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Advanced Air-Cooled Heat Exchangers for Geothermal Power Plants

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ABSTRACT

The application of different heat transfer surface extension techniques to improve the performance of an air-cooled heat exchanger is examined. Particularly, our case study focuses on a Solar Enhanced Natural Draft Dry Cooling Tower (SENDDCT) designed by Queensland Geothermal Energy Centre of Excellence (QGECE), as the air-cooled condenser of a geothermal power plant. The conventional method of extending the heat transfer area by means of fins is compared with a modern technique being the application of a thin metal foam layer to the outer surface of the tube. Both fins and foams lead to heat transfer augmentation, from the cycle fluid flowing in the tube bundle, albeit at the expense of a higher pressure drop when compared to the bare tube bundle as our reference case. Aiming at maximizing the heat transfer enhancement and minimizing the total pressure drop, the two heat transfer surface extension techniques are compared against each other and an optimal solution is obtained. Different tube bundle layouts and tube spacing are examined. Sensitivity analysis was conducted to investigate the effect of sunroof diameter on the overall performance of the system. Aiming at minimizing the flow and thermal resistances for a SENDDCT, an optimum design is presented for an existing tower to be equipped with solar panels to afterheat the air leaving the heat exchanger bundles arranged vertically around the tower skirt. A number of correlations are also proposed to predict the total pressure drop and heat transfer of the extended surfaces considered here.

1. INTRODUCTION

Low-emission power from renewable sources is the way to future with diminishing non-renewable fuels. Enhanced (or Engineered) Geothermal Systems, EGS, is one of the options being considered in Australia over the past decade. However, most of the geothermal resources are located in Australian hinterland with no water to feed wet cooling towers which are very popular options for heat removal from power plants. This leaves dry cooling as the only economic choice in such places. Such systems use the ambient air to cool the cycle fluid which condenses inside the tube bundles. Almost always, these tubes have external fins to reduce the (dominant) air-side resistance by increasing the air-side heat transfer area. Such surface extension techniques improve the heat transfer performance albeit at the expense of extra pressure drops. Commensurate with that, the designer has to consider the tradeoff between these two opposing effects. The problem, however, becomes more complicated in case of thermodynamically less efficient geothermal power plants where the waste heat generated, per generated MW electricity, is almost twice as that of coal-fired power plants. This heat can be dumped through the use of fans or by relying on cooling towers. In either case, a highly efficient heat exchanger operating at low pressure drop is desirable. Fans can consume close to 1% of the net generated power according to Kroger (2004); of course, on top of maintenance costs. This makes the mechanical draft less popular for geothermal power plants which are already having low thermal conversion efficiency.

As such, our research focused on cooling towers. With Natural Draft Dry Cooling Towers, NDDCTs, the pressure difference due to buoyancy (the driving force) is linearly proportional to the height of the cooling tower. This buoyancy-induced pressure difference needs to be large enough to overcome the flow resistance in the tower which is dominated by the bundle resistance. As such, taller towers which are more expensive to build can generate higher flow rates leading to better heat transfer. Alternatively, one could afterheat the air to enhance the buoyancy force leading to higher driving forces even at shorter towers. SENDDCT, as an extension to solar chimneys Akbarzadeh et al. (2009), Ming et al. (2013) and Shen et al. (2014), has then been investigated in details by Zou et al. (2012-2014); see Fig. 1. In his design, Zou (2014) blocked parts of the tower inlet to allow for compact multi-row heat exchangers in the remainder of the inlet area. As a result of buoyancy, the air is drawn to the tower through the heat exchangers and then, after removing the heat from the bundles, the air is then heated by the sun. This further reduces the air density leading to higher air speed at the tower inlet and thereby improving the heat transfer rate.

