Research paper

Design and characteristics of a novel tapered tube bundle receiver for high-temperature solar dish system

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HIGHLIGHTS

- A novel Tapered Tube Bundle Receiver for high-temperature dish system was designed.
- Thermal efficiency of the novel receiver can achieve 80%, heating up air to 1340 K.
- The new designed receiver can receive solar energy uniformly.
- Flow field obtained by CFX indicating the receiver can realize uniform flow.

ABSTRACT

Most of current receivers for high-temperature (>1000 K) solar thermal utilizations were confronted with non-uniform solar flux distribution on absorbers, which would result in cracking and melting as well as flow instability. This paper proposes a new design of tapered tube bundle receiver (TTBR) by investigating the geometrical principle of dish concentrators, which makes each passage of the absorber parallel with the concentrated sun rays to absorb uniform solar thermal. The performance of the new design was confirmed using the theory of energy balance, Ray-tracing simulation and CFX simulation. Calculations based on the theory of energy balance indicated that the thermal efficiency of TTBR achieves 80%. Ray-tracing simulation results showed that solar rays are able to irradiate the entire tubes and uniformly distribute on each tube. And according to the flow field simulation by CFX, attributed to the circumferential inflow structure, almost equivalent air mass rate was obtained in each tube, indicating the designed TTBR receiver can achieve uniform flow in the absorber passages, which laid a good foundation for the further stable and efficient heat transfer procedure.

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1. Introduction

Solar thermal system is one of the most eco-friendly methods to generate electric power. Solar dish tends to have higher collection efficiency comparing with solar tower and trough systems [1]. In the dish system, solar rays are collected by a parabolic dish concentrator that reflects the rays onto a receiver located at the focal plane. In the receiver, working fluid is heated to high temperature utilizing the convert thermal energy, and then the high-temperature working fluid drives a turbine to generate electricity. According to the laws of thermodynamics, the temperature of working fluid determines the efficiency of engine. According to that, the receiver is the key component to increase the efficiency of system. For example, Daguenet-Frick et. al. [2] indicated that an installation with a receiver outlet temperature that reaches 1500 K could have a global efficiency of 40%. Meanwhile, the lower processing and assembling fees of the receiver is benefit to reduce the cost of the solar dish thermal power system [3].

In order to improve the performance of solar dish thermal power systems, many researchers pay attentions to the design of receiver. Comparing with other receivers, cavity receivers have better performances in high temperature [4]. The development of cavity receiver was reviewed by Antonio [4] and Behar [5]. The extrusive problem of these receivers is the non-uniform solar flux distribution that would cause cracking and melting on absorber walls as well as flow instability.

In high-temperature (>1000 K) solar thermal applications, parabolic mirrors concentrate solar rays to high density power (in
the range of 4–10 MW/m²) on the focal plane with 5000–10,000 concentration ratio. The solar flux distribution on the focal plane is inherently non-uniform. The non-uniform solar flux distribution causes high temperature gradients and local over-heating easily that lead to melt and crack on the absorber walls, which will greatly reduce its thermal efficiency as well as life-span [6].

At the same time, in high temperature receivers, especially porous receivers, the flow has the tendency to develop into unstable flow, which is mainly due to the non-homogeneous solar flux distribution [7–9]. This is because that the higher solar flux usually causes a higher local temperature, while the viscosity of air increases with the temperature increasing, so that flow resistance in the hotter region is higher. Additionally, the flow with high temperature (or resistance) aggravate the non-uniform temperature distribution because that the corresponding heat transfer rate reduces sharply due to high temperature and low velocity of air. The high local temperature causes local damaging. For example, a cordierite receiver melts when the measured air outlet temperature is only 1173 K, although the melting temperature of cordierite is 1723 K [10].

