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Numerical analysis of flow and heat transfer characteristics in solar chimney power plants with energy storage layer

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ABSTRACT

Numerical simulations have been performed to analyze the characteristics of heat transfer and air flow in the solar chimney power plant system with an energy storage layer. Different mathematical models for the collector, the chimney and the energy storage layer have been established, and the effect of solar radiation on the heat storage characteristic of the energy storage layer has been analyzed. The numerical simulation results show that: (1) the heat storage ratio of the energy storage layer decreases firstly and then increases with the solar radiation increasing from 200 W/m^2 to 800 W/m^2; (2) the relative static pressure decreases while the velocity increases significantly inside the system with the increase of solar radiation; (3) the average temperature of the chimney outlet and the energy storage layer may increase significantly with the increase of solar radiation. In addition, the temperature gradient of the storage medium may increase, which results in an increase of energy loss from the bottom of the energy storage layer.

1. Introduction

The solar chimney power plant system, which consists of four major components, collector, chimney, turbine and energy storage layer, was first proposed in the late 1970s by Professor Jörg Schlaich and tested with a prototype model in Manzanares, Spain, in the early 1980s [1,2]. Air underneath the low circular transparent glass or film canopy open at the circumference is heated by solar radiation. The canopy and the surface of the energy storage layer below form a hot air collector. The chimney, a vertical tower tube with large air inlets at its base, stands in the center of the collector. The joint between the collector and the chimney is air tight. The wind turbine is installed at the bottom of the chimney where there is a large pressure difference. For large scale solar chimney systems, there may be several wind turbines inside as it is difficult, with current technology, to produce wind turbines with rated load over 10 MW. As the density of the hot air inside the system is less than that of the cold air in the environment at the same altitude, natural convection, affected by buoyancy that acts as driving force comes into existence. The energy of the air flow is converted into mechanical energy by pressure staged wind turbines at the base of the tower, and ultimately into electrical energy by electric generators coupled to the turbines.

As solar chimney power plant systems could make significant contributions to the energy supplies of those countries where there is plenty of desert land that is not being utilized and sunlight is available in Africa, Asia and Oceania, researchers have made many reports on this technology in recent decades. Haaf et al. [1,2] provided fundamental studies for the Spanish prototype in which the energy balance, design criteria and cost analysis were discussed and reported preliminary test results of the solar chimney power plant. Krisst [3] and Kulunk [4] demonstrated different types of small scale solar chimney devices with power outputs not more than 10 W. Pasumarthi and Sherif [5,6] developed a mathematical model to study the effect of various environment conditions and geometry on the flow and heat transfer characteristics and output of the solar chimney and also developed three different models in Florida and reported the experimental data to assess the viability of the solar chimney concept. Lodhi [7] presented a comprehensive analysis of the chimney effect, power production and efficiency and estimated the cost of the solar chimney power plant set up in developing nations. Bernardes et al. [8] presented a theoretical analysis of a solar chimney with natural laminar convection in steady state. Gannon and Backström [9] presented an air standard cycle analysis of the solar chimney power plant for calculation of the limiting performance, efficiency and the relationships between the main variables, including chimney friction, system, turbine and exit kinetic energy losses. Gannon and Backström [10] presented an experimental investigation of the performance of a solar chimney turbine. The measured results showed that the solar chimney turbine presented has a total to total efficiency of 85–90% and a total to static efficiency of 77–80% over the design range. Later, Backström and Gannon [11] presented analytical equations in

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Nomenclature

A	collector area, m^2
H	convection heat transfer coefficient, $W/(m^2 K)$
T	temperature, K
u_d, v_d	Darcy velocity of porous media, m/s
d_b	particle diameter of porous layer, m

Subscripts

G	surface of energy storage layer
A	air in collector
E	environment
stor	energy storage layer
S	solid matrix of energy storage layer
M	apparent value of energy storage layer

Greek symbols

σ	Stefan–Boltzmann constant
α	absorptivity of collector canopy to solar radiation energy
β	thermal expansion coefficient, 1/K
φ	porosity
μ	dynamic viscosity, kg/(m s)
λ	thermal conductivity, W/(m K)

terms of turbine flow and load coefficient and degree of reaction to express the influence of each coefficient on the turbine efficiency. Bernardes and Weinrebe [12] developed a thermal and technical analysis to estimate the power output and examine the effect of various ambient conditions and structural dimensions on the power output. Pastohr et al. [13] conducted a numerical simulation to improve the description of the operation mode and efficiency by coupling all parts of the solar chimney power plant including the ground, collector, chimney and turbine. Schlaich and Weinrebe [14] presented the theory, practical experience and economy of the solar chimney power plant to give a guide for design of 200 MW commercial solar chimney power plant systems. Liu et al. [15] conducted a numerical simulation for the MW grade solar chimney power plant, presenting the influences of pressure drop across the turbine on the draft and the power output of the system. Schlaich et al. [16] presented a simplified theory, some practical experience results and a detailed economic analysis of solar chimneys for the design of commercial solar chimney power plant systems like the one being planned for Australia. Bilgen and Rheault [17] designed a solar chimney system for power production at high latitudes and evaluated its performance. Pretorius and Kröger [18] evaluated the influence of a developed convective heat transfer equation, more accurate turbine inlet loss coefficient, quality collector roof glass and various types of soil on the performance of a large scale solar chimney power plant. Ming et al. [19] presented a mathematical model to evaluate the relative static pressure and driving force of the solar chimney power plant system and verified the model with numerical simulations. Later, Ming et al. [20] developed a comprehensive model to evaluate the performance of a solar chimney power plant system in which the effects of various parameters on the relative static pressure, driving force, power output and efficiency have been further investigated.

