Numerical Simulation of Straight Rib Solar Air Collector

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Abstract. In this passage, a straight rib solar air collector is proposed. The effects of the number of ribs, the height of the ribs, the length of the ribs and the air inlet velocity on the heat collection efficiency and the normalized temperature difference are studied using the orthogonal test method. The orthogonal test plan with 4 factors and 3 levels is selected to perform numerical simulations on different collector models. The results show that the significance of each orthogonal factor on the heat collection efficiency and the normalized temperature difference is: Inlet velocity > Height of ribs > Number of ribs > Length of ribs, and get the best parameter combination: the number of ribs is 9, the rib height is 45mm and the rib length is 2000mm. In addition, the study found that the average heat collection efficiency and normalized temperature difference of the model are 3.98% and 0.00052 (Km²)/W higher than the ordinary solar air collector.

Keywords: Straight rib solar air collector; Heat collection efficiency; The normalized temperature difference; The orthogonal test method; Numerical simulation.

1. Introduction
The solar air collector (SAC) has the advantages of simple structure, reliable operation and convenient installation and maintenance. It is widely used in building auxiliary heating, product drying and seawater desalination. However, the convective heat transfer coefficient between the air inside the heat collector and the heat absorption plate is not high, resulting in a slightly lower heat collection efficiency, which limits the development and promotion of solar air collectors[1].

In order to enhance the convective heat transfer between the air and the heat absorption plate, the structure optimization is mainly carried out by setting a spoiler or ribs on the heat absorption. Cao et al.[2] conducted numerical simulation and theoretical analysis of the serpentine-channel solar air collector, and explored the characteristics of the air inlet velocity and solar radiation intensity on the outlet temperature and heat collection efficiency. Jia et al. [3] designed a vortex solar air collector. Through experiments and simulation studies, they found that the vortex solar air collector has a 2.18% higher heat collection efficiency than the serpentine channel air collector, and the pressure loss of the former is only 37.26% of the latter, which significantly reduces the inlet and outlet pressure loss of collectors. Cheng et al.[4] proposed a new type of parabolic air collector. The numerical simulation analysis shows that compared with the traditional flat and triangular corrugated heat-absorbing plate structure, the parabolic air collector has higher instantaneous heat collection efficiency and smaller pressure drop. Singh[5] conducted an experimental study on the air flow characteristics of the V-rib solar air collector with a certain roughness. Ekadewi et al. [6] installed V-shaped ribs and spoilers on the heat absorption plate, and analyzed the optimal spacing ratio S/H = 1 through experiments and simulations.

In summary, the addition of spoilers or ribs can effectively enhance the convective heat transfer between the air and the heat absorption plate. Therefore, straight ribs are installed on the heat...
absorption plate to optimize the structure. A straight-rib solar air collector is proposed, and the orthogonal test method is used to conduct heat transfer characteristics for numerical simulation and theoretical analysis.

2. Straight Rib Solar Air Collector Model

The model of solar air collector studied in this paper is shown in Figure 1. The size of the collector is 2000mm × 1000mm × 85mm. The main components include glass cover, shell, heat absorption plate, insulation layer and ribs. The upper part of the collector is installed with 3.2mm thick single-layer flat tempered glass; both sides and bottom of the collector are covered with a 40mm thick asbestos layer; the air inlet and outlet are slit-shaped tuyere, the size of which is 920mm × 45mm; The surface is sprayed with a special coating with high absorption rate, and its size is 2000mm × 920mm. The physical property parameters of each material are shown in Table 1.

![Figure 1. Flat solar air collector structure diagram (unit: mm)](image)

**Table 1. Physical property parameter table of solar air collector parts**

| Collector parts | Density (kg/m³) | Specific heat capacity (J/(kg·K)) | Thermal Conductivity (W/(m·K)) | Absorption rate | Transmittance | Refractive index |
|-----------------|-----------------|-----------------------------------|--------------------------------|-----------------|---------------|-----------------|
| Glass cover     | 2500            | 840                               | 0.76                           | —               | 0.92          | 1.3             |
| Heat absorption plate | 2719           | 871                               | 202.4                           | 0.95            | —             | —               |
| Asbestos layer  | 2300            | 836                               | 0.073                           | —               | —             | —               |
| Shell /Ribs     | 2719            | 871                               | 202.4                           | 0.95            | —             | —               |
3. Numerical Simulation Calculation

3.1. Boundary Conditions and Initial Conditions
The internal air velocity of the solar air collector is very low, which can be approximated as an incompressible fluid. The physical properties of the air are based on the Boussinesq assumption. The density is 1.18 kg/m$^3$ and the specific heat capacity is 1006.43 J/(kg·K). The air inlet temperature is 290K, the ambient temperature is 288K, and the ambient wind velocity is 2m/s.

