Simple analysis on thermal performance of solar chimney power generation systems

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A simple analysis is made on the air flow through a solar chimney power generation system and a thermodynamic cycle of the system including the environment is established. Later, mathematical models for the ideal and actual cycle efficiencies are also established. The research results show that the ideal cycle efficiency and actual efficiency of standard Brayton cycle corresponding to medium scale solar chimney power generation system are 1·33 and 0·3% respectively, while the same parameters for large scale solar chimney power generation systems are 3·33 and 0·9% respectively. The results can give a theoretical guidance to the commercial application of solar chimney power generation systems in China.

Keywords: Solar chimney power generation system, Collector, Chimney, Brayton cycle, Thermal efficiency

List of symbols

c_p	Specific heat at constant pressure, kJ kg^{-1} K^{-1}
g	Gravity constant, m s^{-2}
G	Solar radiation intensity, W m^{-2}
h	Enthalpy, kJ kg^{-1}
H	Chimney height, m
P	Pressure, Pa
q	Heat absorbed by the working fluid, kJ
T	temperature, K
W	Work, kJ
α	Heat transfer coefficient, W m^{-2} K^{-1}
Δ	Difference
η	Thermal efficiency
κ	Specific heat ratio
π	Pressure ratio

Subscript

i	Ideal process
inlet	Collector inlet
outlet	Chimney outlet
p	Used to increase potential energy of the air
r	Relative pressure
shaft	Shaft
turb	Turbine
12	Thermodynamic process 12

Introduction

The solar chimney power generation system (SC system), which has the following advantages compared with the traditional power generation systems: easier to design, more convenient to draw materials, lower cost of power generation, higher operational reliability, fewer running components, more convenient maintenance and overhaul, and lower maintenance expense, no environmental contamination, continuous stable running and longer operational life span, is a late model solar power generation system.[1]

No related experimental results on large scale commercial SC system, however, have ever been reported since the first SC prototype was built in Spain in the 1980s, which is mainly due to the excessive early cost required.[2,3] Establishing a large scale commercial SC of over 100 MW output power requires the financial support from both local government and enterprise. In 1985, Kulunk[4] set up a miniature SC experimental facility. In 1997, Pasumarthi and Sherif[5,6] set up three SC models by modifying the shape and radius of the collector or canopy in Florida University, Gainesville, FL, USA, and carried out experiments on the temperature and velocity distributions of the airflow inside the canopy, whose results agree well with the theoretical analysis. Zhou *et al.*[7,8] presented some experimental and numerical results of a pilot SC equipment.

As for the study of theoretical and numerical analysis of the SC systems, many researchers have conducted related mathematical models and simulation results on different kinds of SC systems. Bernardes *et al.*[9] established a rounded mathematic model for SC system on the basis of energy balance principle. Pastohr *et al.*[10] carried out two-dimensional steady state numerical simulation study on the whole SC system which consists of energy storage layer, collector, turbine and the chimney, and obtained the distributions of velocity, pressure and temperature inside the collector. Ming *et al.*[11] developed a comprehensive model to evaluate the performance of a SC system, in which the effects of various parameters on the relative static pressure, driving force, power output and efficiency have been further investigated. Ming *et al.*[12] established different mathematical models for the collector, the chimney and

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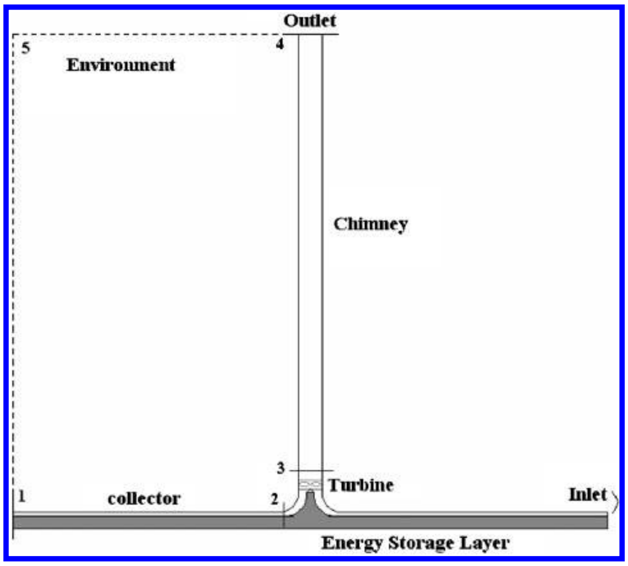


1 Schematic drawing of thermodynamic process of solar chimney (SC) systems

the energy storage layer and analysed the effect of solar radiation on the heat storage characteristic of the energy storage layer. Later, Ming *et al.*[13] carried out numerical simulations on the SC systems coupled with a three-blade turbine using the Spanish prototype as a practical example and presented design and simulation of a MW graded SC system with a five-blade turbine, the results of which show that the coupling of turbine increases the maximum power output of the system and the turbine efficiency is also relatively rather high.