It is clear that the energy storage layer of solar chimney power plant systems is a significant part without which the whole system could not operate during the night. Part of the solar radiation is absorbed in the energy storage layer during the daytime and released during the night or days with cloudy weather. Seldom have simulation studies or experiments concentrated on the performances of the systems with energy storage layer devices. Numerical simulation for the MW grade solar chimney power plant conducted by Liu et al. [15] is insufficient for the design of a commercial large scale solar chimney power plant with energy storage layer that can supply power output continuously all year round. In addition, the energy storage layer system can be regarded as porous media as the air flows inside the solid matrix. Pastohr et al. [13] presented a numerical simulation result in which the storage layer was regarded as solid. In this article, conjugate numerical simulations of the energy storage layer, the collector and the chimney have been conducted, and the characteristics of the heat storage of the energy storage layer, and the flow and heat transfer in the whole system have been studied.

2. Numerical model

2.1. Physical model

In this study, the Spanish prototype shown in Fig. 1 is selected as a physical model for the numerical simulation. The prototype has a chimney of 200 m height and 5 m radius and a collector of 122 m radius and 2 m height. The solid matrix of the energy storage layer is soil or gravel with large heat capacity.

2.2. Mathematical model in the collector and chimney

In natural convection, the strength of the buoyancy induced flow is measured by the Rayleigh number defined as follows:

$$Ra = \frac{g\beta(T_h - T_c)L^3}{a\nu} \tag{1}$$

where T_h, T_c is the maximum and minimum temperature of the system, respectively. L and a are the collector height and the thermal diffusivity, respectively. Boussinesq's approximation is applied to account for the air density variation. In the collector and the chimney, analysis shows that $Ra > 10^{10}$, and thereby, air flow in these

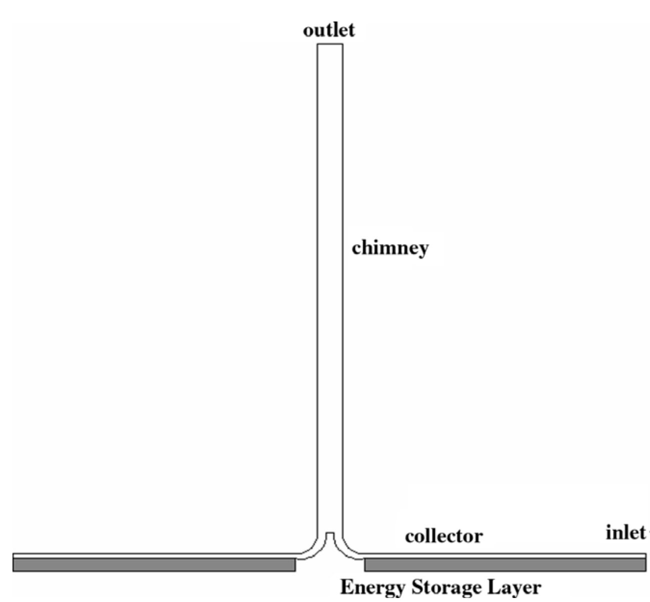

Fig. 1. The schematic drawing of the solar chimney power plant system.

regions may be turbulent flow. Accordingly, the continuity equation, Navier–Stokes equations, energy equation and κ–ε equations are shown as follows:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0 \tag{2}$$

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}=\rho g\beta(T-T_\infty)+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{3}$$

$$\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{4}$$

$$\frac{\partial(\rho cT)}{\partial t}+\frac{\partial(\rho cuT)}{\partial x}+\frac{\partial(\rho cvT)}{\partial y}=\lambda\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{5}$$

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right)+G_k+G_b-\rho\varepsilon+S_k \tag{6}$$

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}+S_\varepsilon \tag{7}$$

where G_κ represents the generation of turbulence kinetic energy due to the mean velocity gradients defined as: $G_\kappa=-\rho\overline{u_i' u_j'}\frac{\partial u_j}{\partial x_i}$; G_b is the generation of turbulence kinetic energy due to buoyancy; σ_κ and σ_ε are the turbulent Prandtl numbers for κ and ε, respectively; and β is the thermal expansion coefficient: $\beta\approx 1/T$.