Therefore, it is significant to design a receiver that can achieve a uniform solar flux distribution on absorber walls. The previous investigations indicated that it can be solved by optical optimization to design the receiver with special geometric configuration to obtain uniform solar rays [11]. Li et. al. [12] illustrated that the solar flux would be more uniform when put the receiver over or under the focal plane. But it is contradicts with the advantage of high concentration ratio to realize high capacity heat transfer. Shuai et. al. [13] employed Monte Carlo Ray Tracing (MCRT) method to trace the rays and proposed a receiver with an upside-down parabola configuration to realize uniform flux distribution. However, such specific ‘pear’ curved surface is difficult and expensively to be manufactured.

On the other hand, porous material is one ideal material to fabricate the absorber of receiver to enhance the capacity of heat transfer in a limited focal space [14–16]. Honeycomb is one typical porous material. It has been proved that honeycomb receivers offer advantages of thin walls, high geometric surface area and, therefore, good gas–solid contact, accommodation of high gas flow rate combined with low-pressure drop and good mass transfer performance [14]. A honeycomb receiver has been operated in 2009 for the first time [17]. However, it also has the problem of non-uniform flux distribution according the experimental results [17].

The previous work indicated that there is no ideal receiver with both uniform flux distribution and simple structure. The drawback of receiver limits the application of solar thermal systems.

This paper puts forward an innovative receiver design. The novel design has both the advantages of a honeycomb configuration in enhancing heat transfer performance and uniform flux distribution. At the same time, the design is bundled using tapered tube. The structure of the receiver is simple and the cost is cheap. The performance of the novel receiver is verified using energy balance analysis, ray-tracing and CFD simulations.

The configuration of the paper is that: first, the innovative structure-tapered tube bundle will be introduced. The principle and theory of design will be explained. And then the receiver with optimized dimensions for a case solar dish system will be modeled; the flux distribution and flow pattern within the receiver will be simulated to validate its proposed performance.

2. Receiver design and optimization

2.1. The innovative structure

The TTBR is proposed basing on the geometrical principle of the dish concentrated rays. Parabolic dish concentrator concentrates the sunlight into the focus and makes the parallel rays shape into circular cone beam, which can be referred from the previous paper [18] in detail. While the design of TTBR is based on the following assumption:

- The concentrator is a perfect parabolic dish and without any tracking errors
- The impact of the solar angle is ignored

The absorber of the TTBR is a conical bundle (Fig. 1(b)) that composed by an array of small tapered tube units (Fig. 1(a)). Under the assumed conditions, such circular array makes it possible that tubes on each ring have the same slope with corresponding incident rays. As shown in Fig. 2, in a 2D view, the vertex angle of the conical bundle \( \alpha \) is same as that of the circular cone beam, which is determined by the focal length \( f(OF) \) and the aperture \( D(AB) \) of the concentrator:

| Nomenclature | \( A \) | \( c \) | \( d \) | \( D \) | \( f \) | \( G \) | \( h \) | \( H \) | \( I \) | \( m \) | \( n \) | \( q \) | \( Q \) | \( T \) | \( v \) | \( X \) | \( Z \) |
|-------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|
| Latin symbols | effective heat transfer area \( (m^2) \) | specific heat capacity \( (J/kg K) \) | diameter \( (m) \) | aperture of the concentrator \( (m) \) | focal length \( (m) \) | incident power onto the absorber \( (W) \) | heat transfer coefficient \( (W/m^2K) \) | height of the tapered tube \( (m) \) | solar irradiance \( (W/m^2) \) | mass flow rate \( (kg/s) \) | the number of the tube rings | effective absorbed energy of air stream \( (W) \) | heat transfer power \( (W) \) | absolute temperature \( (K) \) | velocity \( (m/s) \) | angle factor |
| Greek symbols | \( \alpha \) | \( \varepsilon \) | \( \eta \) | \( \rho \) | \( \zeta \) | \( \tau \) |
| Subscripts: | a | b | c | f | g | i | o | r | w |
| a | ambient | b | bottom of the receiver | c | concentrator | f | air flow | g | glass (window) | i | inlet air/serial number of each tube | o | outlet air | r | reference temperature | w | absorber wall | Z | the receiver designed position |
the center of the parabolic concentrator (O) and the bottom of the receiver (M) can be calculated as following:

$$Z = f - \frac{(2n + 1)(d_3 + 2\delta)}{\alpha}$$

(4)

The tapered tubes are made of ceramics that can bear temperature as high as 1800 K [19] to ensure the requirement of high-temperature.

