The thermal efficiency improvement of a steam Rankine cycle by innovative design of a hybrid cooling tower and a solar chimney concept

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Abstract

In the present work, a dry cooling tower and a solar chimney design are recombined in order to increase the thermal efficiency of a steam Rankine cycle. The rejected heat from the condenser into the dry cooling tower supplemented by the solar radiation gained through its transparent cover are the sources of wind energy generation that is captured by a wind turbine which is located at the beginning of the chimney. In this research a case study for a 250 MW steam power plant of Shahid Rajaee in Iran has been performed. A CFD finite volume code is developed to find the generated wind velocity at the turbine entrance for a 250 m dry cooling tower base diameter and a chimney height of 200 m. Calculations have been iterated for different ambient temperatures and solar irradiances, representing temperature gradient within day length. A range of 360 kW to 3 MW power is obtained for the change in the chimney diameter from 10 to 50 m. The results show a maximum of 0.37 percent increase in the thermal efficiency of a 250 MW fossil fuel power plant unit; which proves this design to be a significant improvement in efficiency of thermal power plants, by capturing the heat that is dissipated from dry cooling towers. © 2012 Elsevier Ltd. All rights reserved.

1. Introduction

A solar chimney consists of three essential components: solar collector, chimney, and wind turbine. Air enters the system from its open periphery and is heated up by the solar radiation that is passed through the transparent roof and trapped due to the greenhouse effect. The density difference between the inside and outside causes the heated air to flow through the chimney. The flowing air through its path drives the turbine which is installed at the base of the tower. This concept was first introduced in 1903, and a prototype was constructed in Manzanares, Spain (Gunter, 1931) [1]. Schlaich (1970s) is known as being the first to submit solar chimney as a means to harness solar energy for power generation [2].

The effects of geometry and Insolation level on plant performance have been examined by several researchers. Haaf et al. (1983) realized that an increase of the collector radius engenders increases in output power but reduction of power plant efficiency [2,3]. Pasumarthi and Sherif (1998) reported that the velocity and mass flow rate are increased as a result of increase in chimney height [4,5]. Lodhi (1999) presented a comprehensive analysis of the chimney effect, power production, efficiency, and estimated the cost of the solar chimney power plant set up in developing nations [6]. Investigations of several researchers including Chitsomboon (2000) yielded to the fact that power and efficiency vary linearly with respect to chimney height [7]. These researches also revealed that efficiency of the plant is invariant with respect to the insolation level, the tower diameter and the elevation of the roof [8]. Dai et al. (2003) proved that by increasing the size of the plant, the power output increases nonlinearly, with rapid rate of change when the sizes are smaller [9]. More recently, Tingzhen (2006) announced that the solar radiation and collector radius as well as tower height influence the plant efficiency [10].

Much work has been carried out on the cycle performance, heat transfer and fluid flow in the solar chimney by Gannon et al. (2000) [11]. Ruprecht et al. (2003) give results from fluid dynamic calculations and turbine design for a 200 MW solar chimney [12]. A thermal and technical analysis targeting computer-aided calculation is described by Bernardes et al. (2003) [13]. Many researchers have shown interest in developing a model to predict the solar chimney performance; Chitsomboon (2001) [8], Schlaich et al. (2005) [1], Tingzhen et al. (2006) [10], Zhou et al. (2009) [14,15] and Koonsrisuk and Chitsomboon (2009) [16,17] suggested several theoretical models for this purpose.

Cooling towers are an internal part of many power generation plants, in which they use ambient air to cool warm water which
leaves the condenser. Dry cooling towers have recently been widely used in regions with limited access to vast water resources. In a natural draft cooling tower the air flow is generated by the density difference between the warm air inside the tower and the cool dense outside ambient air. The engineering knowledge of construction and performance of cooling towers have been developed for many years and are familiar for decades.

In this paper, a dry cooling tower has been redesigned by inspiring from a solar chimney to introduce a new concept in which more electrical energy is generated by recapturing the rejected heat from the condenser supplemented by the solar energy gain from the solar collectors. Yet, the function of cooling system of the fossil fuel power plant is not interfered.