2.3. Mathematical model in the energy storage layer

It is necessary to take into account the collector, the chimney and the energy storage layer as a whole system, but the flow and heat transfer in the energy storage layer may be very complicated. As the material used for energy storage can be regarded as porous media, the Brinkman–Forchheimer Extended Darcy model [21] is used in this paper to describe the flow and heat transfer characteristics in the convective porous layer, which can be expressed as follows:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_d)}{\partial x}+\frac{\partial(\rho v_d)}{\partial y}=0 \tag{8}$$

$$\frac{\rho}{\varphi}\frac{\partial u_d}{\partial t}+\frac{\rho}{\varphi^2}\left(u_d\frac{\partial u_d}{\partial x}+v_d\frac{\partial u_d}{\partial y}\right)=-\frac{\partial p_r}{\partial x}+\frac{\partial}{\partial x}\left(\mu_m\frac{\partial u_d}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_m\frac{\partial u_d}{\partial y}\right)-\left(\frac{\mu}{K}+\frac{\rho C}{\sqrt{K}}\mid u_d\mid\right)u_d+\rho g\beta(T-T_\infty) \tag{9}$$

$$\frac{\rho}{\varphi}\frac{\partial v_d}{\partial t}+\frac{\rho}{\varphi^2}\left(u_d\frac{\partial v_d}{\partial x}+v_d\frac{\partial v_d}{\partial y}\right)=-\frac{\partial p_r}{\partial y}+\frac{\partial}{\partial x}\left(\mu_m\frac{\partial v_d}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_m\frac{\partial v_d}{\partial y}\right)-\left(\frac{\mu}{K}+\frac{\rho C}{\sqrt{K}}\mid v_d\mid\right)v_d \tag{10}$$

$$\rho c\left(\frac{\partial T}{\partial t}+u_d\frac{\partial T}{\partial x}+v_d\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial x}\left(\lambda_m\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda_m\frac{\partial T}{\partial y}\right) \tag{11}$$

where u_d and v_d the are Darcy velocities in the porous media, φ, μ_m and λ_m are the porosity, apparent dynamic viscosity and apparent thermal conductivity of the porous medium, respectively: $\lambda_m=(1-\varphi)\lambda_s+\varphi\,\lambda_a$, $\mu_m=\mu/\varphi$, λ_s and λ_a are the thermal conductivities of the solid and air in the energy storage layer, respectively. K and C are the permeability of the porous absorber and inertia coefficient, respectively, which are defined as follows.

$$K=d_b^2\varphi^3/(175(1-\varphi)^2) \tag{12}$$

$$C=1.75\varphi^{-1.5}/\sqrt{175} \tag{13}$$

where d_b is the particle diameter of the porous layer.

2.4. Boundary conditions and solution method

(1) Heat balance of the glass in the collector

$$Q_{g,air}+Q_{g,e}+Q_{g,stor}+Q_{g,sky}+\alpha Q_{solar}=0 \tag{14}$$

where $Q_{g,air}$ is the convection heat transfer between the surface and the air inside the collector: $Q_{g,air}=A_g h_{g,air}(T_g-T_a)$. $Q_{g,e}$ is the convection heat transfer between the collector surface and the environment, $Q_{g,e}=A_g h_{g,e}(T_g-T_e)$. $Q_{g,stor}$ is the radiation heat transfer between the surface of the canopy and the surface of the collector, $Q_{g,stor}=A_g\sigma(T_g^4-T_{stor}^4)$. $Q_{g,sky}$ is the radiation heat transfer between the collector canopy and the sky: $Q_{g,sky}=A_g\sigma(T_g^4-T_{sky}^4)$. Q_{solar} is the heat transfer of the collector from the sun.

(2) Heat balance of the energy storage layer surface

$$Q_{stor,air}+Q_{stor,g}+Q_{stor,down}+\eta\tau Q_{solar}=0 \tag{15}$$

Apparently, the radiation heat transfer between the energy storage layer surface and the air fluid inside the collector can not be neglected as the temperature of the energy storage layer surface is comparatively higher than that of the other parts of the system. Thereby, $Q_{stor,air}$ is the total heat transfer between the energy storage layer surface and the air fluid inside the collector: $Q_{stor,air}=A_{stor}h_{stor,air}(T_{stor}-T_{air})$. $h_{stor,air}$ is the total heat transfer coefficient between the energy storage layer surface and the air fluid inside the collector, $h_{stor,air}=h+\sigma(T_{stor}+T_{air})(T_{stor}^2+T_{air}^2)$ where h is the convection heat transfer coefficient between the energy storage layer surface and the air fluid inside the collector. $Q_{stor,g}$ is the radiation heat transfer between the energy storage layer surface and the collector canopy: $Q_{stor,g}=-Q_{g,stor}=A_g\sigma(T_{stor}^4-T_g^4)$. $Q_{stor,down}$ is the heat transfer from the energy storage layer surface inside the porous media. Neglected the convection heat transfer inside the porous media, $Q_{stor,down}$ can be computed by Fourier's law: $Q_{stor,down}=-A_{stor}\lambda_m\frac{dT}{dx}$, where λ_m is the apparent heat conduction, τ is the transmissivity of the collector canopy to the solar radiation. Other conditions for the energy storage layer surface are as follows:

$$u|_{x=x_0^+}=u|_{x=x_0^-},\quad v|_{x=x_0^+}=v|_{x=x_0^-},\quad p|_{x=x_0^+}=p|_{x=x_0^-} \tag{16}$$

$$\mu_m\left(\frac{\partial u_d}{\partial y}+\frac{\partial v_d}{\partial x}\right)\bigg|_{x=x_0^+}=\mu_m\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)\bigg|_{x=x_0^-} \tag{17}$$

