating and heat pipe,

$$E_{con} = (T_p - T_h) / R_{ct1} \quad (8)$$

where T_h is the temperature of the evaporating side of the heat pipe.

It can also be heat flux conducted by the heat pipe, which can be expressed as,

$$E_{con} = KL_{hp}(T_h - T_l) \quad (9)$$

where K is the thermal conductivity of the heat pipe, L_{hp} is the length of the heat pipe and T_l is the temperature of the condensate side of the heat pipe.

The heat flux pass through the thermal contact resistance between heat pipe and the TEG (R_{ct2}) can be defined as,

$$E_{con} = (T_l - T_{tegh}) / R_{ct2} \quad (10)$$

where T_{tegh} is the temperature of the hot side of the TEG.

The energy that passed through the hot side of the TEG can be defined by following equation [19].

$$Q_{tegh} = 2n_{teg} \alpha_{teg} T_{tegh} I + 2n_{teg} \frac{a_{teg} k_{teg}}{l_{teg}} (T_{tegh} - T_{tegl}) - \frac{1}{2} I^2 2n_{teg} \frac{r_{teg} l_{teg}}{a_{teg}} \quad (11)$$

Similarly, the heat through the cold side of TEG can be expressed as

$$Q_{tegl} = 2n_{teg} \alpha_{teg} T_{tegl} I + 2n_{teg} \frac{a_{teg} k_{teg}}{l_{teg}} (T_{tegh} - T_{tegl}) + \frac{1}{2} I^2 2n_{teg} \frac{r_{teg} l_{teg}}{a_{teg}} \quad (12)$$

where Q_{tegh} is the energy that passed in hot side of the TEG, Q_{tegl} is the energy that passed in cold side of the TEG, n_{teg} is the numbers of PN junction, α_{teg} is the Seebeck coefficient, I is the current, K_{teg} is thermal conductivity of TEG, l_{teg} is the length of TEG, T_{tegl} is the temperature of cold side of TEG, r_{teg} is the electrical resistivity of a P or N leg, a_{teg} is the Cross-sectional area of a P or N leg.

The electrical efficiency of the TEG can be expresses

$$\eta_{teg} = \frac{Q_{tegh} - Q_{tegl}}{Q_{tegh}} \quad (13)$$

The electrical efficiency of the system can be expresses

$$\eta_{sys} = \frac{Q_{tegh} - Q_{tegl}}{CAG} \quad (14)$$

This heat can also be expressed as the heat flux through the heat sink attached to the cold side of TEG.

$$Q_{tegl} = (T_{tegl} - T_a) / R_{fin} \quad (15)$$

Where R_{fin} is the thermal resistance of the fin which clings to the cold side of TEG, and can be expressed as below,

$$R_{fin} = R_{conf} + R_{covf} + R_{ct3} \quad (16)$$

Where R_{conf} is the resistance of the heat conduction in the heat sink and R_{ct3} is the thermal contact resistance between the TEG and the heat sink.

$$R_{conf} = \frac{H}{K_{fin} S_1} \quad (17)$$

Where H is the height of the heat sink, K_{fin} is the thermal conductivity of the fin and S_1 is the cross section area of the heat sink.

In order to attain the value of R_{conf} which is the thermal resistance of convection heat transfer between fin and ambient air, the convection heat transfer coefficient h_{covf} must obtain [20].

$$C_{fx} = 0.0592 \text{Re}_x^{-1/5} \quad (18)$$

$$\text{Re}_x = \frac{\rho u x}{\mu} \quad (19)$$

where u , ρ , μ represent velocity, density and viscosity coefficient of the fluid

respectively, x is the characteristic length of the fin.

$$Nu_x = \frac{h_x x}{k_{air}} = 0.332 Re_x^{\frac{1}{2}} Pr^{\frac{1}{3}}, 0.6 \leq Pr \leq 15 \quad (20)$$

$$Pr = \frac{\nu}{\alpha} = \frac{\mu C_p}{k_{air}} \quad (21)$$

Where ν is the kinematic viscosity, h_x is the local heat transfer coefficient along the length of the fin, k_{air} is the thermal conductivity of air, C_p is the specific heat capacity of air, α is the heat diffusivity.