One big problem of the traditional solar chimneys is that a large piece of land is required to construct a plant with a reasonable power output because a large surface of collectors are necessary to capture sufficient solar energy from radiation. To mention, the prototype solar chimney constructed in Manzanares with tower height of 194.6 m and collector diameter of 244 m, had a nominal power capacity of only 50 kW. In the new hybrid system the collector surface is reduced and the extra necessary heat for power generation is gained from the radiators of the cooling tower. The objective of this paper is to research the improvements in the efficiency of a fossil fuel power plant by using this innovative concept. For this purpose a CFD finite volume code is developed to find the generated wind velocity on the turbine region for different chimney diameters, solar irradiances and ambient temperatures.

2. Modeling

Modeling of the hybrid cooling tower — solar chimney (HCTSC) system has been developed based on a typical natural draft dry cooling tower of a 250 MW unit in Shahid Rajaee thermal power plant in Qazvin, Iran. The present cross section of this cooling tower with its dimensions is illustrated in Fig. 1. The flow rate and other thermal properties of the circulating water in its vertical cross flow heat exchangers (radiators) are provided in Table 1.

For the hybrid cooling tower — solar chimney model, the base diameter and tower height have been increased in order to harness more solar radiation and increase the wind velocity at the beginning of the tower, respectively.

The schematic layout of the hybrid system and its principal is illustrated in Fig. 2. The ambient cool air enters the system from the open base periphery and passes through the radiators and cools the condenser water within its path. The heated air then passes through the space under the transparent roof and gains more heat from solar radiation, which is trapped in this region because of the greenhouse effect. The transparent roof and the ground below it act as a collector and heat up the flowing air more. The density of the air decreases as its temperature elevates in this region, and causes a stronger natural circulation (compared to alone dry cooling tower) because of the density difference with the surrounding ambient. The buoyant air flows radially towards the center of the system, where some Inlet Guide Vanes (IGVs) divert it through a wind turbine, which is installed at the throat of the chimney. The air drives the turbine in its path and generates electrical power similar to that in solar chimneys. The soaring air loses its temperature as it elevates and is eventually dispersed through the atmosphere.

Geometrical dimensions for current hybrid model are introduced in Table 2. The tower height and the collector radius are considered to be close to the dimensions of Manzanares prototype in Spain to make the power generation enhancements more perceptible through comparison. The roof entrance height from the ground is obtained by keeping this fact in mind that the heat transfer surface of the radiators must remain constant as in the Shahid Rajaee cooling tower, in order not to violate the cooling system operation; i.e. the temperatures of the inlet and outlet circulating water from the condenser will remain the same as in the traditional cooling tower. This height increases smoothly to reach 13 m at the beginning of the chimney base, so that the air is diverted to vertical movement with minimum friction loss [1].

A distance \( d = 20 \text{ m} \) of the roof from the radiators, as illustrated in Fig. 2, is opaque surface in order to prevent the absorption of solar radiation by the ground close to the radiators. Chimney diameter is considered as variable, changing from 10 to 50 m, and the resulting velocities and generated powers are compared for this range. All calculations are iterated for different environmental conditions by altering the ambient temperature and solar radiation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circulating water flow rate ((\text{m}^3/\text{s}))</td>
<td>7</td>
</tr>
<tr>
<td>Volume of water in the system ((\text{m}^3))</td>
<td>4565</td>
</tr>
<tr>
<td>Condenser inlet water temperature (^{\circ}\text{C})</td>
<td>48–50</td>
</tr>
<tr>
<td>Condenser outlet water temperature (^{\circ}\text{C})</td>
<td>60–64</td>
</tr>
</tbody>
</table>
to plot the changes in power output during the day and throughout the year.

3. Computational Model

A finite volume CFD code is used in this study. This approach involves discretizing the spatial domain into finite control volumes using a mesh system. Adaptive unstructured tetrahedral mesh system that is used in the present study consists of almost 5,000,000 cells.