(3) Boundary conditions for inside and outside the energy storage layer

$$\frac{\partial T}{\partial y}=0, u_d=0, v_d=0 \tag{18}$$

(5) Boundary conditions for the chimney wall

$$\frac{\partial T}{\partial x}=0, u=0, v=0 \tag{19}$$

(6) Boundary conditions for the collector outletAccording to the analysis by Ming et al. [20], the relative static pressure of the collector inlet is 0, and the temperature can be the environment temperature:

$$p_{r,inlet}=0, T_{inlet}=T_e \tag{20}$$

The boundary condition for the bottom of the energy storage layer can be selected as a constant temperature condition as the temperature at this place varies only slightly.

3. Numerical procedure

A standard k–ε model is selected to describe the fluid flow inside the collector and the chimney, and the Brinkman–Forchheimer

Extended Darcy model is used to describe the flow in the energy storage layer shown above. The standard near wall function is used to deal with the fluid flow near the wall. In addition, SIMPLE arithmetic is used to describe the pressure–velocity coupling, and the QUICK discretization scheme is selected for the momentum and energy equations. The constant of these models can be found in Ref. [22].

For simulation, the environment temperature is 293.15 K, the transmissivity of the glass is 0.89. For comparison, soil and gravel are used as the materials of the solid matrix of the energy storage layer. The properties of soil are as flows: $\rho_{soil} = 1700\ kg/m^3$, $c_{p,soil} = 2016\ J/(kg\ K)$, $\lambda_{soil} = 0.78\ W/(m\ K)$. The properties of gravel are: $\rho_{gravel} = 2555\ kg/m^3$, $c_{p,gravel} = 814.8\ J/(kg\ K)$, $\lambda_{gravel} = 2.00\ W/(m\ K)$. The absorptivity of the surface and the porosity of the energy storage layer are selected as 0.9 and 0.6, respectively.

4. Results and discussion

Solar energy is converted into heat energy when the solar insolation directly irradiates the surface of the energy storage layer through the transparent canopy, thereby the temperature of the surface of the energy storage layer rises significantly. On the other hand, convection heat transfer between the air inside the collector and the surface of the energy storage layer occurs, which can also make the air temperature increase notably. It should be noted that only a small part of the solar energy is transferred to the air inside the collector, with the rest entering the energy storage layer and being stored inside as heat energy.

Fig. 2 shows the effect of different amounts of solar radiation on the heat storage ratio in the energy storage layer. It is obvious that the heat storage ratio, with the solar radiation changing in from 200 W/m^2 to 800 W/m^2, surpasses 80%, accompanied with a minimum value of the heat storage radio when the solar radiation is about 600 W/m^2, which means that only a small part of the solar radiation is transferred to the air from the solar radiation.

With the increase of the solar radiation on the surface of the energy storage layer, the heat storage capability of the energy storage layer might be seen in the following three cases. In the first place, the temperature of the whole energy storage layer is comparatively low as the solar radiation is weak, and air passes very slowly through the collector, which results in a small convective heat transfer coefficient; thus, a small part of the energy transfers to the air inside the collector, and a larger part of the solar radiation is absorbed by the energy storage layer in increasing its bulk temperature. In the second place, the bulk temperature of the energy storage layer, with the increase of the solar radiation, increases remarkably, and the temperature difference between the surface of the energy storage layer and the air inside the collector also increases notably, which is followed by an enhancement of convection heat transfer between the energy storage layer and the air inside the collector. The air temperature inside the collector may thereby increase, which results in a decrease of the air density and a notable increase of buoyancy and air velocity. This, in turn, enhances the convection heat transfer between the energy storage layer and the air inside the collector, and an increased amount of solar radiation is ultimately transferred to the air flow. By this way, the energy storage ratio decreases. Finally, when the solar radiation is very large, i.e., larger than 600 W/m^2, the temperature difference between the surface of the energy storage layer and the air inside the collector also does not increase so remarkably, and also the air temperature inside the collector increases slightly. Hence, slight decrease of air density and increase of buoyancy happens, which results in a slight increase of enhanced convection heat transfer between the energy storage layer and the air inside the collector. The energy storage ratio, thus, might increase.