Thus St_x can be obtained as below,

$$St_x = \frac{Nu_x}{Re_x Pr} = \frac{h_x}{\rho C_p u_{\infty}} = 0.332 Re_x^{-1/2} Pr^{-2/3} \quad (22)$$

$$St_x Pr^{2/3} = C_{f,x}/2 \quad (23)$$

$$h_x = \frac{C_{f,x} \rho C_p u_{\infty}}{2 Pr^{2/3}} \quad (24)$$

And the local heat transfer coefficient h_x is obtained,

$$h_x = \frac{0.0296 x^{-1/5} \rho C_p u_{\infty}^{4/5}}{\nu^{-1/5} \cdot Pr^{2/3}} \quad (25)$$

Therefore,

$$h_{covf} = \frac{1}{L} \int_0^L h_x dx \quad (26)$$

After finishing calculated tablet on the air, the average heat transfer coefficient can be expressed as:

$$h_{covf} = \frac{0.037 L^{-1/5} \rho C_p u_{\infty}^{4/5}}{\nu^{-1/5} \cdot Pr^{2/3}} \quad (27)$$

Then the thermal resistance between the radiator and the air is:

$$R_{covf} = \frac{1}{h_{covf} S_2} = \frac{\nu^{-1/5} \cdot Pr^{2/3}}{0.037 S L^{-1/5} \rho C_p u_{\infty}^{4/5} S_2} \quad (28)$$

Where S_2 is the total area of the fin.

3. Preliminary experiment validation and comparison

3.1 Experiment condition

In order to analyze the performance of the STEG-MCHP system, the preliminary experiment was made to verify the mathematical model. The main experiment and test elements were shown in Fig. 3. The MCHP array is a flat aluminum plate with multiple parallel micro-channel heat pipes. The thickness of MCHP is 3.0 mm and each micro-channel heat pipe has many inner microgrooves (or micro-fins) to enhance heat transfer by repeated evaporation and condensation of inner working fluid . The solar liner Fresnel concentrator of 4X was chose to attain higher solar radiation. The TEG module was fabricated by ShenMingYang Electronics Ltd. (product code: TEG1-241-1.4-1.2). The size of the TEG was $55\times55\times3.4$ mm³. In addition, the comparison between the STEG-MHCP and two TEGs in series was made. The two systems had the same areas of the selective absorbing coating, which equals to 2 times top surface area of one TEG. The relative parameters were shown in table 1.

Fig.3 Photo of the main experiment and test elements

Table 1 Relative parameters in the experiment

Parameter	Numerical
Environment	
Ambient temperature T_a	301.15 K
Solar radiation G	600 W
Concentration ratio C	4
Ambient wind speed u	0.5 m / s
Wind speed around heat sink u_{wind}	5 m / s
MCHP	
Thermal conductivity of heat pipe K	23000 W / (m·K)
Length of the heat pipe L_{hp}	0.75 m
Thermal contact resistance between TEG and heat sink R_{ct3}	$1 \times 10^{-4} \Omega$
Selective absorbing coating	
Absorptivity of the absorbing coating α_b	0.93
Area of selective absorbing coating A	0.00605 m ²
Reflectivity of the absorbing coating ε	0.8
Thermal contact resistance between selective absorbing coating and heat pipe R_{ct1}	$4 \times 10^{-4} \Omega$
Thermal contact resistance between heat pipe and the TEG R_{ct2}	$1 \times 10^{-4} \Omega$
TEG	
Numbers of P or N junction n_{teg}	241

Cross-section area of one P or N junction	a_{teg}	$1 \times 10^{-8} m^2$
Seebeck coefficient	α_{teg}	$1.35 \times 10^{-4} V / K$
Electrical resistivity of one P or N junction	r_{teg}	$3.35 \times 10^{-8} \Omega / m$
Height of TEG	l_{teg}	$3.4 \times 10^{-3} m$

