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Numerical Analysis and Comparison of Thermal Performance of a New Homogeneous Parabolic Trough Solar Receiver

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ABSTRACT

In order to limit the negative effects of large circumferential thermal gradient in parabolic trough collectors (PTCs), enhancing the flow and heat exchange of the working fluid by embedding spiral slices in the absorber tube is a promising method. The absorber tube with the secondary reflector (SRAT) was proposed which solar flux is significantly uniformly distributed. The thermodynamic performance of the embed spiral slices absorber tube (ESAT) is analyzed and compared with that of the SRAT by building a thermal-structure coupled model. It can be found that compared to traditional absorber tubes, the circumferential temperature difference of ESAT with different pitches is reduced by 13%-53%, and the maximum thermal stress displacement is reduced by 32%-50%, which is much less than that of SRAT. However, under the simulated conditions, ESAT eliminates the high-temperature boundary zone of the working fluid and the useful energy of the working fluid increases by about 7% compared to SRAT, but the outlet pressure loss increases. This research can improve the energy efficiency, economy, and reliability of parabolic trough collectors.

Keywords: homogeneous, spiral slice, secondary-mirror, thermal deformation

NONMENCLATURE

Abbreviations	
DNI	direct normal irradiation
PTCs	parabolic trough collectors
SRAT	absorber tube with the secondary reflector
ESAT	embed spiral slices absorber tube

1. INTRODUCTION

Solar energy plays an important role in solving global warming and energy crisis[1]. Concentrated solar thermal power generation technology is an important approach to realize large-scale solar thermal utilization, among which parabolic trough collectors (PTCs) are the most mature technology[2]. Due to the linear focusing characteristics of the PTCs, the absorber tube has a high circumferential temperature gradient, and the thermal stress is uneven, bending absorber tubes and even damaging external vacuum glass tubes. In addition, the excessively local high temperature can damage the selective coating and increase radiant heat loss. Improving the homogeneity of the temperature distribution of the absorber tube surface is important for improving the operating efficiency and safety of the collector[3].

Wang et al.[4] numerically studied the circumferential temperature difference and deformation of the absorber tube, using Al_2O_3/synthetic oil nanofluid as a working fluid. The results show that nanofluids improve heat transfer performance, avoid high temperature gradients, and reduce thermal stress and thermal deformation. Hui et al.[5] devised three types of novel inclined curved-twisted baffles. The simulation results show that the baffles can generate longitudinal swirl to reduce the wall temperature difference, and the average efficiency of the absorption tube is improved. Gong et al.[6] described an adaptive design method for the secondary reflector (SR) of the large-aperture parabolic trough condenser. The resulting SR is composed of many parabolas with different focal lengths. The focal points of all parabolas are on the focal point of the main reflector to ensure that the unabsorbed light hits the absorber tube after being reflected by the SR.

In order to compare the thermodynamic properties of the absorber tube under different methods, this research proposes the insertion of a spiral pieces in the absorber tube and establishes a three-dimensional heat-stress coupling model. The thermodynamic properties of the new absorber tube under different pitches are analyzed and compared with direct uniform solar flux PTCs.

2. MODEL AND SIMULATION

2.1 New PTCs structure

The parabolic trough collector consists of a parabolic reflector and an absorber tube located at the focal line of the parabola. The light from the sun is concentrated on the bottom of the collector after being reflected by the reflector, and there is almost no reflected light from the top of the collector. The distribution of solar flux is shown in Fig. 1. The average solar flux distribution on the light-receiving side of the collector tube is much larger than that on the backlight side, resulting in a significant circumferential temperature gradient of the absorber tube.

In order to increase the flow and heat exchange of the fluid inside the tube to reduce the circumferential temperature gradient, the ESAT model is proposed, as shown in Fig. 3. The length of the spiral slice is the same as the length of the absorber tube, the spiral slice is tangent to the inner diameter of the absorber tube, and the width *h* is equal to the inner diameter of the absorber tube. The pitch *S* is expressed as the lateral distance required for the fluid to make one revolution along the spiral. Fig. 4 shows the spiral model with different pitches.