Pressure–velocity coupling is achieved by using the SIMPLE algorithm to derive an additional condition for pressure by reformatting the continuity equation. The SIMPLE algorithm uses a relationship between velocity and pressure corrections to enforce mass conservation and to obtain the pressure field.

Second-order upwind scheme is used for spatial discretization of the mesh system. In this scheme, quantities at cell faces are computed using a multidimensional linear reconstruction approach. In this approach, higher-order accuracy is achieved at cell faces through a Taylor series expansion of the cell-centered solution about the cell centroid.

Finally, convergences of the numerical results were assured by requiring that the RMS residuals of all the conservation equations reached their respective minima.

3.1. Governing equations

The finite volume CFD code which is used in the current study for the hybrid system is based on the following assumptions: (1) the ground which functions as the absorber is considered to be perfectly black (\( \alpha = 1 \)); (2) the transparent roof is considered to have a perfect transmissivity of \( \tau = 0.95 \) for solar radiation; (3) Non-uniform heating of the glazing surface in terms of the sun’s altitude angle is neglected; (4) All opaque surfaces are considered to be completely isolated from their surrounding; i.e. heat loss through

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chimney height (( H_c ))</td>
<td>200 m</td>
</tr>
<tr>
<td>Chimney radius (( R_c ))</td>
<td>5–25 m</td>
</tr>
<tr>
<td>Glazing roof radius (( R_g ))</td>
<td>100 m</td>
</tr>
<tr>
<td>Roof entrance height (( H_r ))</td>
<td>9.5 m</td>
</tr>
<tr>
<td>Distance between collector and radiator (( d ))</td>
<td>20 m</td>
</tr>
<tr>
<td>Turbine elevation above the ground</td>
<td>15 m</td>
</tr>
</tbody>
</table>
the wall of the chimney and the opaque part of the roof are neglected; (5) the Boussinesq approximation is adopted to simulate the variation of the air density through whole system.

As previous studies by Tingzhen et al. [10] show, the Ra number of the solar chimney power plant system is higher than the critical Ra number, 10⁶, resulting in turbulent flow almost all over the system, except for the entrance of the air at the base periphery. Hence, the standard k–ε equation is adopted for the turbulence model in the collector and chimney region in this paper. Other control equations including the continuity, momentum and energy equations which are developed for the whole system can be written as follows.

Continuity equation:
\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0.
\]  

Momentum equations:
\[
\frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u^2)}{\partial x} + \frac{\partial (\rho u v)}{\partial y} + \frac{\partial (\rho u w)}{\partial z} = - \frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]

\[
\frac{\partial (\rho v)}{\partial t} + \frac{\partial (\rho u v)}{\partial x} + \frac{\partial (\rho v^2)}{\partial y} + \frac{\partial (\rho v w)}{\partial z} = - \frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) - G_b
\]

\[
\frac{\partial (\rho w)}{\partial t} + \frac{\partial (\rho u w)}{\partial x} + \frac{\partial (\rho v w)}{\partial y} + \frac{\partial (\rho w w)}{\partial z} = - \frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) - G_b
\]

Energy equation:
\[
\frac{\partial (\rho T)}{\partial t} + \frac{\partial (\rho u T)}{\partial x} + \frac{\partial (\rho v T)}{\partial y} + \frac{\partial (\rho w T)}{\partial z} = \lambda \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) - k \epsilon
\]

\[
\epsilon = \frac{k^3}{\epsilon}.
\]

In the equations above, \( G_b \) represents the generation of turbulence kinetic energy due to the mean velocity gradients, and \( S_e \) is the generation of turbulence kinetic energy due to buoyancy; These variables are obtained from following equations:

\[
G_b = g_b \beta \left( \frac{\mu_t}{\sigma_1} \frac{\alpha}{x_i} \frac{\partial T}{\partial x_i} \right)
\]

\[
S_e = S_i + S\theta
\]

(10)

3.2. Near-wall treatments for turbulent flows

Since walls have a significant effect on turbulent flows, the near-wall treatment is important in modeling of the system in order to represent an accurate prediction of the wall-bounded turbulent flows [18].