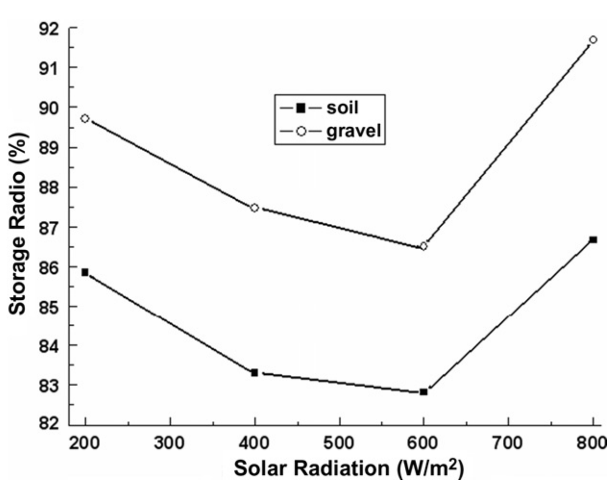

Fig. 2. Variation of heat storage in the energy storage layer on the solar radiation.

It can also be seen from Fig. 2 that energy storage layers made of different materials may have different energy storage ratios with the same solar radiation. The conductivity of the energy storage layer, whose solid material is gravel is about two times that of the energy storage layer whose solid material is soil. It is clear that the energy storage ratio of the gravel energy storage layer at different solar radiations is always a little higher than that of the soil energy storage layer, which indicates that the conductivity might be an important factor to the energy storage ratio of the energy storage layer.

For a given solar chimney power plant system, the position and the number of turbines installed inside the system might be an important problem requiring further discussion. In this paper, the optimum position for the disposition of the turbines can be selected by the concept of relative static pressure advanced by Ming et al. [20]. Fig. 3 shows the relative static pressure distributions of the system for different solar radiations. From this figure, we can see that the minimum values of relative static pressure lie at the bottom of the chimney where there are a dramatic air flow and a large pressure gradient. If a pressure staged turbine with optimized design is placed in this place, we could get a large amount of axial work transferred from the air pressure potential energy and get a large amount of electricity energy accordingly, which means a higher energy conversion efficiency of the system. The relative static pressure, however, inside the collector changes slightly, which shows that the pressure gradient is very small. There is, therefore, not enough pressure head that can be supplied inside the collector for the pressure staged turbine to work efficiently.

Furthermore, from Fig. 3 we can see that the relative static pressure changes significantly with the increase of solar radiation. The minimum relative static pressure inside the system is −74 Pa when the solar radiation is 200 W/m^2, while it is −170 Pa as the solar radiation reaches 800 W/m^2. The main reason lies in the fact that the thermodynamic properties of air inside the system, especially the air density, changes greatly when the solar radiation is very large, which results in a large pressure difference between the system and the environment at the same altitude because of the natural convection phenomenon.

Fig. 4 shows the velocity distributions of the system with different solar radiations. From the figure, we can see that the air velocity inside the collector and chimney is very large, while that inside the energy storage layer is very small. The air velocity of the whole system increases with the increase of solar radiation, and the maximum velocity lies at the bottom of the chimney. When the solar radiation is 200 W/m^2, the maximum velocity inside the system is less than 11 m/s. It can be predicted that the air velocity of the system, if a turbine is placed at the bottom of the chimney, might