Heat sink

Height of the heat sink	H	$1 \times 10^{-2} m$
Length of the heat sink	L	$5 \times 10^{-2} m$
Cross section area of the heat sink	S_1	$1.2 \times 10^{-4} m^2$
Total area of the fins	S_2	$1.46 \times 10^{-2} m^2$

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218 To measure their temperature, thermocouples were attached to the different

219 surfaces, which can be obtained from the data logger. The output of the TEG module

220 was connected to the electronic load which has an adjustable resistance. The current

221 and the voltage of the load resistance were collected by the ammeter and voltmeter.

222 The uncertainty of the thermocouples, the ammeter and the voltmeter are listed in the

223 Table 2.

224 Table 2 Uncertainty of the equipment

Test Equipment	Uncertainty
Voltmeter	$\pm 0.5\%$
Ammeter	$\pm 0.8\%$
Thermocouple	0.2–0.5 K

225

226 3.2 Simulation results and the experimental verification

From the Fig.4, it can be seen that the maximum temperature difference value between the simulation and experiment was within 2 °C and the temperature difference between the MCHP's evaporator and condenser was within 1.5 °C . Therefore, the simulation results fitted well with experiment values. Besides of this, it can also indicate that the temperature difference of the MCHP's evaporator and condenser is small.

Fig.4 Temperature comparison between experiment and simulation

It is clearly from Fig.5 that the output power of the STEG-MCHP system was higher than that of two TEGs in series, which means the STEG-MCHP system maybe have a higher output with less TEG module and lower cost.

Fig.5 Comparison of output power between STEG-MCHP and two TEGs in series

4. System analysis and discussion

4.1 Different areas of selective absorbing coating

In order to keep the same areas of the selective absorbing coating of the STEG-MCHP system and the TEGs in series directly, the area of selective absorbing coating should be the integer multiples for one TEG module. The ambient wind speed equals to that around the heat sink. The solar radiation and the concentration ratio were set to 1000W/m^2 and 4X. The ambient temperature is $25\text{ }^{\circ}\text{C}$.

It can be seen from Fig.6 in this ambient condition, when the area of the selective absorbing coating is larger than $2 \times 0.055 \times 0.055\text{ m}^2$, the STEG-MCHP has the higher temperature difference, which also presents the advantage of MCHP that concentrating heat to the condenser.

Fig.6 Temperature difference with different areas of selective absorbing coating

Form the Fig.7, it is clearly that the STEG-MCHP system has a higher output power when the number of TEG is larger than 2, and the reason of this is that the STEG-MCHP has the larger temperature differences, which also responses to the higher electrical efficiency. But it can also be seen that the increase tendency on output power of STEG-MCHP becomes alleviated.

Fig.7 Output power with different areas of selective absorbing coating

4.2 Different concentration ratio

It can be seen from Fig.8 that with the increase of the concentration ratio, the electrical efficiencies of the STEG-MCHP and the TEGs in series can all be improved. However, on one specific concentration ratio, the electrical efficiency of the STEG-MCHP shows the firstly increase then decrease tendency, which is different from the TEGs in series with the nearly constant. Furthermore, when the selective absorbing coating area is about $5 \times 0.055 \times 0.055 \text{ m}^2$, the electrical efficiency may get a peak value, which is because with the increase of the selective absorbing coating area, the temperature of the MCHP would be increased, which led to a larger thermal losses.

In addition, it can be predicted that if not to consider the temperature limit of the MCHP, or if the HCMF can operate in the middle temperature, the electrical efficiency of MCHP would have a lower value than TEGs in series. Therefore, the STEG-MCHP has the advantage than TEGs in series, but the advantage would exist in a certain range. Besides of these, it also can be seen that the electrical efficiencies in Fig.8 are all low, which is because the largest temperature difference between the hot side and cold side of TEG is below $30 \text{ }^{\circ}\text{C}$.