In the near-wall zone, the solution variables have large gradients, and the momentum and other scalar transports occur most vigorously. For an accurate prediction of the wall boundary layers in this unstructured mesh, prism layers are generated near the walls with 10–20 or more layers. The thickness of the prism layers is designed such that it ensures that around 15 or more nodes are actually covering the boundary layer.

Because of the several privileges of the semi-empirical formulas of the wall function approach, which are introduced by Tingzhen et al. [18], this approach will be adopted in the current modeling of the hybrid system.

Based on the proposal of Launder and Spalding [19], the law-of-the-wall for mean velocity yields

\[
U^* = \frac{1}{k} \ln \left( \frac{Y^*}{S_k} \right)
\]

(12)

where

\[
U^* = \frac{U_P \rho^{1/4} \kappa_{1/2}}{\tau_{sw}/\rho}
\]

(13)

\[
Y^* = \frac{S^2}{\mu P^{1/4} \kappa_{1/2}}
\]

and \( U_{sw} \), \( K_b \), \( Y_{sw} \) are the mean velocity, turbulence kinetic energy of the fluid at point \( P \) and distance from point \( P \) to the wall, respectively. \( U^* \) is the non-dimensional velocity and \( Y^* \) is the non-dimensional distance from the computational point to the wall. In equation (12), \( x \) is Von Karman’s constant which is approximately equal to 0.41.

When \( Y^* > 11.225 \) then the logarithmic law for mean velocity is employed, otherwise, when the mesh is such that \( Y^* < 11.225 \) at the viscous sublayer, the laminar stress—strain relationship can be written as \( \gamma = U^* \). A similar logarithmic law for mean temperature is obtained by Reynolds’ analogy. Similar to the law-of-the-wall for mean velocity, the law-of-the-wall for temperature employed comprises two different laws for two distinguished regions: linear law for the thermal conduction sublayer, where conduction is important and logarithmic law for the turbulent zone, where effects of turbulence dominate conduction shown as follows:
\[ T^* = \text{Pr} Y^* + \frac{1}{2} \text{Pr} C_u^{1/4} k_{Pr}^{1/2} \beta^2 \left( Y^* - Y_T^* \right) \]  
\[ T^* = \text{Pr} \left( U^* + B \right) + \frac{1}{2} \text{Pr} C_u^{1/4} k_{Pr}^{1/2} \beta^2 \left( U^* + B \right) \]  
\[ \times \left( \text{Pr} U^*_w + \left( \text{Pr} - \text{Pr} U_T^* \right) \right), \quad \left( Y^* > Y_T^* \right) \]  

Where \( B \) is computed by using the formula given by Jayatilleke [20]:

\[ B = 9.24 \left( \left( \frac{\sigma}{\sigma_1} \right)^{3/4} - 1 \right) \left( 1 + 0.28 e^{-0.007 \sigma_1/\sigma} \right) \]  

and \( T_b, T_w \) are temperature at the cell adjacent to wall and temperature at the wall, respectively; \( U_c \) is mean velocity magnitude at \( Y^* = Y_T^* \).

The non-dimensional thermal sublayer thickness, \( Y_T^* \), in equations (15) and (16) is computed as the \( Y^* \) value at which the linear law and the logarithmic law intersect, given the \( Pr \) value of the water being modeled.

4. Boundary conditions and numerical methods

Proper boundary conditions are essential for a successful computational work. The developed code is recomputed for several different geometrical and boundary conditions. As previously described in Table 2, all geometrical parameters of the hybrid system are held constant except for the chimney diameter; which is altered from 10 to 50 m in steps of 10 m; i.e. calculations are iterated for chimney diameters of 10, 20, 30, 40 and 50 m, and the variations in the turbine inlet velocity and power output are investigated.