The air passage is designed as shown in Fig. 3. It has the obvious merit that the inflowing air from the annular channel with lower temperature can absorb the heat transferring from the high-temperature inner walls without heat loss. Meanwhile, such air inflowing pattern can lead to more uniform air flow in the tube bundle, which will be demonstrated in part 4 of the paper.

2.2. Mathematical model and optimization

Energy balance analysis of the TTBR was applied to obtain the optimized geometric structure to satisfy both the heat transfer demand and the highest efficiency, whilst to achieve low flow resistance. Fig. 3 shows the schematic diagram of the TTBR. The assumed conditions for the theoretical calculation are:

- The receiver is operating in a steady-state regime.
- Fluid flow in the tubes is fully developed and no phase change.
- The walls and the glass window are treated as gray bodies.
- All temperatures, heat fluxes and thermodynamic properties are uniform in each of the elements.

An energy balance of the air stream gives

$$Q = Q_{\text{air-w}} - Q_{\text{air-g}}$$

that is

$$cm(T_0 - T_{i}) = h_{\text{f-w}}A_w(T_w - T_{f}) - h_{\text{f-g}}A_g(T_1 - T_g)$$

(5)

where $T_i$ is the reference temperature that is defined as $T_r = (T_1 + T_0)$.

An energy balance on absorber walls gives:

$$I_A c_P c_g \sigma_w = Q_{\text{air-w}} + Q_{\text{w-g}}$$

$$= h_{\text{f-w}}A_w(T_w - T_{f}) + \frac{\sigma(T_w^4 - T_f^4)}{\frac{1}{\lambda_{w\text{air}}} + \frac{1}{\lambda_{w\text{g}}} + \frac{1}{\lambda_{w\text{g}}} + \frac{1}{\lambda_{g\text{g}}}}$$

(6)

An energy balance on the glass window gives:

$$I_A c_P c_g + Q_{\text{air-g}} + Q_{\text{w-g}} = Q_{\text{con-loss}} + Q_{\text{rad-loss}}$$

that is

$$I_A c_P c_g + h_{\text{f-g}}A_g(T_1 - T_g) + \frac{\sigma(T_w^4 - T_g^4)}{\frac{1}{\lambda_{w\text{air}}} + \frac{1}{\lambda_{w\text{g}}} + \frac{1}{\lambda_{w\text{g}}} + \frac{1}{\lambda_{g\text{g}}}}$$

$$= h_{\text{g}}A_g(T_g - T_a) + c_g \sigma A_g \left( T_g^4 - T_a^4 \right)$$

(7)

The thermal efficiency can be defined as the ratio between the effective absorbed energy of air stream and the incident power onto the absorber [11]:

$$\eta = \frac{Q}{G} = \frac{cm(T_0 - T_i)}{I_A c_P}$$

(8)
Flow resistance of the tubes bundle is evaluated by:

$$
\Delta p = \frac{1}{2} \xi \rho v^2 \frac{H}{d}
$$

(9)

where \(d\) is the equivalent diameter, \(d = (d_1 + d_2)/2\) and the frictional resistance coefficient \(\xi\) is referred from flow in a reducer with circular cross section in flow Handbook [20].

The studied case solar system is a 20 kW hybrid solar-gas Stirling power system that was designed by the National Key Lab. of Science and Technology on Aero-Engines, Beihang University, aiming to achieve high overall efficiency of 30%. The responsibility of receiver in the system is to heat up airflow (0.175 kg/s) from 1033 K to 1340 K to carry out heat transfer power as high as 60 kW under a normal pressure. Parameters regarding to the receiver are shown in Table 1.