Fig. 2(a) shows the structure of the double-plane secondary reflector. The design process and geometric parameters of the double-plane secondary reflector can be found in the relevant research. It directly homogenizes the solar flux to eliminate the circumferential temperature difference of the absorber tube. After installing the secondary mirror, Fig. 2(b) shows the solar flux distribution of the absorber tube with the same size as Fig. 1. The solar flux in Fig. 2(b) is used as the heat flow boundary of the conventional absorber tube (CAT-2), serving as the experimental group compared with ESAT.

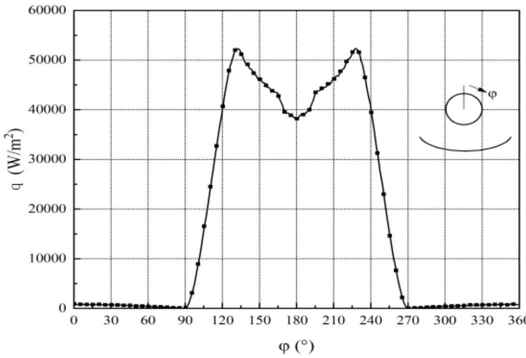
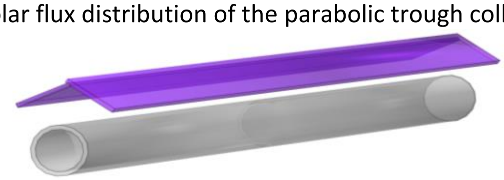
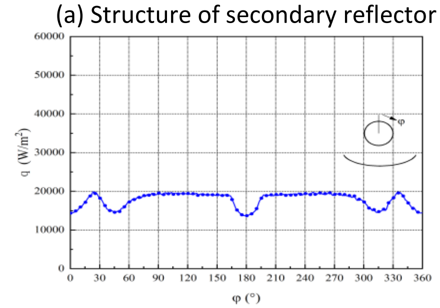


Fig. 1. Solar flux distribution of the parabolic trough collector



(a) Structure of secondary reflector



(b) distribution of solar flux

Fig. 2 Parabolic trough collector with secondary mirror

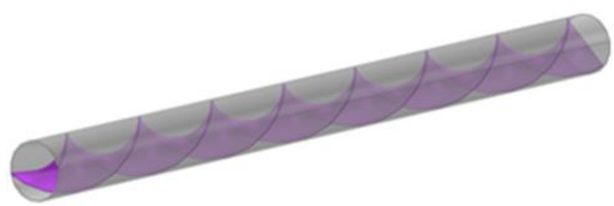


Fig. 3 Model of built-in spiral slice in the absorber tube

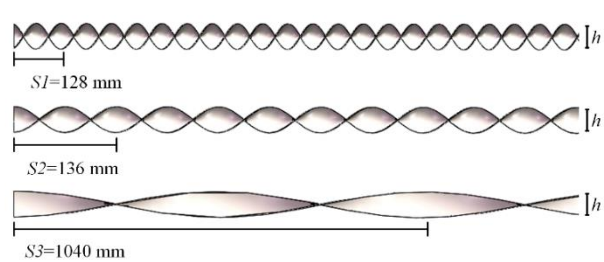


Fig. 4 Spiral pieces of diffent pitch

Table 1 basic parameters of PTCs

Item	Value/m
width of reflector	5
focal of reflector	1.84

Cover outer diameter	0.115
Cover inner diameter	0.112
Absorber outer diameter	0.070
Absorber inner diameter	0.066
The length of absorber	7.8
Reflector reflectance	0.93
Cover transmittance	0.95
Coating absorbance	0.95
Coating emissivity	0.00042T(K)-0.0995

2.2 Heat transfer model

The working fluid is Syltherm 800 oil[7].The oil is turbulent in the absorber tube, and the mathematical models involve:

The continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \boldsymbol{u}) = 0 \tag{1}$$

and the momentum equation:

$$\rho \frac{\partial \boldsymbol{u}}{\partial t} + \rho \boldsymbol{u} \cdot \nabla \boldsymbol{u} = -\nabla p + \nabla \cdot \left(\mu \left(\nabla \boldsymbol{u} + (\nabla \mathbf{u})^{\boldsymbol{T}} \right) - \frac{\mathbf{2}}{\mathbf{3}} \mu (\nabla \cdot \mathbf{u}) \mathbf{I} \right) + \mathbf{F} \tag{2}$$

Energy equations:

$$\rho C_p \left(\frac{\partial T}{\partial t} + (\mathbf{u} \cdot \nabla) T \right) = -(\nabla \cdot \mathbf{q}) + \tau : \mathbf{S} - \frac{T}{\rho} \frac{\partial \rho}{\partial T} \bigg|_p \left(\frac{\partial p}{\partial t} + (\mathbf{u} \cdot \nabla) p \right) + Q \tag{3}$$

where ρ is the total density; $\boldsymbol{u}$ is the velocity vector; p is pressure; τ is the viscous stress tensor; **F** is the volume force vector; C_p is the specific heat capacity at constant pressure; **q** is the heat flux vector; Q contains the heat sources.

The general governing equations are discretized by finite elements method (FEM). A typical working condition is selected to investigate the thermal performances of PTCs. The oil inlet velocity, the DNI and the oil inlet temperature are 3 m/s, 1000 W/m^2 and 573.5 K. The flow boundary conditions are set as follows[8]:

(1) Inlet: u_z=V_{in}, u_x = u_y =0;

(2) Outlet: fully developed assumption;

(3) The solar fluxes distribution shown in Fig. 1 and Fig. 2(b) are boundary heat fluxes, respectively:

Table 2 boundary heat fluxes

Type of the absorber tube	Solar flux distribution
CAT-1	Fig. 1
ESAT-S1	Fig. 1
ESAT-S2	Fig. 1
ESAT-S3	Fig. 1
CAT-2	Fig. 2(b)

2.3 Model validation

The vacuum cover adopts structured grid, and the fluid and metal part adopt unstructured tetrahedral grid. When the number of grids increases from 0.85 million to 1.5 million, the change in the average friction coefficient and the average Nusselt number is almost negligible. Therefore, the grid with 0.85 million nodes is the most effective grid in terms of accuracy and computational time-saving.

In order to verify the simulation method of the coupled heat transfer process, simulations were carried out according to the experimental conditions of Dudley[9]. The relative errors between the average outlet temperature and thermal efficiency of the simulation results and the experimental results are within 0.18% and 4.46%, respectively, indicating that the proposed numerical model is feasible.

3. RESULTS AND DISCUSTION

3.1 Performance of absorber tube wall

The wall temperature distribution of the absorber tube is almost identical along the flow direction. Fig. 5 shows the circumferential wall temperature distribution of the absorber tube at z=3.9 m. Compared to traditional absorber tubes, the maximum surface temperature of absorber tubes with built-in spirals of 128 mm, 260 mm, and 1040 mm are reduced by 8 K, 4 K, and 2 K, while the maximum temperature of direct uniform solar flux absorber tubes is reduced by 10K. The circumferential temperature differences of absorber tubes with different pitches are 7K, 11K, and 13K, which are respectively reduced by 53%, 27%, and 13% compared to traditional absorber tubes. While the circumferential temperature difference of the direct uniform solar flux absorber tube is only 1K, which is reduced 93%. This is because the spirals change the flow field of the working fluid and enhance the convective heat transfer between the fluid and the inner wall of the absorber tube. The heat on the light-receiving side of the absorber tube is transferred to the backlight-side through the working fluid. The smaller the pitch, the stronger the convective heat transfer of the fluid, and the more obvious the uniform effect of the spiral.

The thermal boundary of the absorber tube that directly uniforms the solar flux is uniform, and its light-receiving-side and the backlight-side are evenly heated, and there is almost no circumferential temperature difference on the wall surface of the absorption tube. Differently, when the built-in spiral fin absorber tube works, the convective heat transfer is affected by conditions such as flow velocity, and the circumferential temperature difference of the absorber tube cannot be eliminated.