The properties of the cooling water in the radiator tubes and the circulating air through the system are indicated in Table 3. Using the properties of water and its inlet and outlet temperatures through the radiator, which is introduced in Table 1, the amount of heat transfer to the air flowing over the radiators can be shown by the following equation.

\[ \dot{Q} = \dot{m} C_p (T_{out} - T_{in}) \]  

Knowing the heat transfer surface from the geometrical dimensions of the Shahid Rajaei cooling tower (Fig. 1), the heat flux in the radiator zone can be obtained. This value, which is given in Table 4, is used as a boundary condition in the CFD code.

The main boundary conditions used in the numerical scheme are represented in Table 4. Note that the solar irradiance varies from 300 to 700 W/m² in this simulation. Also the simulation is run in different ambient temperatures in a range of 7–27 °C in steps of 5 °C to show the variation of output power on the wind turbine.

Turbines in a solar chimney do not work with staged velocity as a free-running wind energy converter, but as a cased pressure-staged wind turbogenerator, in which similar to a hydroelectric power station, static pressure is converted to rotational energy using a cased turbine [9].

The maximum obtainable power output \( P_{out} \) for this type of wind turbine is found from the Betz’ law [21], as follows:

\[ P_{out} = \frac{8}{27} \rho A_{ch} V^3 \]  

in which \( \rho \) is the air density, \( V \) is the air velocity on the turbine blades, and \( A_{ch} \) is the cross sectional area of the chimney. In the 1920s, Albert Betz first published the above theoretical derivation of the maximum extractable wind power, which is entirely independent from the wind energy converter type.

5. Results and discussion

5.1. Validity of the method

In order to validate the simulation method used in this paper, numerical simulation results (except for the radiator part) are compared with the theoretical results of Dai et al. [9], which has the performance analysis for a typical solar chimney power plant with 200 m chimney height. The other geometrical dimensions of this prototypical plant are as follows: chimney diameter, 10 m; collector diameter, 500 m; average roof height above the ground, 2.5 m.

When the ambient temperature is 20 °C, the mathematical model shows that the power output alters in a range of 30–280 kW for changes in solar irradiance from 100 to 1000 W/m². Fig. 3 shows the mathematical results of the solar chimney power plant system and the three current simulation results for 300, 600 and 800 W/m² solar irradiances. It can be seen from Fig. 3 that the simulation results are well in scope of the mathematical results of [9].

The computed power generations by the simulation method are 77 kW, 168 kW and 222 kW for solar irradiances of 300, 600 and 800 W/m², respectively; which are approximately 4% less than the

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient inlet temperature ((T_{amb}))</td>
<td>280–300 K</td>
</tr>
<tr>
<td>Inlet total pressure ((P_{in}))</td>
<td>0 Pa</td>
</tr>
<tr>
<td>Outlet static pressure ((P_{out}))</td>
<td>0 Pa</td>
</tr>
<tr>
<td>Solar irradiance ((G))</td>
<td>300–700 W/m²</td>
</tr>
<tr>
<td>Collector heat loss coefficient ((h))</td>
<td>10 W/m² K</td>
</tr>
<tr>
<td>Radiator heat flux ((H))</td>
<td>51.67 kW/m²</td>
</tr>
</tbody>
</table>

### Table 3

The cooling water and circulating air properties.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Water</th>
<th>Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density ((\rho))</td>
<td>998.2 kg/m³</td>
<td>1.225 kg/m³</td>
</tr>
<tr>
<td>Specific heat ((C_p))</td>
<td>4182 J/kg K</td>
<td>1006.43 J/kg K</td>
</tr>
<tr>
<td>Thermal conductivity ((k))</td>
<td>0.6 W/m K</td>
<td>0.0242 W/m K</td>
</tr>
<tr>
<td>Viscosity ((\mu))</td>
<td>~</td>
<td>1.780 x 10⁻⁵ kg/s/m</td>
</tr>
<tr>
<td>Thermal expansion coefficient ((\beta))</td>
<td>~</td>
<td>0.0033 K⁻¹</td>
</tr>
</tbody>
</table>