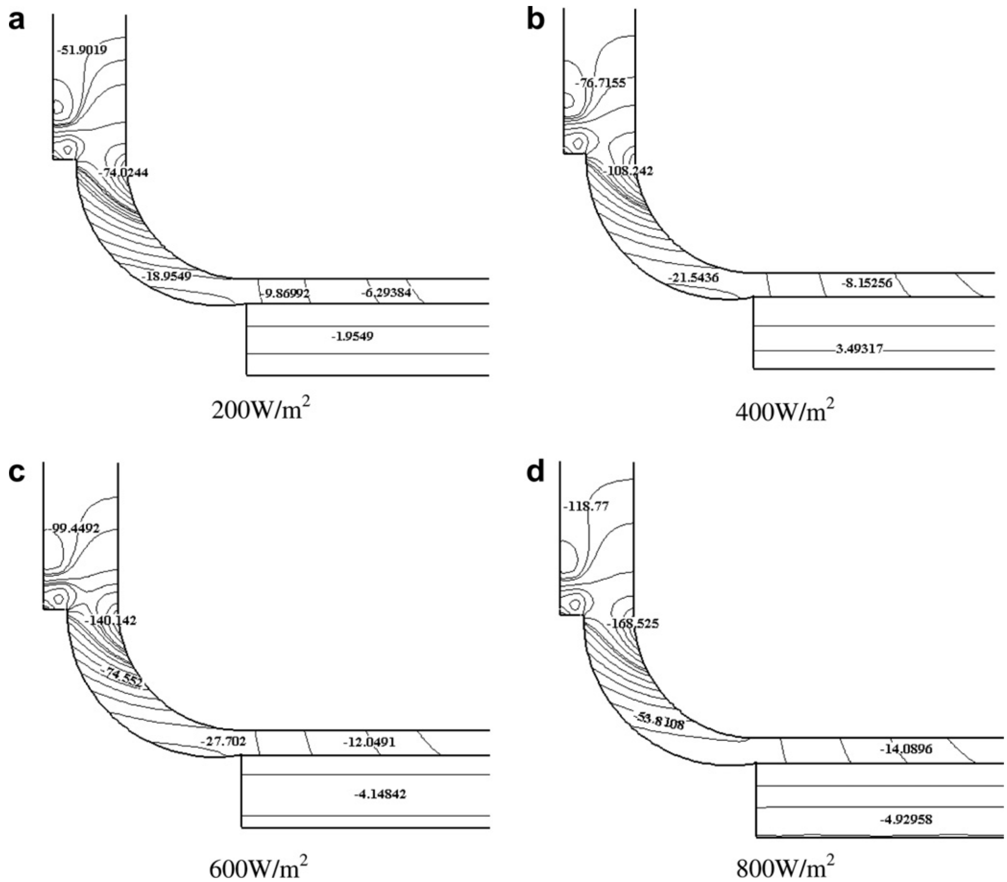

Fig. 3. Relative static pressure distributions of the system in different solar radiation (Pa).

be decreased greatly as the blades of the turbine will increase the resistance. When the solar radiation is 800 W/m^2, the maximum velocity of the system with no load conditions is about 16 m/s. The operation velocity for the turbine is 8 m/s, hence, there is a large scope within which to adjust the turbine rotation velocity in order that the system can be in good operations when the solar radiations is larger than 200 W/m^2.

In order to have a close look at the air flow and heat transfer characteristics inside the energy storage layer, the authors shows the velocity vectors of the system with the same length in Fig. 5. From the figure, we can see that near the collector outlet, air flows into the collector from the energy storage layer, mixing with the air flow inside the collector. Simulation results also show that a part of the air from the environment enters the energy storage layer near the collector inlet before it flows almost parallel to the surface of the energy storage layer in the direction of the collector center. This is beneficial for the energy transfer inside the system, as there are mechanisms of heat radiation inside the matrix and heat convection between the air inside the energy storage layer. By comparison with the results shown in Fig. 7, the mechanisms of heat radiation and heat convection inside the porous media have significant effects on the temperature distribution, hence, a notable difference can be seen between the simulation results conducted by the authors in this paper and those by Pastohr [13]. Fluid flow inside the energy storage layer, which is determined with porous media in this paper and the porosity of the system have great effects on the properties of the energy storage layer, especially the superficial conductivity and capacity, which thereby also influence the energy transfer mechanism inside the energy storage layer and significantly affects the energy storage characteristic.

Fig. 6 shows the temperature distributions of the system for different solar radiations. It can be easily seen from this figure that the temperature inside the chimney is only 304 K when the solar radiation is 200 W/m^2, while it reaches 321 K when the solar radiation is 800 W/m^2, which shows that the solar radiation has a significant effect on the air temperature of the chimney.

In addition, the temperature distributions at the surface and inside the energy storage layer vary greatly with different solar radiations. The reason is that when the solar radiation is very large, the energy storage ratio increases, resulting in a significant increase of the surface temperature of the energy storage layer without notable change of the superficial conductivity of the energy storage layer. Apparently, the temperature gradient of the energy storage layer increases when the solar radiation increases, which consequently results in an increase of heat loss from the bottom of the energy storage layer.

5. Feasibility of the numerical method

Numerical simulation has been conducted for the solar chimney power plant system with collector, chimney and energy storage layer, which has been regarded as porous media in this paper. The feasibility of the numerical method may have a significant effect on the design and commercial application of large scale solar chimney power plant systems. Fig. 7 gives a detailed comparison of the simulation results among the energy equivalent method [5], the Pastohr method [13] and the method in this paper. The results shown in this figure were all determined by the authors of this paper according to the methods listed above.