The simulation time is 13:00 on November 21, 2019, and the location is Hengyang city, Hunan province, China (112.6 ° E, 26.9 ° N). The solar calculator provided by Fluent is used to calculate the solar radiation intensity and the solar vector direction. The intensity is 768.05W/m$^2$, and the vector direction: x is -0.18162, y is -0.173493, and z is 0.967943. The boundary condition Settings are shown in Table 2.

### Table 2. Boundary condition setting table

| Boundary name | Boundary condition setting |
|---------------|---------------------------|
| Inlet         | Velocity inlet            |
| Outlet        | Pressure outlet            |
| Glass cover   | Mixed boundary conditions, translucent medium, composite heat transfer coefficient of glass cover: $h_g=13.3\text{W/(m}^2\text{K)}$ |
| Heat absorption plate | Coupled heat transfer boundary, opaque medium, emissivity 0.8 |
| Ribs          | Coupled heat transfer boundary |
| Insulation layer | Convection heat transfer boundary, $h_{ins}=8.8\text{W/(m}^2\text{K)}$ |
| Shell         | Coupled heat transfer boundary |

Among them, the outside of the glass cover plate has both natural convection caused by wind and radiation heat transfer with the sky, and its composite convective heat transfer coefficient $h_g$ is calculated by Watmuff empirical formula$[^7]$, as shown in equation (1).

$$h_g=5.7+3.8v_w$$

In the formula: $h_g$ is the composite convective heat transfer coefficient of the glass cover, which includes the radiation and convective heat transfer coefficient between the glass cover plate and the sky and the outside environment; $v_w$ is the wind velocity of the external environment.

3.2. Solution
In Fluent, the steady-state pressure base solver was used for numerical calculation, and the coupling between velocity and pressure base was SIMPLE algorithm. The air in the solar collector is turbulent flow and the fluid is separated near the wall. Therefore, Realizable k-$\varepsilon$ model with advantages in simulating negative pressure gradient flow, fluid separation and complex secondary flow is selected$[^8]$. At the same time, DO radiation model suitable for semi-transparent medium was selected to solve the radiant heat transfer problem of glass cover.

3.3. Evaluation Index of Solar Air Collector

3.3.1. Heat collection efficiency. The heat collection efficiency $\eta$ of the solar air collector is described as the ratio of the energy absorbed by the air in the flow channel to the amount of solar radiation absorbed by the surface of the collector, as shown in the following equations (2) and (3):

$$\eta=\frac{c\rho vi(T_o-T_i)}{AG}$$

$$G=I/(\tau\alpha_e)$$

Where: $c$, $\rho$, and $v_i$ are the specific heat capacity, density, and inlet velocity of air; $T_i$ and $T_o$ are the air inlet and outlet temperatures; $A_i$, $A_c$ are the air inlet area and glass cover lighting area; $G$, $I$ are The
solar radiation intensity of the lighting surface of the heat collector and the surface of the heat absorption plate; \( \tau \) is the transmittance of the glass cover plate 0.92, and \( \alpha \) is the absorption rate of the heat absorption plate 0.95. \((\tau \alpha)_e\) uses the empirical formula proposed by Duffie\[9\], \((\tau \alpha)_e = 1.02\tau \alpha\).

3.3.2. Normalized temperature difference. The normalized temperature difference of the solar collector refers to the average temperature of the collector or the ratio between the difference between the inlet temperature and the ambient temperature and the solar radiation intensity\[10\]. The greater the value is, the higher the air outlet temperature is after the collector is heated. In this paper, the average temperature \( T_m \) of the air in the collector is used to calculate the normalized temperature \( T_m^* \), as shown in equation (4) and (5) below.

\[
T_m = T_i + \frac{\Delta T}{2} \quad (4)
\]

\[
T_m^* = \frac{T_m - T_a}{G} \quad (5)
\]

Where: \( T_m \) is the average temperature of air; \( T_i \) is the air inlet temperature; \( \Delta T \) is the temperature difference between the air outlet and the inlet; \( T_m^* \) is the normalized temperature difference, \((K\cdot m^2)/W\); \( T_a \) is the ambient temperature 288K; \( G \) is the solar radiation intensity on the collector surface, \( G = 768.05W/m^2 \).