### Table 4

Boundary conditions for the CFD analysis of the system.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet total pressure</td>
<td>(P_{in})</td>
</tr>
<tr>
<td>Outlet static pressure</td>
<td>(P_{out})</td>
</tr>
<tr>
<td>Solar irradiance ((G))</td>
<td>300–700 W/m²</td>
</tr>
<tr>
<td>Collector heat loss coefficient ((h))</td>
<td>10 W/m² K</td>
</tr>
<tr>
<td>Radiator heat flux ((H))</td>
<td>51.67 kW/m²</td>
</tr>
</tbody>
</table>

**Fig. 3.** Comparison between simulation results and mathematical results.
theoretical results obtained from the case study of [9]. As can be seen in this comparison (Fig. 3), it can be concluded that the simulated results are in close agreement with the results calculated from the theoretical model [9].

5.2. Comparison of the hybrid concept with conventional solar chimneys

For the fixed chimney diameter of 10 m and all other geometrical dimensions introduced previously in Table 2, the computation is run for different solar irradiances in a series of ambient temperatures. The amount of turbine power that could be extracted in different solar irradiances and environmental temperatures is illustrated in Fig. 4. Since the chimney diameter is considered to be constant for all cases, the turbine power output is only a function of air velocity on the turbine section. As introduced earlier in equation (19), this obtained power is proportional to the cubic of the air velocity. Therefore, it can be observed that the amount of power output soars significantly due to higher air velocities caused by increase in solar radiation. The same behavior can be seen for power generation against ambient temperature. This fact is completely reasonable since the air flow becomes more buoyant in higher temperatures, resulting in a faster air circulation. In almost all cases, each 5 °C increase in ambient temperature engenders 20 kW more output power from the turbine.

The results of accessible power, which are illustrated in Fig. 4, are important in this respect that they can be easily compared with the Spanish solar chimney prototype in Manzanares. Since both systems have almost similar geometrical dimensions (chimney height, chimney diameter and collector diameter), a comparison between the Manzanares solar chimney and the hybrid system can obviously reveal the differences in output power of the two concepts. This comparison is shown in Fig. 5.

The measured data from Manzanares prototype [1] indicates that the maximum output power that can be obtained from the solar chimney turbine in environmental conditions of 600 W/m² solar radiation and about 20 °C ambient temperature is approximately 25 kW; while the computational results that are shown in Fig. 5 indicate that the amount of obtainable power for the hybrid system, which is designed for the 250 MW Shahid Rajaee power plant, is almost 350 kW, for the same environmental conditions. This is about fourteen times greater than the output power of the former system. This is due to the initial increase in the air flow temperature that is caused by the dry cooling tower radiators as a significant heat source.

5.3. Chimney diameter effects on the HCTSC power generation

Fig. 6 shows the effect of different chimney diameters on the average flow velocity at the throat of the chimney and the resulting output power. All of the cases are obtained for constant environmental condition and solar radiation as follows: ambient temperature, 22 °C; solar irradiance, 600 W/m².

It is evident from the results that the velocity decreases significantly with chimney diameter increase due to continuity of the air flow. Fig. 6 demonstrates that the air velocity decreases from 22.5 m/s to approximately 15.5 m/s according to the increase in chimney diameter from 10 to 50 m. However, this decrease is more rapid for smaller chimney diameters.

Although the velocity is remarkably reduced with diameter increase, Fig. 6 shows that the power production of the HCTSC system would be greater, the larger the chimney diameter becomes. According to equation (19), as the intake area of the air flow to the turbine grows, greater power output would be the result. Also, as demonstrated in Fig. 6, the chimney power output increases non-linearly with the increase of chimney diameter. About 1.5 MW electric power can be generated in the HCTSC when the diameter of the chimney is 30 m; this amount will increase to over 3 MW when the chimney diameter reaches 50 m. It is important to consider that the collector diameter and the chimney height are both still 200 m, but MW-graded solar chimneys have been reached. Prior to this time tower heights of more than 500 m, and collector diameters of more than a kilometer were required to reach such power outputs.

