Calculation and Performance Analysis of a Solar/Gas Receiver for a Stirling Power Generation System in Gas-Only Mode

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Keywords: Solar/gas heat receiver, Zonal model, Radiative heat transfer, Stirling engine, Thermal model.

Abstract. A thermal model for solar/gas Stirling power generation system was developed to achieve characteristics of the system in thermal balance. The thermal model was developed by building math models for a heat receiver, an air/gas heat exchanger, a Stirling engine and so on and coupling them. The effects of biogas flow rate, air flow rate, and CH4 concentration of biogas on the heat transfer characteristics of the solar/gas heat receiver (SGHR) was analyzed selectively in this paper. The calculation result shows that, with biogas mass flow rate increasing by 25%, the heat transfer power of the SGHR increases by 19.97% and the heat transfer efficiency of the SGHR decreases by 4.44%. With air mass flow rate increasing by 30%, the heat transfer power and efficiency of the SGHR both decreases by 13.84%. With CH4 concentration of biogas increasing by 18.18%, the heat transfer power of the SGHR increases by 27.14% and the heat transfer efficiency of the SGHR decreases by 2.96%.

Introduction

Stirling engine is an external heating (or burning) piston engine, usually with helium as the working medium, working in a closed regeneration cycle[1]. At present, dish/Stirling solar power generation systems have matured, but its power generation capacity is affected by the inherent defect of solar energy, as a result, its power output is discontinuous and unstable. There are two main solutions to this problem: the first solution is to configure the energy storage subsystem in the system to store the collected solar heat energy or converted electrical energy to ensure that it is used at night or when solar radiation is insufficient; the second solution is to build a refuel system that provides heat to the system in the absence of solar energy.

For example, in 1996, the NASA Lewis Research Center built the first ground-based prototype of molten salt solar thermal power generation system worldwide[2]; Hamidreza Shabgard [3] developed a two-dimensional numerical model to simulate the transient response of a heat pipe-assisted latent heat thermal energy storage unit integrated with dish/Stirling solar power generation systems. Duarte Laing developed a hybrid sodium heat pipe receiver[4-6]. It is necessary to build an energy storage device that is large enough to cope with a long period of unlit or insufficient sunshine weather when the energy storage subsystem is configured in the solar power generation system. The entire power generation system will cover more area. The advantage of distributed energy vanishes as well. During the experiment, various little cracks developed on the surface of the sodium heat pipe of hybrid sodium heat pipe receiver which is designed by Duarte Laing and other people. By this, it is decided to restrict the helium temperature to 600 °C in the Stirling engine, which caused the experiment efficiency to stay below the expected values.

For solving the above problems, a new SGHR was designed, which can directly utilize the solar radiation and the heat energy generated by gas combustion. This paper presents the performance of the SGHR in the solar/gas Stirling power generation system in gas-only mode. A combustion radiation model was developed to calculate radiative heat transfer in combustor and a thermodynamic model was developed to analyze the Stirling engine. The air/gas heat exchanger was simulated by CFD method. The performance calculation and analysis of the system was done by coupling components.
Solar/gas Stirling Power Generation System

The solar/gas Stirling power generation system mainly includes: SGHR, Stirling engine. The energy conversion process is: biomass energy → thermal energy → mechanical energy → electric energy. The corresponding mathematical model is established for each component of the solar/gas Stirling power generation system.

Solar/Gas Heat Receiver

A schematic drawing of the SGHR structure is shown in Figure 1. The SGHR is a combination of solar energy receiver of the solar Stirling power generation system and combustor of gas Stirling power generation system. It can accept solar radiant energy and can also accept combustion heat. According to the structural characteristics and heat transfer process of the SGHR, it is divided into two parts: the first is the external air/gas heat exchanger which can heat air by using exhaust gas, to improve thermal efficiency of the system. The second is the receiver cavity which is used as a combustor or solar energy collection cavity.

Air/Gas Heat Exchanger. The air/gas heat exchanger is shown in Fig 1 (b). The air/gas heat exchanger is an axially symmetric ring recuperator which is composed of 20 identical units along the circumferential direction.

Combustor/Solar Energy Collection Cavity (CSECC). The heat pipes of Stirling engine is shown in Figure 1(a). It includes fin-free heat pipes and finned heat pipes. The combustion of biogas and the radiative heat transfer and convective heat transfer between gas and the fin-free heat pipes carry out in the CSECC. Because of the temperature of gas dropping when the gas swept through the fin-free heat pipes, the radiative heat transfer is very less than the convective heat transfer between the gas and finned heat pipes. Therefore, the radiative heat transfer between the gas and the finned heat pipes is neglected. In this paper, the zonal model [7] is adopted to calculate the radiative heat transfer between the gas and the fin-free heat pipes, and the characteristic number equation is adopted to calculate the convective heat transfer between the gas and the heat pipes.