4. Design and Analysis of Orthogonal Experiment

4.1. Orthogonal Experimental Design
In this paper, using the orthogonal test method as a guide, nine straight rib solar air collector models were designed and numerically calculated. The length, height, number of ribs and air inlet velocity were analyzed using statistical methods. The significance of these factors on the heat collection efficiency leads to the optimal rib combination corresponding to the highest heat collection efficiency. The factor level of each parameter is shown in Table 3 below.

| Level | Number of ribs (pieces) | Rib height (mm) | Rib length (mm) | Inlet velocity(m/s) |
|-------|------------------------|----------------|----------------|-------------------|
| A     | 5                      | 15             | 1200           | 1                 |
| B     | 7                      | 30             | 1600           | 2                 |
| C     | 9                      | 45             | 2000           | 3                 |

4.2. Orthogonal Result Analysis
In this paper, the interaction between orthogonal factors is not considered, so L9(3^4) orthogonal test scheme is selected. As shown in Table 4 below, a total of 9 working conditions were numerically simulated. The 4 factors of each working condition were determined according to the corresponding test number, and the 3 levels of each factor were guaranteed to occur three times. The other model conditions were set exactly the same. The optimization objectives of this orthogonal test are heat collection efficiency and normalized temperature difference. The results of the orthogonal test are shown in Table 4 below.
Table 4. Orthogonal test parameters and results

| Test number | Factor | Heat collection efficiency | Normalized temperature difference ((K·m²)/W) |
|-------------|--------|----------------------------|---------------------------------------------|
|             | A      | B             | C | D   |                               |                             |
| 1           | 1      | 1          | 1 | 1   | 34.99%                       | 0.00887                     |
| 2           | 1      | 2          | 2 | 2   | 46.89%                       | 0.00690                     |
| 3           | 1      | 3          | 3 | 3   | 54.86%                       | 0.00602                     |
| 4           | 2      | 1          | 2 | 3   | 52.26%                       | 0.00572                     |
| 5           | 2      | 2          | 3 | 1   | 38.32%                       | 0.00967                     |
| 6           | 2      | 3          | 1 | 2   | 48.40%                       | 0.00706                     |
| 7           | 3      | 1          | 3 | 2   | 47.51%                       | 0.00687                     |
| 8           | 3      | 2          | 1 | 3   | 53.74%                       | 0.00594                     |
| 9           | 3      | 3          | 2 | 1   | 40.70%                       | 0.01013                     |

Heat collection efficiency

| k₁ | 45.58% | 44.92% | 45.71% | 38.00% |
| k₂ | 46.33% | 46.32% | 46.62% | 47.60% |
| k₃ | 47.32% | 47.99% | 46.90% | 53.62% |

Range R | 1.74% | 3.07% | 1.19% | 15.62% |

Priority | D>B>A>C

Optimal combination | A₃B₃C₃D₃

Normalized temperature difference

| k₁ | 0.00726 | 0.00715 | 0.00729 | 0.00956 |
| k₂ | 0.00748 | 0.00751 | 0.00758 | 0.00694 |
| k₃ | 0.00765 | 0.00774 | 0.00752 | 0.00589 |

Range R | 0.00039 | 0.00058 | 0.00029 | 0.00367 |

Priority | D>B>A>C

Optimal combination | A₃B₃C₂D₁

From the above table, it can be seen that the significant influence of the four orthogonal factors on the heat collection efficiency and the normalized temperature difference is: inlet velocity > rib height > rib number > rib length. The optimal rib parameter combination based on heat collecting efficiency was the number of ribs 9, the rib height 45mm and the rib length 2000mm.