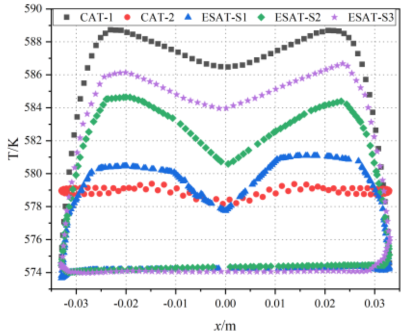


Fig. 5 the temperature distribution of absorber tube

3.2 *Characteristics of working fluid*

Fig. 6 shows the radial cross-sectional temperature distribution of the working fluid along the flow direction. The temperature of the working fluid increases in the direction of flow as the fluid continues to absorb heat from the boundary. The cross-sectional temperature difference of the working fluid in ESAT is eliminated, and the maximum temperature is reduced. Compared with SRAT, the boundary temperature is the same as the intermediate temperature. The reason is that the flow changes after the built-in spirals, the difference in heat transfer between the backlight side and the light receiving side of the fluid is reduced, and the internal heat transfer is enhanced.

With the decrease of the pitch, the average cross-sectional temperature of the fluid in ESAT-S1 is higher than that of ESAT-S3 and ESAT-S2. The larger the pitch, the high temperature area appears at the fluid boundary of ESAT-S3 and ESAT-S2. The possible reason is that the smaller the pitch, the smaller the vortex generated per unit time, and the more sufficient heat exchange.

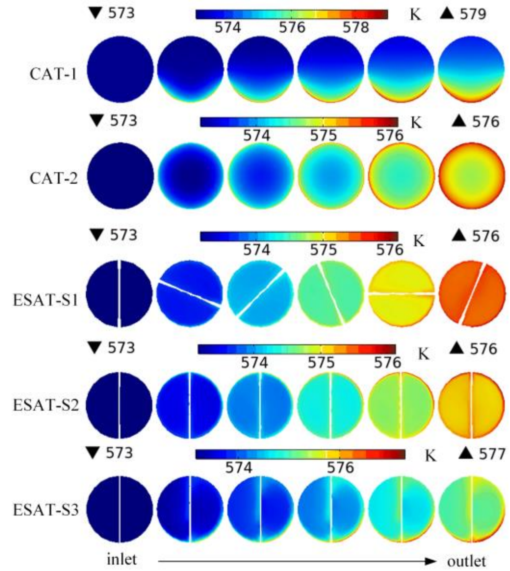


Fig. 6 cross-section temperature distribution of working fluid

3.3 *Energy efficiency analysis*

Fig. 7 shows the useful work and pressure loss at the outlet of the working fluid. With the built-in spiral, the useful work of the working fluid is increased by about 7% compared to the working fluid in the CAT. This is because the spirals change the flow field and center of the fluid, and the convective heat transfer is enhanced, and more heat flow from the wall is absorbed as the fluid flows. Compared to the CAT working fluid, the pressure loss of the fluid in ESAT increases. The smaller the pitch, the greater the pressure loss. The pressure loss of the working fluid in ESAT-S3 is reduced by 96% compared to ESAT-S1. The possible reason is that the smaller the pitch, the greater the flow rate of the fluid, and the resistance loss along the way is proportional to the square of the speed, resulting in an increase in the resistance loss of the fluid along the way.

The results show that the spiral fins change the flow field and enhance the convective heat transfer between the fluid and the tube wall. The increase of the fluid velocity leads to the increase of the pressure loss. Changing the flow field but keeping the fluid velocity constant is the best choice for increasing the useful work of the fluid without the increase of the pressure loss along the way, which is a more effective structure.