All the parameters, especially optical properties of the concentrator mirror and window glass in Table 1 are given by the case system. It should be mentioned that the absorptivity of ceramic materials is 0.2 [21] in the case system. But 0.95 is presented as the absorptivity of the TTBR absorber for the theoretical calculation, it attributes to the shape of the tapered tube which is like the effect of blackbody cavity. Rays irradiate into the tapered tube, 20% will be absorbed by the tube wall firstly while 80% will be reflected. As the aperture of tube is very small comparing with the length, the reflected rays are hard to escape, while they will reflect to the tube wall again and again and be absorbed. The only fraction that cannot be absorbed by the tube is the rays that leak from the aperture of the tube. According to ray-tracing simulations, the fraction of the leakage is 5.18%. Therefore, 0.95 is obtained as the absorptivity of the TTBR absorber for the theoretical calculations.

Programming above relations (1–9), a series of dimensions of satisfied tapered tube model can be obtained as shown in Table 2. These nine models have the same open diameter (that is of the window (Fig. 3), 0.45 m). The features, in terms of thermal efficiency, temperature, weight and flow resistant of each model are shown in Fig. 4.

As can be seen from Fig. 4, the model with relatively lower weight possesses a lower flow resistance but a higher wall temperature to achieve the certain heat transfer power. This is because the inverse relationship between flow and heat transfer [22], whilst the weight of the tubes presents the heat transfer area that is also inversely proportional with the temperature drop between absorber walls and the air flow as the heat transfer power was given. Meanwhile, the thermal efficiency decreases with the wall

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### Table 1

<table>
<thead>
<tr>
<th>Content</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Focal length of the concentrator</td>
<td>(f)</td>
<td>m</td>
<td>7.2</td>
</tr>
<tr>
<td>Diameter of the concentrator</td>
<td>(D)</td>
<td>m</td>
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<tr>
<td>Diameter of the window glass</td>
<td>(d_g)</td>
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<tr>
<td>Heat transfer power</td>
<td>(Q)</td>
<td>kW</td>
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<td>Inlet temperature of air flow</td>
<td>(T_i)</td>
<td>K</td>
<td>1033</td>
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<tr>
<td>Outlet temperature of air flow</td>
<td>(T_o)</td>
<td>K</td>
<td>1340</td>
</tr>
<tr>
<td>Mass flow rate</td>
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<td>kg/s</td>
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<td>Ambient temperature</td>
<td>(T_a)</td>
<td>K</td>
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<tr>
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<td>Absorptivity of the TTBR absorber</td>
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<td>0.95</td>
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<td>Emissivity of the TTBR absorber</td>
<td>(\varepsilon_w)</td>
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<tr>
<td>Transmissivity of the window glass</td>
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<tr>
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<tr>
<td>Emissivity of the window glass</td>
<td>(\varepsilon_g)</td>
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<td>0.9</td>
</tr>
</tbody>
</table>

### Table 2

<table>
<thead>
<tr>
<th>Model</th>
<th>Diameter ((d_1 – d_2)\ mm)</th>
<th>Height (mm)</th>
<th>Quantity of the tubes</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>2–5</td>
<td>105</td>
<td>2269</td>
</tr>
<tr>
<td>2</td>
<td>2–4</td>
<td>83</td>
<td>3169</td>
</tr>
<tr>
<td>3</td>
<td>3–5</td>
<td>105</td>
<td>1261</td>
</tr>
<tr>
<td>4</td>
<td>3–6</td>
<td>90</td>
<td>1657</td>
</tr>
<tr>
<td>5</td>
<td>3–5</td>
<td>70</td>
<td>2269</td>
</tr>
<tr>
<td>6</td>
<td>4–7</td>
<td>79</td>
<td>1261</td>
</tr>
<tr>
<td>7</td>
<td>3–4</td>
<td>42</td>
<td>3169</td>
</tr>
<tr>
<td>8</td>
<td>4–6</td>
<td>60</td>
<td>1657</td>
</tr>
<tr>
<td>9</td>
<td>4–5</td>
<td>35</td>
<td>2269</td>
</tr>
</tbody>
</table>

Fig. 3. Schematic diagram of the TTBR.
temperature increasing, which is reasonable that temperatures have a direct influence on efficiencies [11].