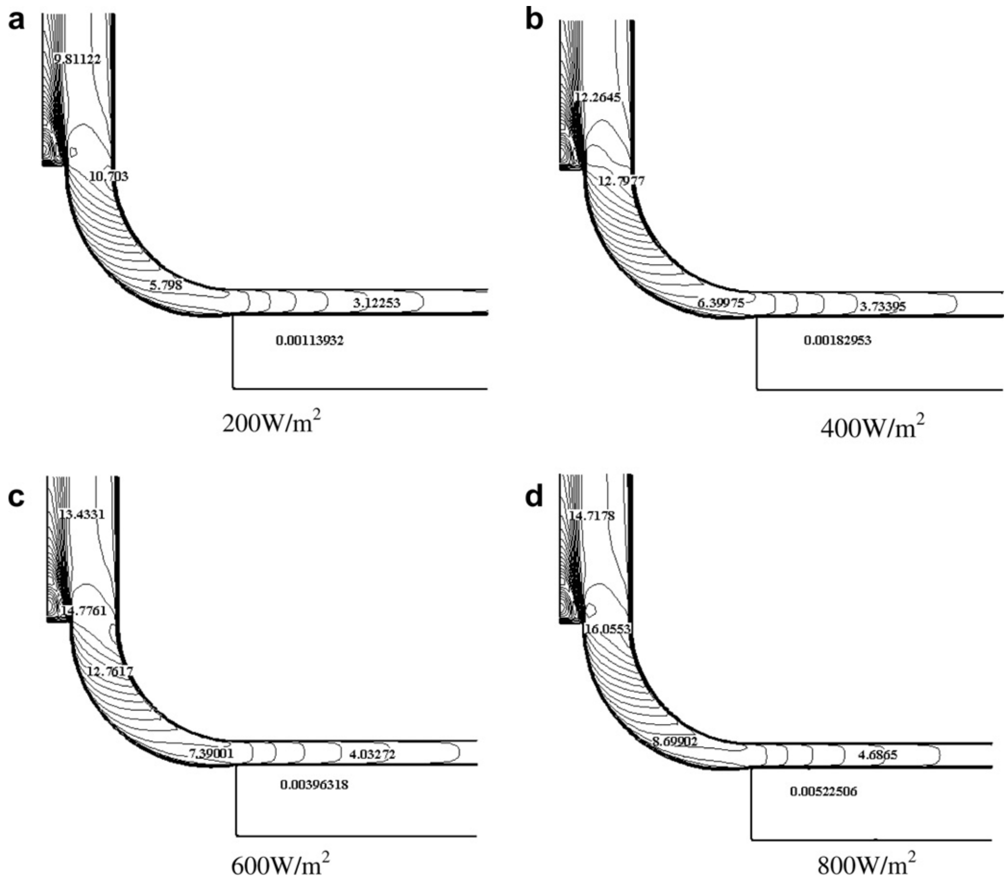

Fig. 4. Velocity distributions of the system in different solar radiation (m/s).

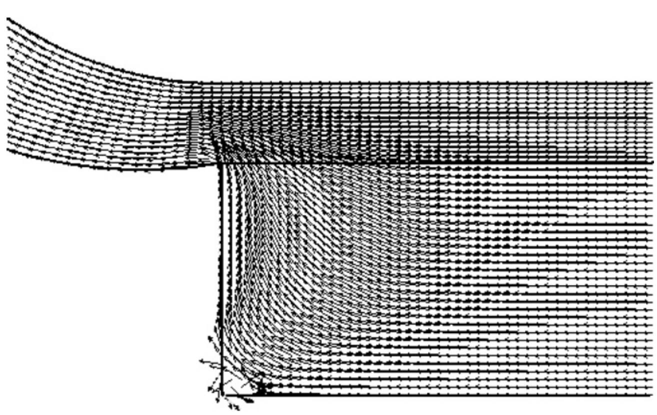

Fig. 5. Velocity vectors of the system with the solar radiation 200 W/m^2.

From this figure, we can see that the temperature profile of the surface of the energy storage layer is lowest by using the Pasumarthi model, and the results shown with the model in this paper are a little higher than those obtained by using the Pasumarthi model but much lower than those obtained by using the Pastohr model. The reasons are as follows. When using the Pasumarthi model to explore the parameters of solar chimney power plant systems, all the solar radiation is considered to be absorbed by the air in the collector without taking into account the effect of the energy storage layer on the whole system. This can give a comparatively lower temperature profile for the storage surface. This method, therefore, will not give a detailed temperature profile of the energy storage layer. On the other hand, the soil is regarded as solid in the Pastohr model, in which heat conduction is the only heat transfer mechanism inside. The model does not take into account the energy transfer mechanisms inside the porous media including radiation and convection, thereby, the temperature profile is the highest. In this paper, a flow and heat transfer model for the porous media is introduced to describe the energy transfer and flow for the energy storage layer of solar chimney power plant systems. The results show that it is reasonable to take into account the energy storage layer as porous media instead of as solid.

No detailed experimental data of the surface temperature profile of the energy storage layer has been found in the Spanish prototype report [2], but the maximum surface temperature in the center of the collector is 348 K when the solar radiation is about 800 W/m^2. There is a temperature difference of 20–25 K between the simulation results in this paper and the experimental data. The main reasons are as follows: (1) different plants and vegetables grew inside the collector of the prototype plant, which enveloped the surface of the energy storage layer and reduced the direct solar insolation; thus, the branches and leaves of the plants received more solar radiation, resulting in the low surface temperature of the energy storage layer; (2) the surface of the energy storage layer in the prototype plant was scraggy, which also decreases the heat transfer rate from the solar insolation, while the surface of the simulation model is regarded as an ideal smooth area; (3) the solar radiation is regarded as a heat source, which happens in a 0.1 mm depth of the energy storage layer during the simulations, so it is apparently different from the real energy transfer process. The difference between the simulation results and the experimental data could be acceptable as the reasons shown above. Thus, the method of this paper is feasible by regarding the energy storage layer as porous media.