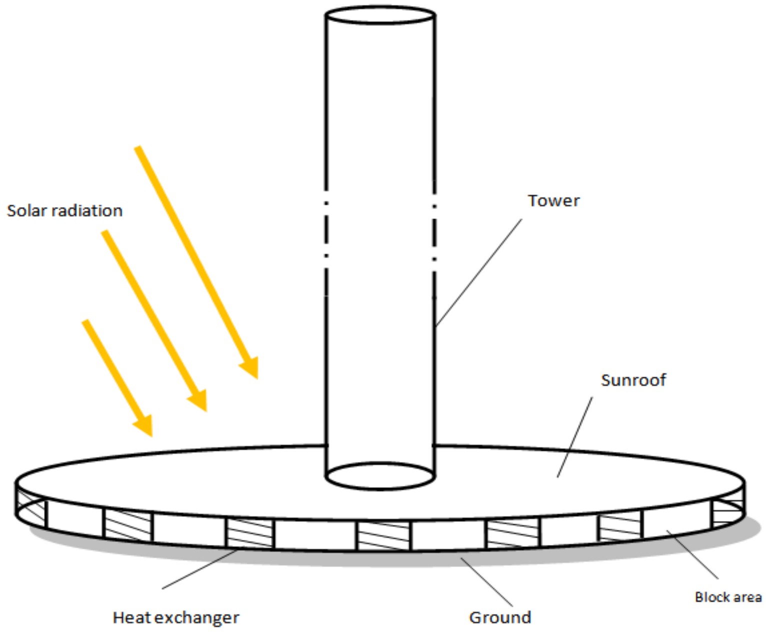

Figure 1: The SENDDCT concept investigated by Zou (2014).

At the same time, in an attempt to improve the heat transfer from the cycle fluid, metal foam-wrapped tubes, as alternatives to finned-tubes, have been investigated by Odabaee et al. (2011-2013). Metal foams offer better flow mixing and lead to heat transfer augmentation as a result of boundary layer interruptions at the pore level. They embrace small continuously-connected ligaments in an open-celled structure. These cells are usually polyhedrons, of 12–14 faces, each of which with a pentagonal or hexagonal shape (by five or six filaments). Figure 2 shows a sample of the aluminium foam wrapped around 3 cm diameter tubes used for experiments in our laboratory. Chumpia and Hooman (2014) proposed experimentally-obtained correlations for flow and thermal resistance of such foamed-wrapped tubes in cross-flow where hot water was flowing in the tube and cold air was pushed to flow across the tube.

In what follows, it will be shown that proper combination of single-row widely-spaced foam-wrapped tubes, instead of multi-row finned-tubes, in a SENDDCT leads to shorter towers to meet the required heat rejection rate for an air-cooled condenser. In order to do so, we used the experimental data from Chumpia and Hooman (2014) to address the thermo-hydraulics of the foamed tube bundles used as air-cooled condensers. We also used the 3D numerical and 1D theoretical model of Zou (2014) as a case study for comparison purpose.

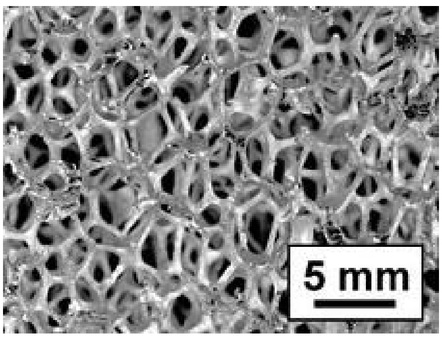

Figure 2: The close-up of an aluminum foam sample attached to a model condenser tube where the cells' structures leading to induced tortuous flow path are shown.

2. ANALYSIS

As mentioned, SENDDCT uses solar energy to afterheat the air leaving the heat exchangers with the goal of increasing the buoyancy forces thereby leading to higher draft speed or shorter towers. The heat transferred from the hot fluid in the heat exchanger, Q_a, increases the air temperature by

$$\Delta T_{hx} = \frac{Q_a}{\dot{m}c_p} \qquad (1)$$

Going through the heated area, under the panels, the solar heat is then picked up by the air stream. This excess heat is then given by

$$q_s A_s = \dot{m} c_p \Delta T_s \quad (2)$$

where A_s is the sunroof (panel) area substituting for which leads to

$$q_s \frac{\pi}{4}\left(D^2 - D_b^2\right) = \dot{m} c_p \Delta T_s \quad (3)$$

Here, the air specific heat at constant pressure is given by c_p and the air flow rate is given by $\dot{m} = \rho\pi D l U$ with l being the tower inlet opening height, D the sunroof diameter, D_b the tower base diameter and ρ the air density. Aiming at finding the temperature rise due to solar heating, one rearranges the above equation and substitutes for the air flow rate to get

$$\Delta T_s = \frac{q_s}{4\rho c_p} \frac{\left(D^2 - D_b^2\right)}{UDl} \quad (4)$$