Gannon and von Backstrŏm[14] carried out a fine analysis on the thermal cycle and efficiency of the whole system, and studies on the thermal characteristics, exergy and thermal efficiency have been carried out by Ninic[15] and Ming *et al.*[16] This paper tends to make further analysis on the thermal performance of the SC system, aiming at seeking the offset distance of the efficiencies between the actual SC system and the ideal Brayton cycle and correspondingly, the uppermost reason for it.

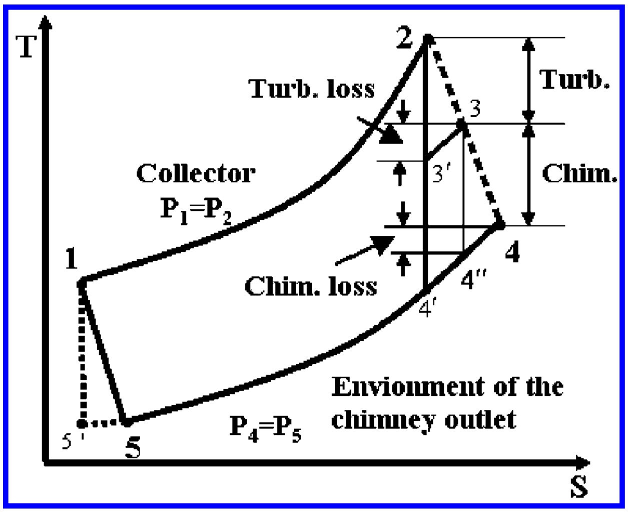
Thermodynamic cycle

The schematic drawing of the thermodynamic process of a SC system is shown in Fig. 1. The significant state points of air flow are as follows:

(i) the state of the collector inlet
(ii) the state of the collector outlet, which is also the state of turbine inlet
(iii) the state of the chimney inlet, which is also the state of the turbine outlet
(iv) the state of the chimney outlet
(v) the state of the environment at the same height of the chimney outlet.

Analysis on the thermodynamic process inside the regions, such as the collector, the turbine, the chimney and the environment, can be referred to in Ref. 14

Starting at the inlet of the collector, the working fluid sequentially flows through the turbine and chimney; finally it releases energy into the environment and again flows back to the collector inlet. Figure 2 shows the ideal standard temperature–entropy diagram for air in SC system, including all systematic loss except for the



2 Temperature–entropy diagram for air in solar chimney (SC) systems

negligible macroscopic kinetic energy of the chimney outlet. The thermodynamic cycle of the working fluid can be, based on the analysis in Ref. 14, simplified into the following four basic thermodynamic processes: process 1–2 refers to the constant pressure heat addition process inside the collector; process 2–4′ refers to the isentropic expansion process under the condition that shaft power is output from the turbine while no shaft power is output from the chimney; process 4′–5 refers to the constant pressure heat rejection process during which air flows from the chimney outlet to the environmental upper air; process 5–1 refers to the isentropic compression process during which the air flows from the environmental upper air to the collector inlet. Among the processes mentioned above, process 4′–5 and 5–1 both take place in the atmospheric environment, and the above four processes constitute a closed thermodynamic cycle.