From the perspective of heat transfer, the heat on the surface of the heat absorbing plate is transferred to the rib in the form of heat conduction, and the heat on the surface of the rib is transferred to the air in the form of convection, so as to increase the convective heat transfer coefficient between the air and the heat absorbing plate to improve the heat collection efficiency. Firstly, the air in the collector is turbulent, and the convective heat transfer coefficient is positively correlated with the Reynolds number Re. The influence of the change of air velocity is more significant than the change of characteristic length, so the change of air inlet velocity has the greatest impact on the test index. Secondly, increase the number of ribs in a certain range, obviously increase the heat transfer area and improve the convective heat transfer coefficient. According to the theory of rib heat transfer, in a certain range, the rib efficiency after increasing the rib height is better than that after increasing the rib width (namely, the rib length).

According to Table 4, when other parameters remain unchanged, the heat collection efficiency increases with the increase of air inlet velocity. On the contrary, the normalized temperature difference decreases with the increase of air inlet velocity. Therefore, this paper will explore the influence of air inlet velocity change on heat collection efficiency and normalized temperature difference by using the model optimized by ribbed structural parameters.
5. Working Condition Optimization Analysis

5.1. Temperature Cloud Diagram of the Central Section of the Collector
The number of ribs (9), the height of ribs (45mm) and the length of ribs (2000mm) were taken as rib parameters to establish a model for numerical simulation calculation. Other conditions remained unchanged. The following Figure 2–6 shows the temperature cloud diagram of the central section of collector Z=0.0625m with inlet velocity of 1m/s, 2m/s, 3m/s, 4m/s and 5m/s respectively.

As can be seen from the cloud images, the air temperature in the central section of the collector gradually increases from left to right, which is the result of heat exchange between the air flowing through the collector and the heat absorbing plate and ribs. According to the convective heat transfer formula, the heat flow density is directly proportional to the convective heat transfer coefficient, which increases with the increase of the velocity, but the solar radiation heat absorbed by the heat absorbing plate is constant, so when the inlet velocity increases to a certain value, the air outlet temperature gradually tends to be stable, and the thermal efficiency of the collector is no longer increased. By comparing the cloud images, it can be found that with the increase of air velocity, the outlet temperature of the collector decreases gradually, and the temperature of the central section of the collector is also close to the inlet temperature.

5.2. Thermal Performance Index Analysis
Based on the above results, this article will further explore the characteristics of the effect of air inlet velocity changes on the heat collection efficiency and normalized temperature difference of straight rib solar air collectors and ordinary solar air collectors. The model and numerical simulation of a straight rib solar air collector were established with the best rib parameters—the number of ribs 9, the...
height of the ribs 45mm and the length of the ribs 2000mm, and compared with the ordinary solar air collector without ribs. "velocity-heat collection efficiency/normalized temperature difference characteristic curve" is shown in Figure 7.

![Figure 7. Characteristic curve](image)

As shown in Figure 7, it is clear that the heat collection efficiency increases with the increase of the air inlet velocity, while the normalized temperature difference decreases with the increase of the air inlet velocity, the two thermal performance index curves gradually slow down and tend to a constant value. Obviously, when \( v = 1 \text{m/s} \), the difference between the thermal performance indexes of the two collector models is the largest. It can be speculated from the previous research results: When the air inlet velocity is smaller, the velocity has less influence on the thermal performance of solar air collector. On the contrary, the influence of rib parameters on thermal performance index is more significant. At the same air inlet velocity, the heat collection efficiency and normalized temperature difference of the straight rib solar air collector are higher than that of the ordinary solar air collector, and the average value is calculated to be 3.98% and 0.00052 (K·m²)/W respectively. Therefore, compared with ordinary solar air collectors, the straight rib solar air collectors increase the heat collection efficiency and increase the air outlet temperature to a certain extent.

6. Conclusion
(1) The significant influence of the four orthogonal factors on the heat collection efficiency and the normalized temperature difference is as follows: the inlet velocity > rib height > rib number > rib length. The optimal rib parameter combination based on heat collecting efficiency is: the number of ribs is 9, the rib height is 45mm and the rib length is 2000mm.
(2) When other parameters are unchanged, the heat collection efficiency increases with the air inlet velocity. On the contrary, the normalized temperature difference decreases with the increase of air inlet velocity.
(3) Straight rib solar air collector based on the best combination of rib parameters compared with ordinary solar air collector, the average heat collection efficiency and normalized temperature difference are 3.98% and 0.00052 (K·m²)/W higher, respectively.

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