In addition to the thermal efficiency, flow resistance is another important factor regarding to the performance of a solar system, as a higher flow resistance will need a larger output power to drive the flow of air, and then decrease the whole system efficiency. With the preference of flow resistance lower than 300 Pa for the studied case system on receiver part, Model 4 (that is 290 Pa with thermal efficiency 80%) has been chosen as the optimized parameter.

3. Ray-tracing simulation

Ray-tracing simulation by TracePro software has been taken to identify that the TTBR can receive solar radiation uniformly. In the simulation, the directional performance of concentrated solar rays was predicted using Monte-Carlo method [23-25].

3.1. Model simplification and parameters setting

The dimensions of the optimized TTBR (Model 4) have been employed here for the simulation. Fig. 1 indicated that the structure of receiver is ring, and each tube within the same ring has identical characteristic. Therefore, the model can be simplified as one radius of the ring, which can be seen from Fig. 5, in order to save the computational space and time.

Parameters are set as shown in Table 3. The errors of the parabolic dish concentrator is neglected. The surface of the concentrator is set as reflectivity \( r = 0.92 \), same as the theoretical calculation. Solar angle (32') is taken into consideration in the simulation and the solar radiation is set as 800 W/m². The tapered tubes are made of ceramic that absorptivity is 0.2 [21].

As the design parameters, the standard position of the model 4 TTBR relatively to the vertex of the parabolic concentrator is \( Z = 6990 \) mm. And also, the positions deviate vertically and horizontally from the standard position 10 mm, 50 mm and 100 mm are simulated respectively to investigate its installed tolerance.

3.2. Ray-tracing results and discussion

Validation of TracePro simulations can be referred from the previous research [18].

In order to quantify the maximum deviation in the positive and negative value among the tubes, two parameters are defined as

\[
\varepsilon_+ = \frac{E_{\text{max}} - E_{\text{ave}}}{E_{\text{ave}}} \times 100\% \quad (10)
\]

\[
\varepsilon_- = \frac{E_{\text{min}} - E_{\text{ave}}}{E_{\text{ave}}} \times 100\% \quad (11)
\]

where \( E_{\text{max}} \) is the maximum solar energy that received among the tubes, \( E_{\text{min}} \) is the minimum value, and \( E_{\text{ave}} \) is the average value in arithmetic.

3.2.1. Standard position

The simulation result of TTBR on the standard position is shown in Fig. 6(a). As can be seen from the energy distribution diagram, solar rays are able to irradiate the entire tubes and uniformly distribute on each tube.

Energy received by each tube is listed in Fig. 6(b). The maximum solar energy that received by the tubes is 62.21 W, the minimum is 54.06 W and the average is 58.04 W. The standard deviation [26] is 2.49 W. According to the expressions of (10) and (11), the positive deviation \( \varepsilon_+ \) is +7.2% while the negative deviation is −6.4%.

3.2.2. The effect of deflected positions

The performance of the TTBR will be significantly affected by position as it is designed restrictively following with the direction of the concentrated rays. The installation accuracy in real engineering practices must be considered. Therefore, it is necessary to investigate the energy received by TTBR located at deflected positions.