The thermodynamic cycle of the SC system is a typical Brayton cycle, but there is an apparent difference between its net work and that of a conventional Brayton cycle. The SC ideal cycle 1–2–3′–4′–5–1 is a reversible cycle without irreversible energy loss, and the heat absorbed during process 1–2 in the collector is the total heat input of the cycle, the output power curve is process 2–3′, but the energy consumed curve is process 3′–4′, and process 5–1 which occurs in the environment will not consume any technical work from the system. The shaft work by the turbine for the SC system can be written as follows

$$w_{\mathrm{shaft,i}} = h_2 - h_{3'} \quad (1)$$

The energy used by increasing potential energy of the working fluid when flowing through the chimney is

$$w_{\mathrm{p,i}} = h_{3'} - h_{4'} \quad (2)$$

As analysed above, process 5–1 occurs in the environment which will not cost any technical work from the system. However, the energy consumed by process 3′–4′, which does come from the technical work of the system, will be approximately equal to that by process 5–1. Thereby, most of the technical work in the ideal cycle 1–2–3′–4′–5–1 is used by increasing potential energy of the working fluid when flowing through the chimney, leaving a minimum proportion to be output through shaft work by the turbine installed at the bottom of the chimney. The cycle 1–2–3–4–5–1 represents the actual irreversible Brayton cycle for the SC system, including the turbine loss and chimney loss. The shaft work by the turbine for the SC system can be written as follows

$$w_{\text{shaft}} = h_2 - h_3 = c_P T_2 \left[1 - \left(\frac{p_3}{p_2}\right)^{(k-1)/k}\right] \quad (3)$$

The energy used by increasing potential energy of the working fluid when flowing through the chimney is

$$w_p = h_3 - h_4 \quad (4)$$

Compared with the results shown in equations (3) and (4), the value of w_p will be much higher than that of w_{shaft}.

Thermal efficiency

Under the condition of steady solar radiation during daytime, the thermal efficiency of the cycle η_d can be written as follows

$$\eta = \frac{w_{\text{shaft}}}{q_{12}} \quad (5)$$

where q_{12} can be written as

$$q_{12} = h_2 - h_1 = c_P(T_2 - T_1) \quad (6)$$

Substituting equations (3) and (6) into equation (5), the following can be obtained

$$\eta = \frac{T_2}{T_2 - T_1}\left[1 - \left(\frac{p_3}{p_2}\right)^{(k-1)/k}\right] = \frac{(T_1 + \Delta T)}{\Delta T}\left[1 - \left(1 - \frac{\Delta p_{\text{turb}}}{p_1}\right)^{(\kappa-1)/\kappa}\right] \quad (7)$$

In the above equation, Δp_{turb} stands for the pressure drop across the turbine: $\Delta p_{\text{turb}} = P_2 - P_3 = P_1 - P_3$, while ΔT represents the temperature increase of working fluid inside the collector: $\Delta T = T_2 - T_1$. In equation (7), the value of ΔT can be provided explicitly by numerical simulations, which will be strongly related to the following parameters: solar radiation, turbine pressure drop, etc. Thereby, the actual efficiency of the SC system will be strongly related to the actual operation process which is controlled by various parameters: solar radiation, turbine pressure drop, ambient temperature, geometric dimensions of the SC system, etc.

The ideal thermal efficiency of the solar chimney cycle, if all irreversible losses are neglected, can be expressed as follows

$$\eta_i = \frac{q_{12} - |q_{4'5'}|}{q_{12}} = \frac{\Delta h_{24'} - \Delta h_{3'4'}}{q_{12}} \quad (8)$$

thus

$$\eta_i = \frac{(h_2 - h_{4'}) - (h_{1'} - h_{5'})}{h_2 - h_1} = \frac{(T_2 - T_{4'}) - (T_{1'} - T_{5'})}{T_2 - T_1} \quad (9)$$

According to the characteristics of each process in the Brayton cycle, the following can be obtained

$$\frac{T_1}{T_{5'}} = \frac{T_2}{T_{4'}} = \left(\frac{p_1}{p_{5'}}\right)^{(\kappa-1)/\kappa} = \left(\frac{p_2}{p_{4'}}\right)^{(\kappa-1)/\kappa} = \pi^{(\kappa-1)/\kappa} \quad (10)$$

where π is the pressure ratio $\pi = p_1/p_{5'}$. Substituting equation (10) into equation (9), the following can be obtained

$$\eta_i = 1 - \frac{T_{5'}}{T_1} = 1 - \frac{1}{\pi^{(\kappa-1)/\kappa}} \quad (11)$$