Stirling Engine System

A 4-cylinder double acting Stirling engine is selected in this paper. The three-order node method model [8], [9] is used to analyze the thermodynamic cycle of the Stirling engine. Detailed calculation procedure is shown in the literature [10].

Numerical Method

Numerical Simulation of Air/Gas Heat Exchanger
A unit of the air/gas heat exchanger was simulated by CFD method to obtain performance of the air/gas heat exchanger. The ICEM was adopted to generate hexahedral structure grid, and the grid number was about 1.81 million. A S-A turbulence model is selected in the simulation. Mass flow rate boundary conditions are adopted at air inlet and gas inlet. Pressure boundary conditions are adopted at air outlet and gas outlet.

**Radiative Heat Transfer in Combustion Chamber**

The combustor can be approximated as a frustum of cone cavity, and its geometrical model schematic drawing is shown in Figure 2. The flow of the combustor is simplified to one-dimensional flow. The combustor is divided into N units along the X direction and each unit is divided into the gas zone and the wall region. The energy balance equations of each region are established according to the energy conservation law. A closed equation group is obtained according to the energy equilibrium equation and the boundary condition, then the concrete expression of each parameter in the equation group can be obtained by the heat transfer analysis. The closed equation group is solved to get temperature distribution of the combustor and radiation heat transfer between gas and fin-free heat pipes.

Figure 1 (a) shows the combustor surrounded by a thick layer of insulating material, so the combustor wall heat conduction loss can be neglected. The sketch of heat balance for combustor cell is shown in Figure 3. The thermal equilibrium analysis of a unit in the combustor under steady state flow can be expressed as follows:

![Figure 2. Schematic drawing of combustor geometric model.](image-url)

![Figure 3. Sketch of heat balance for combustor cell.](image-url)

Gas energy balance equation:

\[ R_1 + R_2 - R_3 + Q_f - C_1 - \Delta H = 0 \]

Wall energy balance equation:

\[ R_4 + R_5 - R_6 + C_1 = 0 \]

where \( R_1 \) is the radiative heat transfer rate (kW) of the wall to the gas, \( R_2 \) is the radiative heat transfer rate (kW) of the gas to the gas, and \( R_3 \) is the radiant heat flow (kW) emitted by the gas itself. \( Q_f \) is the methane combustion heat (kW), \( C_1 \) is the convective heat loss of gas (kW), \( \Delta H \) is the gas enthalpy difference (kW) between outlet and inlet in a unit, \( R_4 \) is the radiative heat transfer rate (kW) of the wall to the wall, \( R_5 \) is the radiative heat transfer rate (kW) of the gas to the wall, \( R_6 \) is the radiant heat flow (kW) emitted by the wall itself.

The concrete expression of each parameter in Eq.1 and Eq.2:

1. Radiative heat transfer rate between units: R1, R2, R4 and R5
\[ R_i = \sum_i \bar{S}_i \bar{G}_i \sigma i^4 \]  
(3)

\[ R_2 = \sum_i \bar{G}_i \bar{G}_i \sigma i^4 \]  
(4)

\[ R_3 = \sum_i \bar{S}_i \bar{S}_i \sigma i^4 \]  
(5)

\[ R_5 = \sum_i \bar{G}_i \bar{G}_i \sigma (\frac{T_{G_i} + T_{S_i}}{2})^4 \]  
(6)

where \( S \) represents the wall, \( G \) represents the gas, subscript \( i, j \) is the unit number, \( \bar{S}_i \bar{G}_j \) is the directional total exchange area \((m^2)\) from \( S_i \) to \( G_j \) [7], \( T_i \) is the temperature of \( S_i \), \( \sigma \) is the Boltzmann constant.

2. Radiative heat flow emitted by the gas and the wall \( R_3 \) and \( R_6 \)

\[ R_3 = 4kV_{G_i}a(\frac{T_{G_i} + T_{S_i}}{2})\sigma (\frac{T_{G_i} + T_{S_i}}{2})^4 \]  
(7)

\[ R_6 = A_j \varepsilon_j \sigma T_j^4 \]  
(8)

where \( k \) is the gas (gray body) absorption coefficient \((m - 1)\), \( V_{G_i} \) is the volume of the gas \( G_j \) \((m^3)\), \( a(\frac{T_{G_i} + T_{S_i}}{2}) \) is the weight factor of the gas for calculating the actual gas blackness, [12] [13], \( A_j \) is the wall surface area \((m^2)\), \( \varepsilon_j \) is the wall blackness.