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</tr>
<tr>
<td>Reflectivity of the concentrator mirror</td>
<td>r</td>
<td>/</td>
<td>0.92</td>
</tr>
<tr>
<td>Absorptivity of the TTBR absorber</td>
<td>( \alpha_w )</td>
<td>/</td>
<td>0.2</td>
</tr>
<tr>
<td>Number of the traced rays</td>
<td>N_r</td>
<td>/</td>
<td>1917601</td>
</tr>
<tr>
<td>Standard position of the TTBR (Z)</td>
<td>Z</td>
<td>mm</td>
<td>6990</td>
</tr>
</tbody>
</table>
The deflected positions studied here are the positions deviate vertically or horizontally from the standard position respectively 10 mm, 50 mm and 100 mm. According to the simulation results, the energy received by each tube at the deflected positions were shown in Fig. 7. The results indicated that the energy distribution becomes non-uniform when the position diverges. The deviation of energy received by individual tube can be evaluated by $\psi_i$, which is defined as:

$$\psi_i = \frac{E_i - E_{i,\text{standard}}}{E_{i,\text{standard}}} \times 100\%$$

where $E_i$ is the energy received by individual tube on the deflected positions, $E_{i,\text{standard}}$ is the corresponding tube on the standard position, and $i$ is the sequence number of each tube.

Fig. 7 (a) shows the results of ray-tracing simulation of that the TTBR deviates from the standard position vertically upwards.
10 mm, 50 mm and 100 mm. As the TTBR deviates upwards to 10 mm, the maximum positive deviation of energy received occurs in the 7th tube where $j_7$ is 19.2%, while the maximum negative is the tube on the rim with $j_23 = -12.4\%$. As the deflected distance is 50 mm, all the deviation of energy received is negative, indicating the energy received by each tube at deflected positions is lower than that at the standard positions. At the downward deflected position of 50 mm, the maximum deviation of received energy is $-50.3\%$ and the minimum is $-4.1\%$. As the deflected distance is 100 mm, the maximum deviation is $-71.4\%$ while the minimum is $-14.9\%$. It is noticeable that the energy received by tubes that at the downward positions is mostly lower than that at the standard positions, which will reduce the energy fully utilization but will not bring dangerous by high temperature. In real practices, 20% margin on heat transfer power should be considered during the receiver design process. Therefore, the installation error of TTBR should be downside deviations in a range of 10 mm, all the deviation of energy received is negative value, indicating the energy received by each tube is in the range of 5% to $-20\%$ that will not influence the performance of the TTBR.

On the other hand, as the TTBR deviates downwards to 10 mm, as can be seen from Fig. 7(b), the maximum positive deviation is $\psi_{23}$ is 4.7% and the negative is $\psi_{10}$ is $-19.5\%$. As the deflected distance is 50 mm, all the deviation of energy received is negative value, indicating the energy received by each tube at deflected positions is lower than that at the standard positions. At the downward deflected position of 50 mm, the maximum deviation of received energy is $-50.3\%$ and the minimum is $-4.1\%$. As the deflected distance is 100 mm, the maximum deviation is $-71.4\%$ while the minimum is $-14.9\%$. It is noticeable that the energy received by tubes that at the downward positions is mostly lower than that at the standard positions, which will reduce the energy fully utilization but will not bring dangerous by high temperature. In real practices, 20% margin on heat transfer power should be considered during the receiver design process. Therefore, the installation error of TTBR should be downside deviations in a range of 10 mm, all the deviation of energy received is negative value, indicating the energy received by each tube is in the range of 5% to $-20\%$ that will not influence the performance of the TTBR.

4. Flow field simulation

The flow field in the receiver was simulated by the CFX software.

4.1. Model simplification and parameters setting

The simplified model of the TTBR for the simulation can be seen in Fig. 8. The tube bundle is simplified as corresponding tapered annulus structure, which can be referred from Ref. [18]. Due to the similar structure in circumference, only part of the receiver (a 10° piece of the whole circle through the axis of the receiver) is employed for the simulation. To obtain a stable velocity distribution and decrease backflow, the air inlet and outlet is extended to a length of 1.5 times of the diameter of pipe.