So the ideal efficiency of the SC system is exactly equal to that of the conventional Brayton cycle. When the hot air flows from the bottom through the chimney to the environment, or the same mass of cold air comes from the environment at the same height of chimney outlet to the collector inlet, the energy transfer can be written as

$$c_P \mathrm{d}T = g\mathrm{d}z$$

By integrating this equation, the following can be obtained

$$c_P(T_1 - T_{5'}) = gH \quad (12)$$

From equations (12) and (11), the following can be obtained

$$\eta_i = 1 - \frac{1}{\pi^{(\kappa-1)/\kappa}} = \frac{gH}{c_P T_1} \quad (13)$$

Thereby, the ideal efficiency of the SC system is related to the chimney height and the ambient temperature, if the difference between the process 5–1, process 5′–1 and process 3′–4′ is neglected. The analysis result agrees well with that shown in Ref. 14. This efficiency in equation (13) cannot be obtained because many influence factors are not taken into account.

Results and analysis

To validate the theoretical analysis and to compare with the results between the ideal efficiency in equation (13) and the actual efficiency in equation (7) shown above, three typical types of solar chimney power generation systems were analysed by numerical simulation. The mathematical models to describe the flow and heat transfer characteristics can be found in Refs. 11 and 12, the geometric dimensions for the models can be found in Table 1 and the related parameters with the same values are shown in Table 2.

Boundary conditions are set as follows: for the roof of collector, the authors take convection boundary into account, and coefficient of convection is set as 10 W m^{-2} K^{-1} which can be accepted when the environment air velocity is not very large, i.e. 1–2 m s^{-1}. The temperature of the environment is set as 293 K, inlet of collector is set as the pressure inlet boundary, the chimney wall can be set as an adiabatic boundary, the chimney outlet is set as pressure outlet boundary, and the bottom of the energy storage layer is set as temperature constant boundary, whose

Table 1 Geometric dimensions of three typical solar chimney (SC) systems

Model	Chimney		Collector	
	Height, m	Diameter, m	Height, m	Diameter, m
Spanish prototype	200	10	2–6	122
MW graded	400	50	2–10	1000
100 MW graded	1000	130	3–25	2500

Table 2 Basic parameters of three typical solar chimney (SC) systems

Parameters	Value
Transmissivity of canopy	0·9
Emissivity of canopy	0·85
Absorptivity of storage layer	0·9
Conductivity of storage layer, W $m^{-1}K^{-1}$	1·2
Thickness of storage layer, m	2·0
Ambient temperature, K	293·15
Ambient air velocity, m s^{-1}	2
Solar radiation intensity, W m^{-2}	500, 750, 1000
Turbine pressure drop, Pa	0–1500
Efficiency of turbine, %	72

temperature is 300 K. Solar radiation which projects through the transparent ceiling into the ground can be considered as a heat source for the ground thin layer.[10] The running conditions of a SC system on the whole can be simulated by setting solar radiation as 500, 750 and 1000 W m^{-2} respectively. The acceleration of gravity is 9·8 m s^{-2}, and the air density changing with altitude can be found in Ref. 9. Thereby, the pressure ratio can be calculated accordingly.

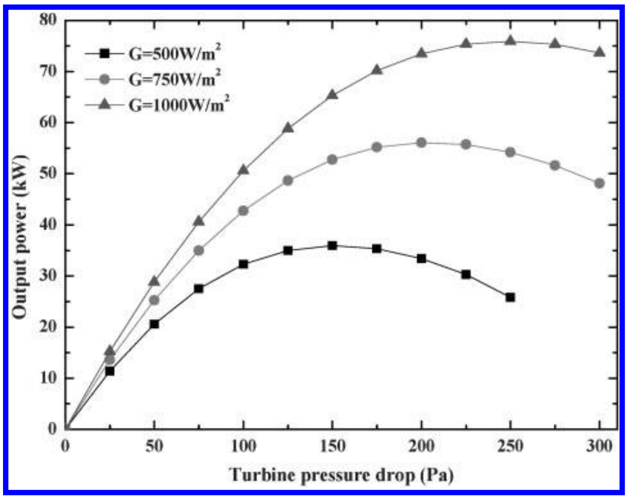
In addition, it is necessary to explain the reason for which the authors consider the collector inlet and chimney outlet both as pressure boundaries and have their pressures set as 0 Pa is that the authors simultaneously take the inner and outer pressure distributions of the system into account; $p_{r,i}=0$ means that for both the inside and outside of the collector inlet, the static pressures at the same height are the same.[10–12]