The total temperature rise is then given by adding Eqn. (1) to (4), i.e.

$$\Delta T_a = \Delta T_{hx}\left(1 + \frac{\Delta T_s}{\Delta T_{hx}}\right) \quad (5)$$

Then the required pressure difference to be provided by the tower is given by

$$\Delta p \cong \rho g \beta \Delta T_a H \quad (6)$$

wherein β is the thermal expansion coefficient, H is the tower height and g is the gravitational acceleration. Now the temperature difference is higher than the case with no solar heating. As such, the above equation reads

$$\Delta p \cong \rho g \beta \Delta T_{hx}\left(1 + \frac{\Delta T_s}{\Delta T_{hx}}\right) H \quad (7)$$

Rearranging the above, the following equation is obtained

$$\frac{\pi D_b^2 U \Delta p}{4 Q_a} \cong \frac{g \beta H D_b^2}{4 l c_p}\left(\frac{4}{D} + \frac{\pi q_s}{Q_a}\frac{\left(D^2 - D_b^2\right)}{D}\right) \quad (8)$$

The left-side of the above equation indicates the pumping power divided by the heat transferred from the bundle. As such, it makes perfect engineering sense to try to minimize this function; which, in turn, means the right hand side has to be minimized. For a tower with set heat transfer duty, height and base diameter, one can change the sunroof diameter to optimize the system. As such, one can differentiate the above equation with respect to D and set it to zero to find the optimal D value for which the excess pressure drop is minimized. This will lead to

$$D^2 = \frac{4 Q_a}{\pi q_s} - D_b^2 \quad (9)$$

Making use of the above optimal sunroof diameter, we have

$$\frac{\pi U D_b^2 \Delta p}{4 Q_a} = \frac{g \beta H D_b^2 \sqrt{\pi q_s\left(4 Q_a - \pi D_b^2 q_s\right)}}{8 c_p l Q_a} \quad (10)$$

If rearranged, it will correspond to the following heat transfer rate

$$Q_a = \frac{\pi q_s D_b^2}{4}\left(1 + \left(\frac{2 \Delta p U c_p l}{g \beta H D_b q_s}\right)^2\right) \quad (11)$$

The above formula optimizes a *designed* tower for addition of sunroof area. However, it does not tell us how a tower should be optimized with *built-in* solar heating. One answer to this question will be through minimizing losses in the tower. The dominant flow resistance is posed by the heat exchangers which are usually designed in the form of multi-row bundles simply because area is limited. Moreover, the tubes in the first rows of a multi-row finned-tube bundle act as turbulence generators for subsequent rows. This leads to higher hat transfer rates for deeper tubes. Furthermore, spacing the tubes densely leads to high speed local jets for the subsequent rows which, in turn, bring in more efficient heat exchange process compared to sparsely-bundled finned-tubes. Obviously, denser arrangement of the tubes leads to higher pressure drops which have to be compensated for using taller towers.

As the heat exchanger area is now extended, thanks to the addition of collector plates, one can use single row heat exchanges leading to much lower pressure drops and, of course with lower heat transfer rates compared a compact multi-row heat exchanger. For a vertical arrangement of the heat exchangers, this total heat is transferred by a number of tubes of external (fin or foam) diameter d, spaced sd apart, N, which is given by

$$N = \frac{\pi D}{sd} \quad (12)$$

With an average heat transfer per tube given by q, one has $Q_a = Nq$. If the tubes are widely spaced, then q is the same as that given by a single tube in cross flow. With denser tube bundles, however, this is not the case. One can still define a per-tube heat transfer but that will not be the same as the heat transfer from a single tube in cross flow. Combining the above equation with Eq. (1), and substituting for the air mass flow rate therein one has

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$$\Delta T_{hx} = \frac{1}{sdl}\frac{q}{\rho c_p}\frac{1}{U} \tag{13}$$

Note that this temperature rise is not the same as that of a tower with no solar heating as the fluid velocity is not the same as a result of different flow resistance offered by a single-row and a multi-row heat exchanger.