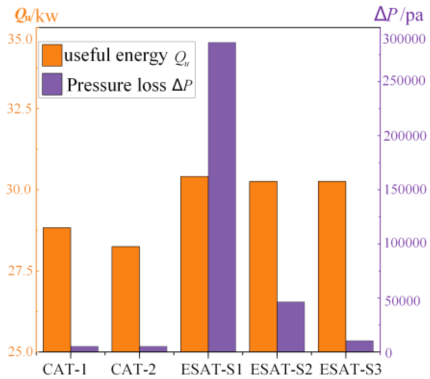


Fig. 7 useful energy and pressure loss of the working fluid

3.4 The results of thermal-stress coupling

Due to the uneven distribution of solar flux on the surface of the absorption tube, the temperature difference between the light-receiving side and the backlight side is obvious, which causes mutual restraint between internal parts, making it impossible to expand and contract completely freely, and results in thermal stress and deformation. The thermal stress displacement of different absorber tubes in the direction parallel to sunlight is shown in Fig. 8. The maximum thermal stress displacement of CAT-1 is 10.8mm, the maximum thermal stress displacement of CAT-2 is 0.2mm, and the maximum thermal stress displacements of ESAT are 5.5mm, 7mm and 7.5mm, respectively. As the pitch becomes smaller, the maximum thermal stress displacement of the absorber tube becomes smaller. This is because the boundary heat flux of CAT-2 is uniform, the circumferential temperature gradient of the absorption tube is eliminated, and the thermal stress is eliminated. The smaller the pitch of ESAT, the less obvious the circumferential temperature difference of the absorber tube.

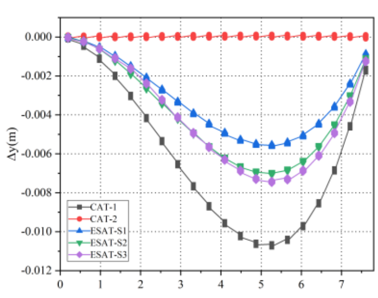


Fig. 8 the displacement of the absorber tubes in the y-direction

4. CONCLUSIONS

Improving the surface temperature gradient of trough solar absorber tubes is of great significance for improving the operating efficiency and safety of collectors. In order to enhance the flow and heat transfer of the working fluid to reduce the circumferential temperature difference of absorber tubes, this study inserts a spiral sheet into the absorber tube. The installation of a secondary reflector on the surface of the heat absorption tube to directly homogenize the solar flux is the most effective method. In order to compare the heat transfer characteristics of the built-in helical absorber tube and the direct uniform solar flux absorber tube, heat-stress coupling model with different heat flow boundaries were developed. The temperature distribution, the characteristics of the working fluid and the useful work are analyzed. The main analysis results are summarized as follows:

(1) Compared to traditional absorber tubes, the reduced circumferential temperature difference of built-in spiral absorber tube is reflected in a lower temperature on the light-receiving side and almost unchanged temperature on the backlight side, and the smaller the pitch, the lower the maximum temperature. The circumferential temperature difference of the built-in spiral fin absorption tube cannot be eliminated compared to direct uniform heat flux absorption tubes.

(2) The cross-sectional temperature difference of the working fluid in the built-in spiral absorber tube is eliminated. Compared to uniform heat flux absorber tubes, the boundary high-temperature area is reduced, and the boundary and center of the fluid are at the same temperature. The smaller the pitch, the more obvious this effect becomes.

(3) Under simulated conditions, the average useful work of the outlet of the working fluid in the built-in spiral absorber tube is increased by about 7% compared to the direct uniform heat flux absorber tube, and the average pressure loss of the outlet is increased by 50%-98%. The smaller the pitch, the greater the pressure loss. The pressure loss of a screw with a pitch of 128mmm is 96% greater than that of a screw with a pitch of 1040mm.

(4) The thermal stress displacement of the built-in spiral absorber tube is reduced. Under the design conditions, compared to traditional absorber tubes, the thermal stress displacement of the absorber tube is reduced by 50%, 36%, and 32%, respectively, when the pitches are 128mm, 260mm, and 1040mm. The thermal stress displacement of the direct uniform heat flux absorption tube is reduced by 98%.

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