However, the setting of pressure drop across the turbine in this paper differs from the processing method applied by Pastohr *et al.*[10] The turbine of SC system, as explained earlier above, belongs to a pressure based wind turbine, the fore and aft air velocities are almost the same but the pressure changes significantly, and its power output does not follow the Beetz power limit theory. Thereby, the output power through the turbine can be calculated according to equation (14) by presetting the pressure drop across the turbine

$$W_{shaft}=\eta_{shaft}\Delta pV \quad (14)$$

where, W_{shaft} represents the shaft power output through the turbine, η_{shaft} represents the energy conversion efficiency of the turbine, which can be preset as 80%, less than the optimised data,[17] Δp represents the pressure drop across the turbine, V represents the air volume flow rate of the system flowing through the chimney outlet. The boundary conditions for different places are shown in Table 3.[10]

Standard $k-\varepsilon$ model is applied during the numerical simulation of air flow in the collector and chimney, SIMPLE algorithm is applied for pressure–velocity coupling, and the momentum equation, energy equation and other equations all apply the second order upwind discretisation scheme. The mesh numbers of the 50 kW, MW graded and 100 MW graded SC systems are nearly 500 000, 1 200 000 and 2 500 000 respectively, where grid independent simulation results can be obtained.



3 Output power of solar chimney (SC) prototype in Spain

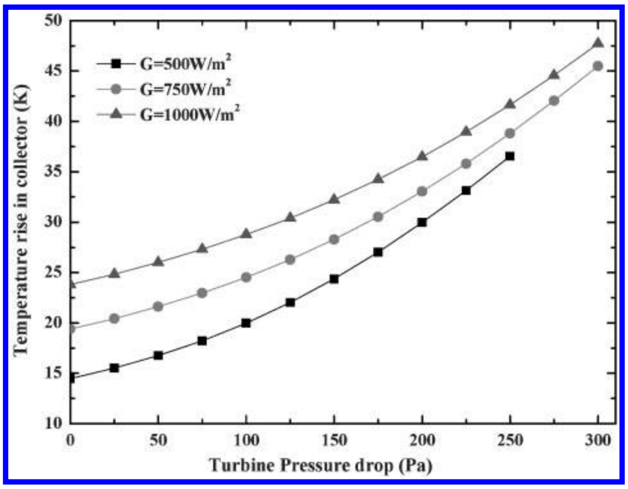
Computation results for Spanish prototype

Figures 3 and 4 show the numerical results of the SC prototype in Spain. As shown in Fig. 3, when the solar radiation intensity is 1000 W m^{-2}, the maximum output power is ~75 kW, ~50% higher than the design value of the Spanish prototype.[2] This is because the efficiency of turbine shown in Table 2 is almost 50% higher than the design value[2] which is based on the Beetz theory. The efficiency of a conventional turbine, which is used in free wind farm, with a good design based on the Betz theory, is a little less than 50%, while the turbines used in the solar chimney systems are pressure staged turbine and their efficiencies will be higher than 85%.[17] Thereby, the efficiency of turbine selected in Table 2 is acceptable and reasonable.

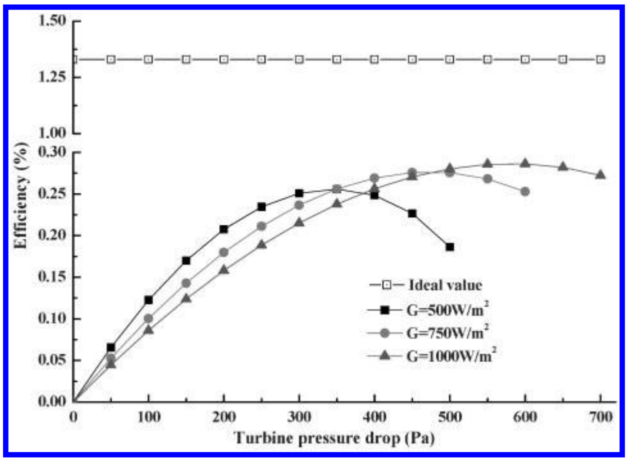
Figure 4 shows the temperature rise of the airflow inside the collector. It can be easily seen that the air temperature rise inside the collector increases significantly with increasing solar radiation intensity and turbine pressure drop. Apparently, the total pressure drop of the SC system is P_1–P_5, which is exactly determined by the chimney height and the air density distribution along the altitude of the ambience. With increasing pressure drop across the turbine, the pressure drop by harnessing the air flow through the chimney decreases, and the air flow rate and velocity will decrease. Thereby, the air temperature rise inside the collector will increase. In addition, the air temperature rise inside the collector also increases with increasing