Here, the heat transfer per tube is predicted using the available experimental data and correlation given by Chumpia and Hooman (2014), i.e.

$$\Delta T_{hx} = \frac{1}{sd}\frac{1}{\rho c_p}\frac{\Delta T_{sa}}{RU} \tag{14}$$

where R is the overall thermal resistance given as a function of approach velocity U by

$$R = 0.145 D_r^{-1.617} U^{-0.595} \tag{15}$$

Note that D_r is the tube outer (foam or fin) diameter divided by the bare tube diameter. For a bare tube bundle, D_r=1. Furthermore, the temperature difference ΔT_{sa} is the difference between the ambient air and average tube surface which is very close to the average cycle fluid temperature. As such, this ΔT_{sa} can be thought of as *approach*.

Temperature rise due to solar heating is still expressed by $\Delta T_s = \frac{q_s}{4\rho c_p}\frac{(D^2-D_b^2)}{UDl}$ and as result the total temperature rise is given by

$$\Delta T_a = \frac{6.8971 D_r^{1.617}\Delta T_{sa}}{\rho c_p s d U^{0.404}} + \frac{q_s}{4\rho c_p U}\frac{D}{l}\left(1 - \frac{D_b^2}{D^2}\right) \tag{16}$$

This leads to the total driving force of

$$\Delta p = \frac{g\beta H}{c_p}\left(\frac{6.8971 D_r^{1.617}\Delta T_{sa}}{s d U^{0.404}} + \frac{q_s}{4U}\frac{D}{l}\left(1 - \frac{D_b^2}{D^2}\right)\right) \tag{17}$$

Furthermore, the bundle pressure drop, which is the dominant resistance, is given by

$$\Delta p = 0.807 D_r^{0.859} U^{1.828} \tag{18}$$

Through the use of complex 3D CFD simulations, Zou et al. (2012) showed that the heat exchanger losses account for about 90% of the total losses. This is in line with the scale analysis and numerical results reported in Hooman (2010) and Tanimizu and Hooman (2013) for a NDDCT with no solar enhancement. Going with a more conservative assumption of 20% non-heat exchanger losses, one can balance the projected total loss with the driving force, leading to the SENDDCT height given by

$$H = \frac{c_p D_r^{0.859} U^{1.828}}{g\beta\left(6.8971 D_r^{1.617}\frac{\Delta T_{sa}}{s d U^{0.404}} + \frac{q_s D}{4U\, l}\left(1 - \frac{D_b^2}{D^2}\right)\right)} \tag{19}$$

Note that depending on U, one can get different tower heights from the above equation. These H values satisfy the draft equation for a certain velocity, approach and geometrical values; see Eq. (7) above. Furthermore, the total heat dump associated with each U and H combination is fixed and is given by Eq. (1).

3. **RESULTS**

In order to compare the single-row design suggested here with the multi-row data pertinent to SNDDCTs, the assumptions made in Zou et al. (2012) are recovered. There it was shown that with a tower height of 140 m, about 135MW of heat can be dumped. The tower base diameter was 100 m with the bundle height being equal to 15 m. A total of 243 vertical 3-row bundles with 36 tubes per row were used. This translates into 26244 tubes with an inner diameter of just under 4 cm with a D_r value of 1.75. Different sunroof diameters, ranging from D = 195 m to D = 475 m were examined. The whole 3D system was simulated using ANSYS and the numerical results and 1D theoretical prediction were cross-validated. Figure 3 shows a comparison between the new design offered here and that of Zou et al. (2012- 2014) as summarized in Zou (2014).