Although it seems that the wind power will increase exponentially as the chimney diameter increases, but it should be considered that this generated power is bounded because of the characteristic power curve of the wind turbine converters. This curve reveals the dependency of the average electrical power from the respective average wind velocity and thus shows the operational characteristics of the converter [21].
If the air flow speed exceeds 4 m/s, which is required for turbine start-up, as introduced by Kaltschmitt and his colleagues [21], the converter will start and generate electrical energy. The theoretical useful wind power increase is proportional to the wind speed in its third power. Yet, the useful electrical energy at the generator outlet is not exactly proportional to the theoretic useful energy as losses which are not linear to speed (e.g. aerodynamic friction losses) occur within this range of the characteristic power curve. Hence the electric output power of the wind turbine is only a portion of the power that the system can convert at high wind speeds, at which it works with its full capacity. The generator capacity of the convertor increases proportional to nominal wind speed, until it reaches its hundred percent capacity in average wind speeds higher than 12–14 m/s [21].

As it is shown in Fig. 6, the average wind speed on the turbine blades reduces to approximately 15 m/s as the chimney diameter increases to 60 m. Since the results in Fig. 6 are for the ambient temperature of 22 °C, it should be considered that the flow speed may undergo a 1–2 m/s reduction as the temperature descends 10 °C because of the ambient effect which was introduced earlier in Fig. 4. Regarding the operational characteristics of convertors, the wind turbine may enter the phase in which the convertor cannot operate in its full capacity, as the nominal wind speed descends under 14 m/s. Hence the theoretical power output cannot be calculated by equation (19), with which the earlier results were obtained. Therefore, larger chimney diameters are not recommended for the current design and environment condition.

The effect of chimney diameter on the thermal power plant efficiency will now be considered in Fig. 7. In this case, the Shahid Rajaee typical unit of 250 MW thermal power plant with nominal efficiency of 30% (for sole fossil fuel power plant without hybrid solar chimney system) is been considered. Since the solar energy gain is known to be an everlasting free input source, the hybrid efficiency is only based on the fossil fuel energy input which is gained at great expenses. According to the amount of output powers that have been generated by the HCTSC, shown in Fig. 6 for each chimney diameter, this value is added to the nominal power output of the fossil fuel thermal power plant (250 MW), and the new efficiency for the hybrid system is then calculated from dividing this sum by the gross fossil fuel input power. Considering the conventional fossil fuel power plant efficiency to be 30%, the gross fossil fuel input power is obtained and is 833.33 MW.

Hybrid efficiency
\[
\text{Hybrid efficiency} = \frac{\text{fossil fuel power output} + \text{hybrid power output}}{\text{gross fossil fuel input power}}
\]  

The results indicate that when the chimney diameter is 10 m, the efficiency improvement is less than 0.1%, but as the diameter increases to 50 m, the efficiency improvement is almost 0.4% and the overall thermal power plant efficiency reaches remarkable value of 30.4%. This increase in Rankine cycle efficiency is significant considering that lots of fossil fuel usage is saved, and consequently the amount of generated CO2 is reduced in long term utilization.

Also shown in Fig. 7, it is conspicuous from the results that when the chimney diameter is 50 m, the excess generated power output from the HCTSC system is approximately 1.2% of the total nominal power output of the thermal power plant unit, which is 250 MW.

5.4. Cooling system operation restrictions

As discussed earlier, power generation is not the only purpose of the hybrid cooling tower – solar chimney system. The primary goal of this concept is to cool the warm water leaving the thermal power plant condensers and passing through the radiators. Hence, the cooling section of the hybrid system should operate as efficient as the sole dry cooling tower unit. That is, the parameters that may violate the cooling process in the HCTSC should be controlled in order to prevent losses. For instance, the amount of heat that is rejected from the circulating water in the radiators should not decrease, and so the thermal efficiency of the Rankine cycle should remain constant.