Table 3 Boundary conditions[10]

Place	Type	Value
0·1 mm top layer of the ground	Heat source	2×10^6–8×10^6 W m^{-3}
Bottom of the ground	Temperature	300 K
Surface of the canopy	Wall	T=293 K, α=10 W m^{-2} K^{-1}
Surface of the chimney	Wall	q_{chim}=0 W m^{-2}
Collector inlet	Pressure inlet	$p_{r,inlet}$=0 Pa, T_0=293 K
Chimney outlet	Pressure outlet	$p_{r,outlet}$=0 Pa



4 Temperature rise inside collector

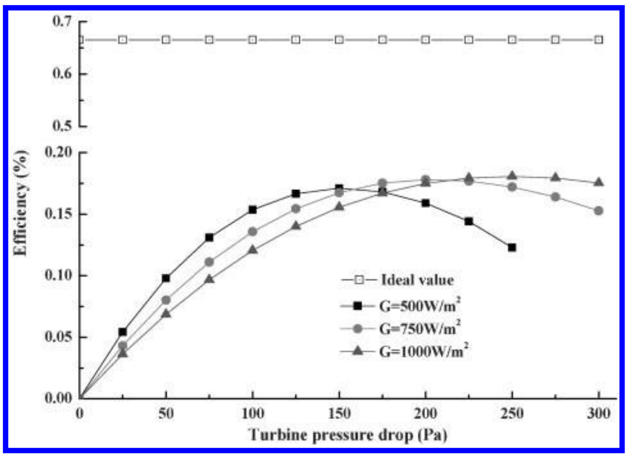


6 Cycle efficiencies of MW graded solar chimney (SC) system

solar radiation intensity. From Figs. 3 and 4, it can be seen that the turbine pressure drops according to the maximum output power, when the solar radiation intensities are 500, 750 and 1000 W m^{-2}, are 150, 200 and 250 Pa respectively, and the temperature increases are 28·28, 33·04 and 41·65 K respectively.

Figure 5 shows the effects of pressure drop across the turbine on the cycle thermal efficiency, in which the efficiency curve of the ideal value refers to the efficiency of Brayton cycle shown in equation (13). From this figure, it can be found that the ideal value of the efficiency of the Spanish solar chimney power plant prototype is 0·665%, while the actual efficiencies corresponding to changing turbine pressure drop of the prototype are all less than 0·2% under different solar radiation intensities. The ideal thermal efficiency of the system is lower than 1%, for which the main reason lies in that most of the heat energy from the solar energy cannot transfer into shaft work during the isentropic expansion process in the turbine (turbines), and it can just be used to overcome gravity when flowing through the chimney.

Moreover, the ideal thermal efficiency of the system is much higher than the actual efficiency, because the increase in turbine pressure drop results in larger temperature rise in the collector shown in Fig. 4, accompanied by a larger heat loss through the collector canopy and a larger energy loss through the chimney



5 Cycle efficiencies of solar chimney (SC) prototype in Spain

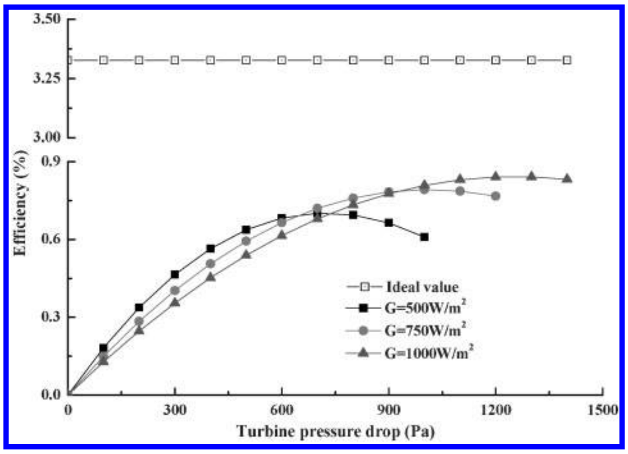
outlet. Therefore, the SC system is unable to reach the ideal thermal efficiency level of standard Brayton cycle shown in Fig. 5. In addition, the actual efficiency of the system is strongly related to the geometric dimensions, the turbine pressure drop, the solar radiation intensity and all the parameters shown in Table 2, while the ideal efficiency is only subject to the chimney height and ambient temperature. Thereby, the actual efficiency of the SC system can never reach the value of ideal efficiency, and the former will be more useful for the design and commercial application of different kinds of SC systems.