3. Methane combustion heat \( Q_f \)

\[ Q_f = w_{CH_4}m_f Q_{di} \]  
(9)

where \( w_{CH_4} \) is the quality fraction of CH4 in biogas, \( m_f \) is the mass flow rate of methane \((kg/s)\), \( Q_{di} \) is the low heating value of CH4 \((kJ/kg)\).

4. Convective heat loss of gas \( C_i \)

\[ C_i = h_1A_j(\frac{T_{G_i} + T_{S_i}}{2} - T_{S_i}) \]  
(10)

where \( h_1(kW/(m^2*K)) \) is the convective heat transfer coefficient for the gas sweeping wall.

5. Gas enthalpy difference between outlet and inlet in a unit

\[ \Delta H = mC_p \Delta T \]  
(11)

Boundary conditions: the combustor entrance is equivalent to gray surface and the gray surface’s temperature is the preheated air temperature, the combustor outlet is equivalent to gray surface and the gray surface’s temperature is the Stirling engine heat pipe wall temperature and the gray surface’s blackness is the rate of effective area for the Stirling engine heat tube bundles. The heat released from the combustion of methane is taken as the heat source and located in the units corresponding to the fuel nozzle.

**Convective Heat Transfer between Gas and Fin-Free Heat Pipes and Finned Heat Pipes**

1. Convective heat transfer between gas and fin-free heat pipes \( Q_{sf} \)

\[ Q_{sf} = h_sA_s(T_f - T_w) \]  
(12)

\[ h_s = \frac{Nu}{d} \lambda \]  
(13)
\[ \overline{Nu} = Nu_f c_{\beta} c_z \]  
(14)

where \( Nu_f \) is got from the literature \([11]\), \( c_{\beta} \) is the corrected coefficient due to the change of the impact Angle, \( c_z \) is the coefficient due to the change with bundle rows \([14]\), the physical parameters of gas are obtained from the literature \([15]\).

2. Convective heat transfer between gas and finned heat pipes \( Q_{i2} \)

\[ Q_{i2} = h_2 A_2 (T_f - T_w) \]  
(15)

\[ h_2 = \frac{Nu_z \lambda}{d} \]  
(16)

\[ Nu_z = Nu c_z \]  
(17)

\[ Nu = 0.245 \text{Re}^{0.58} \text{Pr}^{0.333} \]  
(18)

\[ A_2 = A_b + \eta_f A_f \]  
(19)

where, \( \eta_f \) is the fin efficiency \([16]\).

Results and Analyses

The fuel is biogas and the temperature of air in the inlet is 300K. The working medium is helium in Stirling engine. The average helium pressure is 14MPa and the engine speed is 1500RPM.

Influence of the Biogas Flow Rate and the Air Flow Rate

The performance of the system is affected by the biogas flow rate and the air flow rate when the Stirling generation power system uses the biogas.

The biogas flow rate ranges from 5.5g/s to 8g/s. The air flow rate range from 55g/s to 105g/s. The efficiency of the SGHR is defined by \( \eta = Q_{i2}/Q_f \). \( m_a, m_f \) are, respectively, the air flow rate and the biogas flow rate. As shown in Figure 4 to Figure 6, the maximum temperature of gas in combustor and the heat transfer power and the heat transfer efficiency of the SGHR decrease when the air flow rate increases. As the air flow rate increases from 65g/s to 84.5g/s, the maximum temperature of gas in combustor decreases by 19.62%, the heat transfer power and the heat transfer efficiency of the SGHR decrease by 13.84% when the biogas flow rate is 6.5g/s. The reason is that as the air flow rate increases, the excess air coefficient increases, the gas temperature in the combustor decreases, the temperature difference between gas and heat pipes of Stirling engine decreases dramatically, so the heat transfer power and the heat transfer efficiency decrease.

As the biogas flow rate increases, the maximum temperature of gas in combustor and the heat transfer power increase, and the heat transfer efficiency of the SGHR decreases. As the biogas flow rate increases from 6g/s to 7.5g/s, the maximum temperature of gas in combustor increases by 18.37%, the heat transfer power of the SGHR increases by 19.97%, and the heat transfer efficiency of the SGHR decrease by 4.44% when the air flow rate is 75g/s. The reason is that as the biogas flow rate increases, the excess air coefficient decreases, the gas temperature in the combustor increases, the temperature difference between gas and heat pipes of Stirling engine increases, so the heat transfer power increases. Due to the flue gas temperature in the environment increasing, and the exhaust heat loss increasing, the heat transfer efficiency of the SGHR decreases. So, increasing the biogas flow rate can increase the heat transfer power of the SGHR, but not improve the heat transfer efficiency of the SGHR.
Influence of the Concentration of CH4 in Biogas

In the actual production process, the concentration of CH4 in biogas is not static, of which volume fraction fluctuates between 50% and 72%, and the calorific value of CH4 is very high, at the same time the increase of its concentration will cause the gas temperature to rise rapidly, which will cause damage to the Stirling engine heat pipes. Therefore, it is important to determine a reasonable biogas component for the SGHR.