Fig.8 Electrical efficiency with series of concentration ratios

4.3 Different ambient temperatures

Different ambient temperatures can affect the output performance, so different ambient temperatures such as 0 °C, 25 °C, 35 °C were employed to analyze the electrical power with 3X concentration ratio (Fig.9).

Fig.9 Hot side and cold side temperature with different ambient temperature

It is clearly that with the higher ambient temperature, the hot side and cold side temperatures of TEG would all be higher. For the selective absorbing coating, the losses will be reduced, but for the TEG, the high temperature of cold side will lead to a negative impact on the output power.

For the STEG-MCHP system, the effect of the ambient temperature is significant, and the difference of the output power among the different ambient temperatures became obvious with the area increase of the selective absorbing coating (Fig.10). However, for the TEGs in series, the differences of the output power among the different ambient temperatures are relatively small. However, it must be mentioned

that the differences on output power between the STEG-MCHP and TEGs in series firstly became larger then smaller.

Fig.10 Output power with different ambient temperatures

4.4 Different wind speeds

The ambient temperature is set as 25 °C, and the wind speed is set as 1, 2, 3, 4, 5, 6, 7, 8 m/s respectively.

The temperature difference with 3X concentration ratio is shown in Fig.11. For STEG-MCHP system, when wind speed is 1 m/s, the temperature difference between the hot side and cold side of TEG is lowest, which means the electrical efficiency of the TEG is lowest (Fig.11 (a)). However, the maximum temperature difference appears when the wind speed is 4 m/s, so there is a coupling relationship between the

wind speed and the area of the selective absorbing coating.

For the TEGs in series, the maximum temperature difference appears when wind speed is 3 m/s (Fig.11 (b)).The large wind speed can lead a large temperature difference, but it can also cause a large amount of heat losses.

(a)

(b)

Fig. 11 Temperature difference with different wind speeds

With the increase of the wind speed, the heat losses and the efficiency of the TEG all will be improved. At the same time, the temperature of the MCHP will also be reduced, which will have a higher concentration ratio and will also not reach the temperature limit of the MCHP. Fig.12 shows the TEG efficiency with the wind speed of 8 m/s.

Fig.12 Electrical efficiency of TEG with the wind speed of 8m/s

It can be seen that with the wind speed of 8m/s, the electrical efficiency of TEG is improved significantly. In addition, with larger concentration ratio, the electrical efficiency has a larger value. For example, with the concentration ratio of 8X, the electrical efficiency of TEG has about 0.82% based on the temperature difference of 59.6 °C. However, for the whole system, the electrical efficiency is different this. Taking the STEG-MCHP system with concentration ratio of 8X for example, the system has a largest efficiency with the area of selective absorbing coating of $5 \times 0.055 \times 0.055 \text{ m}^2$, which is because the larger area can also lead to larger heat losses.

5. Conclusion

The flat-plate micro-channel heat pipe (MCHP) has a higher heat transfer performance than the common heat pipe, and can be closer to the surface of the TEG,

so the MCHP would have a larger advantage than the common heat pipe. Therefore, in this paper, we present a solar concentrating TEG with MCHP system (STEG-MCHP).

The mathematical model of the STEG-MCHP was built, and the simulation and experiment was compared. The outcome indicated that the STEG-MCHP has a higher output power than the TEGs in series with the same area of the selective absorbing coating ($2 \times 0.055 \times 0.055 \text{ m}^2$).

In addition, the performance of the STEG-MCHP system was analyzed based on the different areas of the selective absorbing coating, concentration ratios, ambient temperatures and wind speeds. The temperature of the hot and cold side of TEG, temperature difference, output power, TEG efficiency, system electrical efficiency were all demonstrated. The results indicated the effects of series of factors on the performance of STEG-MCHP system, and showed the variation tendency of the performance.

The advantage of STEG-MCHP is decided by many factors, such as the type of MCHP and TEG, the structure of the inner channel and the length of the MCHP, the heat loss of the system and the temperature difference of the TEG etc. Therefore, the overall performance improvement will be based on the comprehensive effect of many factors optimization together.

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