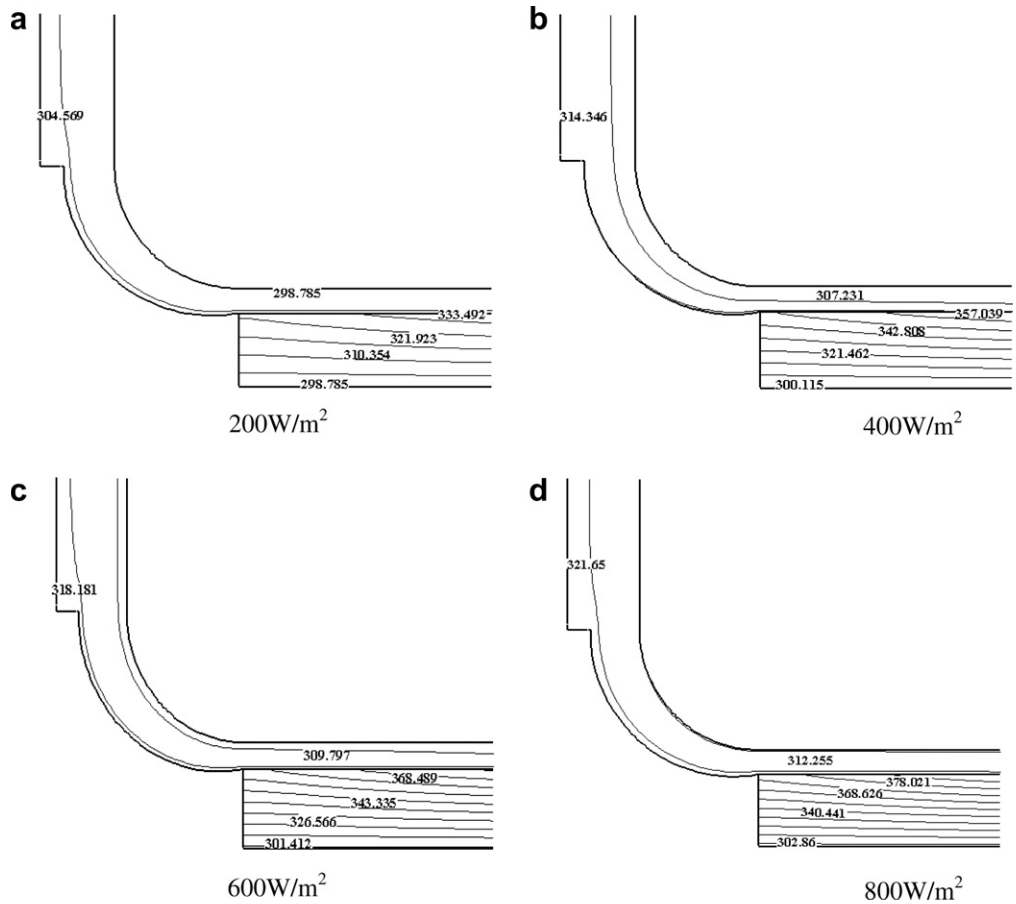

Fig. 6. Temperature distributions of the system in different solar radiation (K).

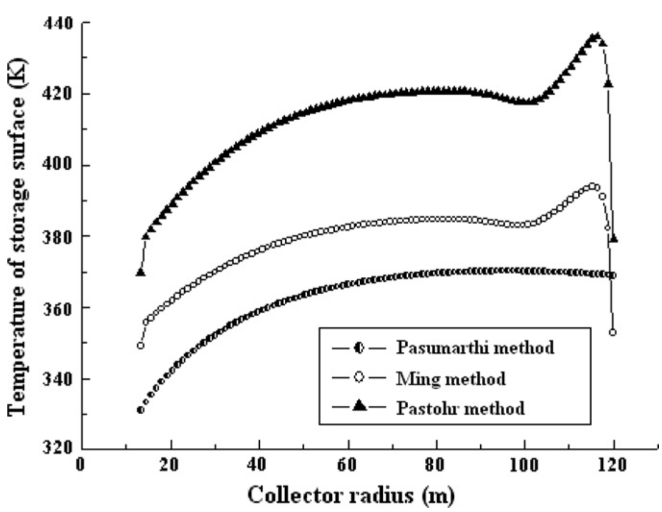

Fig. 7. Temperature profiles of surface of the energy storage layer when the solar radiation is 800 W/m^2.

6. Conclusions

Numerical simulations have been performed to analyze the characteristics of flow and heat transfer of the solar chimney power plant system that includes the energy storage layer. Mathematical models of the collector, chimney and the storage medium have been established, and the effects of different solar radiations on the heat storage characteristic of the energy storage layer have been analyzed. The numerical simulation results show that:

(1) The heat storage ratio of the energy storage layer, which is higher than 80%, decreases firstly and then increases when the solar radiation increases from 200 W/m^2 to 800 W/m^2.
(2) The relative static pressure decreases while the velocity increases significantly inside the system with the increase of solar radiation, but the air velocity inside the energy storage layer is comparatively less.
(3) The temperature of the system and the energy storage layer may increase significantly, and the heat storage ratio of the energy storage layer may increase with the increase of solar radiation.

In addition, the temperature gradient of the energy storage layer may increase, which results in the increase of the energy loss from the bottom of the energy storage layer.

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