In this paper, five sets of biogas and air flow rates were selected to study the effect of biogas composition on the performance of the SGHR. Table 1 shows the control parameters of the five schemes.

<table>
<thead>
<tr>
<th>No.</th>
<th>Biogas flow rate: $m_f$ g/s</th>
<th>Air flow rate: $m_a$ g/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6</td>
<td>60</td>
</tr>
<tr>
<td>2</td>
<td>6.5</td>
<td>71.5</td>
</tr>
<tr>
<td>3</td>
<td>7</td>
<td>77</td>
</tr>
<tr>
<td>4</td>
<td>7.5</td>
<td>90</td>
</tr>
<tr>
<td>5</td>
<td>8</td>
<td>96</td>
</tr>
</tbody>
</table>

The main performance parameters of the SGHR vary with the concentration of CH4, as shown in Figure 7 to Figure 9. It can be seen that the maximum temperature of the gas in the combustor and the heat transfer power of the SGHR increase linearly with the increasing of CH4 concentration, but the heat transfer efficiency presents different variative rules. As shown in Figure 9, the efficiency of heat transfer increases first and then decreases in schemes 1 and 2, and in schemes 3, 4 and 5, the heat transfer efficiency decreases all the time. When the concentration of CH4 increases from 55% to 65% in the scheme 3 (mf = 7 g / s, ma = 77 g / s), the maximum temperature of the gas in the combustor increases by 26.19% and the heat transfer power increased by 27.14%, however, the heat...
transfer efficiency of the SGHR reduces by 2.96%. The increase of heat transfer power and the gas temperature is due to the high calorific value of CH4. With the increase of CH4 concentration, the heat released by the combustion of biogas increases sharply, so the gas temperature increases obviously, which makes the temperature difference between gas and the heat pipe of Stirling engine increases, and the heat transfer power increases.

It can be seen from Figure 9 that the heat transfer efficiency does not increase as the CH4 concentration increases as the heat transfer power does, but decreases with the increasing of CH4 concentration. In scheme 1 and 2, extreme values appear. The reason is that the amount of heat required for a Stirling engine is constant at the operating point of a solar/gas Stirling power generation system. With the increase of CH4 concentration, the required biogas flow rate decreases, the amount of air required for complete combustion of biogas remains unchanged, so the gas flow rate decreases, the gas velocity decreases, and the convective heat transfer power decreases. As shown in Figure 10, In the heat transfer process of gas and the heat pipes of the Stirling engines, convection heat transfer accounted for a major part and the radiation heat transfer accounted for a small proportion, although the increasing of gas temperature enhances the radiation heat transfer, the convection heat transfer decreases more. Therefore, heat transfer efficiency and heat transfer power show different trends.

**Summary**

In this paper, a thermal model of the solar/gas Stirling power generation system is established which based on the law of conservation of energy. Through this model, the steady-state performance
calculation and analysis of the SGHR are carried out, and the conclusions are shown as following:

1. The heat transfer power increases as the biogas flow rate increasing. Meanwhile, due to the gas temperature increases, resulting in loss of exhaust increases, heat transfer efficiency decreases. As the biogas flow rate increases from 6g/s to 7.5g/s, the maximum temperature of gas in combustor increases by 18.37%, the heat transfer power of the SGHR increases by 19.97%, and the heat transfer efficiency of the SGHR decrease by 4.44% when the air flow rate is 75g/s. As the air flow rate increases, resulting in the decrease of the heat transfer power and the heat transfer efficiency of the SGHR. As the air flow rate increases from 65g/s to 84.5g/s, the maximum temperature of gas in combustor decreases by 19.62%, the heat transfer power and the heat transfer efficiency of the SGHR decrease by 13.84% when the biogas flow rate is 6.5g/s.

2. The heat transfer power increased linearly with CH4 concentration increasing. As shown in Figure 9 and Figure 10. When the concentration of CH4 increases from 55% to 65% in the scheme 3 (mf = 7g / s, ma = 77g / s), the maximum temperature of the gas in the combustor increases by 26.19% and the heat transfer power increased by 27.14%, however, the heat transfer efficiency of the SGHR reduces by 2.96%.

Acknowledgement

This research was financially supported by the National High Technology Research and Development Program of China.

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