Computation results for commercial SC systems

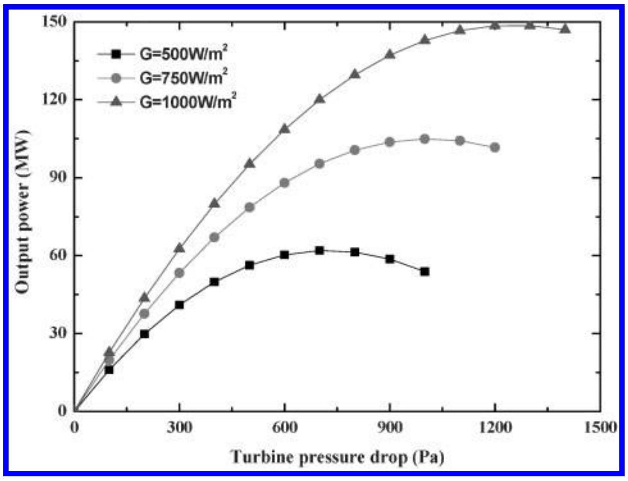
Now the authors carry out thermodynamic analysis on MW graded solar chimney whose geometric dimensions are shown in Table 1, computing for its output power and cycle thermal efficiency. Figure 6 shows the comparison relationship among the efficiencies of actual cycle, ideal cycle of the MW graded SC system. As shown in this figure, the MW graded system ideal cycle efficiency is ∼1·33%, which is independent of all the actual operational parameters such as solar radiation intensity and turbine pressure drop. The maximum value of the system actual efficiencies is near 0·3%, which is lower than one-quarter of the ideal efficiency value. By comparison, the difference between the actual and ideal efficiencies becomes larger with the dimensions of the solar chimney system, because the increase in collector area will decrease the system efficiency due to a larger heat energy loss through the canopy.

The geometric dimensions of large scale SC system are shown in Table 1, and the simulation results are shown in Figs. 7 and 8. As shown in Fig. 7, the ideal efficiency of large scale SC system with chimney height 1000 m is ∼3·33%, while the maximum value of the actual efficiencies is ∼0·9%. Obviously, the actual efficiency is also much lower than the ideal value, which is also due to much larger heat energy loss from the collector canopy.

Although the actual efficiency of the large scale SC system is not higher than 0·9%, the maximum output power is over 100 MW when the solar radiation intensity is 750 W m^{-2} or more. Taking into consideration that the solar chimney power generation system will not consume any fossil energy, the raw material used to build the system is easy to access and the investment for



7 Cycle efficiencies of large scale solar chimney (SC) system



8 Output power of large scale solar chimney (SC) system

the system is relatively lower, the actual efficiency, which is ~1%, is acceptable for the commercial application.

Further study will focus on the following two points:

(i) quantitatively exploring the detailed factors which have effect on the actual efficiency

(ii) finding a new way to increase the total efficiency and output power of the system, for instance, combining the solar chimney power generation system and the photovoltaic power system to build a new synthetic system to highly improve thermal efficiency of the system.

Conclusions

In this paper, analysis is made on the flow of working fluid within various parts of SC system and a thermodynamic cycle, starting from the collector inlet, passing through collector and chimney outlet and finally back to the collector inlet from the environment, is established; besides, numerical models for ideal cycle efficiency and actual cycle efficiency are also established. Checking computations and predictions are carried out for SC systems of various scales, and the research results show that under the same pressure drop as Spanish SC power plant prototype, the ideal efficiency of standard Brayton cycle is ~0·665%, while its actual cycle efficiency is less than 0·2%; the ideal cycle efficiency and maximum actual efficiency of standard Brayton cycle corresponding to medium scale SC system are 1·33 and 0·3% respectively, while the same parameters for large scale SC systems are 3·33 and 0·9%. The actual efficiency of any type of SC system is much lower than the ideal efficiency based on the Brayton cycle, and the former can be used for the design and commercial application of large scale SC systems.

Acknowledgement

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