One of the most important parameters that may have been altered because of the variations in geometrical dimensions of the hybrid tower (especially the pressure drop caused by the turbine) compared to the dry cooling tower is the flow velocity on the radiators. It should be considered that if the entrance wind speed on the radiators reduces significantly, it causes a malfunction in cooling system. That is, the amount of heat, which is rejected from the radiators by air flow, will decrease due to reduction in convective heat transfer. Thus, the inlet velocity to the system should be retained above a limit, by making constraints for geometrical dimensions.
Fig. 8 shows the air flow average velocity variation in the Shahid Rajaee cooling tower (dimensions are introduced in Fig. 1) for the 22 °C outside temperature. As the results indicate, the flow speed on the radiator section, which is located at a distance approximately 60 m from the tower center, is a little higher than 4 m/s. Therefore, proper dimensions for the hybrid system that maintain an inlet velocity in this range or even higher than this limit should be chosen. For this purpose, velocity of the circulating air in radiator section of the hybrid system for different chimney diameters is computed.

Fig. 9 compares the air velocity in various distances from chimney center of the HCTSC system for different chimney diameters. The wind speed at the distance of 120 m from chimney center is important to be controlled since the radiators are located at this distance. The chosen dimensions should be such that the generated inlet wind has a speed higher than 4 m/s to guarantee sufficient heat transfer on radiator section, as described previously.

Regarding cooling system effectiveness, the chimney diameters lower than 40 m are not proper in the current design since the air velocity is lower than the criterion speed which is necessary for efficient heat transfer. A chimney with diameter of 50 m is suitable for this purpose since the velocity of the wind is slightly higher than 5 m/s on radiator region for this dimension. It also guarantees proper function in lower ambient temperatures, when reduction in inlet velocity is inevitable.

Combination of a dry cooling tower with a solar chimney power plant will not only enhance the Rankine cycle efficiency of the thermal power plants but will also make the conventional solar updraft systems an efficient and economical medium for power generation by boosting their output power. This hybrid design is mainly proper for dry regions that use dry cooling towers because of the lack of reliable water resources. In such regions, the environmental conditions, ambient temperature and solar radiation, are also high enough for a suitable performance of solar chimney power plant.

6. Conclusions

An innovative concept for recombining a thermal steam power plant dry cooling tower with a solar chimney is introduced in this paper. A model has been designed using the typical dimensions and properties of Shahid Rajaee 250 MW steam power plant and the Manzanares solar chimney. A numerical simulation for the hybrid system including solar collectors, cooling tower radiators and wind turbine is then developed. The effects of environmental temperatures and solar irradiations on the generated turbine power have been illustrated. At the end, the effects of chimney diameter on the hybrid system (HCTSC) power output and the total fossil fuel power plant efficiency have been researched.

The results indicate an over ten times increase in output power of the hybrid system compared to experimental results for the conventional solar chimney power plant prototype with similar geometrical dimensions in Manzanares, for the same environmental conditions.

In addition, with increase of chimney diameter, the power generation can reach to MW-graded power output without the necessity of building huge individual solar chimney power plants. The results show a maximum of 3 MW power output from the HCTSC system that results in 0.37% increase in the thermal efficiency of the Shahid Rajaee 250 MW fossil fuel power plant, when the chimney diameter is 50 m.

Two different design criteria (convertors characteristic curve and cooling system function) are considered to help selecting the proper chimney dimensions. An upper limit for the chimney diameter is indicated to maintain the wind speed on the turbine blades to remain in full convertor capacity phase. Also, the wind velocities in different distances from chimney center are computed for different chimney diameters and the results are compared to the sole dry cooling tower unit to prevent cooling system malfunction. A lower boundary for chimney diameter is then introduced.

From the results, this design proves to be economical in saving lots of construction cost as well as time requirements. It also recaptures the heat of radiators that are thrown out to the atmosphere without efficient utilization, and prevents generation of excess greenhouse gases. Thus, the hybrid combination of the two systems has several benefits over both cooling tower and solar chimney power plants that would perform individually.
References