Hexahedral mesh is employed here by ICEM as it allows mesh refinement near the wall and reduces the number of meshes without any influence on the computational accuracy [27]. The size of the grids in the overall domain is 10 mm while grids on the walls are intensified. The grid on the absorber wall is 0.5 mm and that on the corners of the TTBR flow pass is 1 mm. Grid independence is imperative to be tested before the simulations. Six different scales of grid were processed in the numerical calculation, which grid numbers ranging from 1,900,000 to 5,800,000. It indicated that the results were grid-independent when the grid number was larger.
than 2,680,000. Therefore, the grid of 2.68 million was applied for the computations.

The working fluid here is air with temperature of 1033 K and normal pressure. The inlet air flow rate of the case solar dish system is 0.175 kg/s, while the cross-sectional area of the inlet pipe is 0.031 m² so the speed of the inlet flow is 17 m/s. As shown in Fig. 8, the boundary condition of inlet is that the air has a normal speed of 17 m/s and that of outlet is set as static pressure of 0 Pa. The ventral surfaces are set as symmetry type, and the others are no slip walls. Shear Strain Transport (SST) model was employed here for its accurate and robust near wall treatment [28].

4.2. Flow field simulation results and discussion

The structures of receiver, especially the angle and the radian of the turning, were modified several times according to the simulation results until make sure that the air can uniformly flow through each tapered tube passage.

Fig. 9 shows the flow field of the optimized configuration of the TTBR. It can be noticed that there are two vortexes before the air flow into the tapered tubes. One is a big vortex that on the flank of the cavity, and the other is a small vortex that closes to the window. These two vortexes are beneficial for the flow distribution because they reassign the direction of the air stream. Meanwhile, the simulation result also indicates that the total pressure drop between the inlet and outlet is 320 Pa.

Mass rate of the corresponding each tapered tube can be gained as shows in Fig. 10 by capturing the data from the inlet and outlet sections of each annulus. The y axis stands for the mass rate, while the x axis represents the sequence number of each annulus/tube counting from the center. The mass rate in each tube along the radial direction can be got by converting each annulus area to the corresponding tubes. According to Fig. 10, the average mass flow rate is 10.8 × 10⁻⁵ kg/s, the maximum is 11.8 × 10⁻⁵ kg/s, and the minimum is 9.75 × 10⁻⁵ kg/s. It has almost equivalent mass flow in each tube with the maximal standard deviation 0.62 × 10⁻⁵ kg/s.

Therefore, a conclusion can be drawn that the designed TTBR receiver can achieve uniform flow in the absorber passages, which laid a good foundation for the further stable and efficient heat transfer procedure. However, the fluid properties influenced by temperature variations were not considered in the CFX simulation. It would be best for further researches if the irradiated energy distribution on the TTBR, which was obtained from ray-tracing simulations, can be added into the CFX simulation as a boundary condition to investigate the distributions of both flow and temperature on the TTBR. Besides, experiments on the TTBR for real parabolic dish concentrators are also necessary to be performed, in support of an evaluation of whether the TTBR can truly bring uniform flow and high efficient heat transfer.

5. Conclusion

Design methodology of the novel tapered tube bundle receiver (TTBR) for high-temperature solar dish system was presented. A model TTBR for a case solar system was demonstrated. The responsibility of receiver for the case system is heating up inlet air stream (with a mass flow rate of 0.175 kg/s) from 1033 K to 1340 K under a normal pressure, achieving 60 kW heat transfer power. According to the calculations, the optimized TTBR can achieve thermal efficiency of 80% to satisfy the demand of the case system, occupying 1657 tapered tubes, and each tube has the diameter of 3–6 mm and the height of 90 mm. Ray-tracing simulations showed that solar rays are able to irradiate the entire tubes and achieve uniform distribution but restricted control of position deviations is significant in real practices. Flow field simulation by CFX indicated...
the TTBR can achieve uniform flow in the absorber passages, which laid a good foundation for the further stable and efficient heat transfer procedure.

References