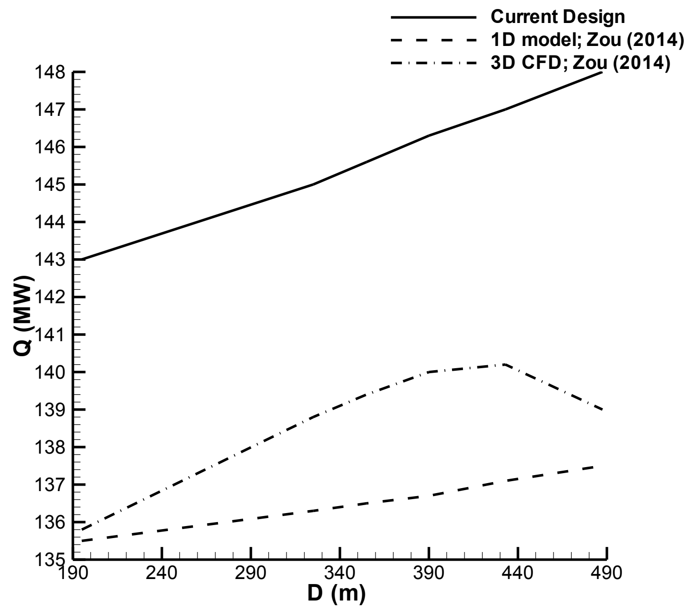

Figure 3: Heat transfer from the current design versus that of Zou (2014) for an identical tower size and height.

Note that about 7% increase in net heat dump can be expected based on the design offered here which is significant. Moving from a tower that rejects 136 MW to a tower of identical size which can remove 143 MW is really notable. For a plant that generates power at about 40% efficiency, this 7MW higher heat duty translates into about 3MW more generation or about 3% higher net efficiency. With increasing costs and demand for electricity, the economical saving associated with this saving is even more valuable in the long run. Another interesting observation can be made when one compares the number of tubes used in Zou et al. (2012)'s design which is 26244 tubes whereas the proposed design only uses about 7000 tubes. This is significant saving in material and also, in the long run, maintenance cost. For a more comprehensive understanding of the problem, Fig. 4 is presented to show the total heat transfer versus the tower height. Equation (19) is used to plot the required height for a tower to dump 135 MW of heat with and without solar enhancement for a single row metal foam heat exchanger. The predicted data is also curve-fitted for easier use and future application as the total heat transfer is implicit in Eq. (19). The curve fit is simple, implies a $H \sim U^2$ law, and reads

$$Q_a \cong 40H^{0.26} \tag{20}$$

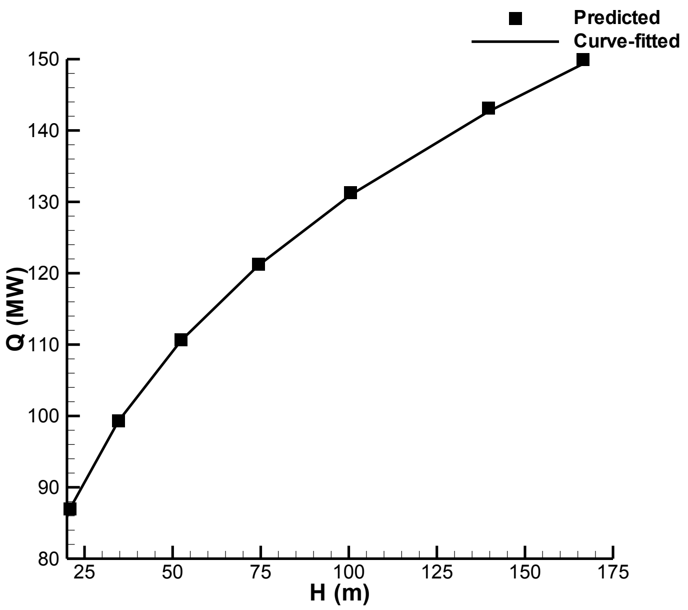

Figure 4: Heat transfer versus height for current design with the tower details the same as that of Zou (2014) except for *H*.

The height is shown versus the sunroof diameter in Fig. 5. As seen, increasing the sunroof diameter leads to lower tower height. In the limit, when $D \to \infty$, then one expects $H \to 0$, as Eq. (19) indicates. Interestingly, when the sunroof diameter is tripled, then the tower height is reduced by an order of magnitude. This is significant tower cost reduction which will be balanced with the extra sunroof cost. Obviously, factors like easy installation at a lower height and the possibility of selecting a less expensive sunroof material as opposed to concrete tower structure need to be taken into account. A cost optimization is not, however, the scope of this study and can be postponed to a future study.

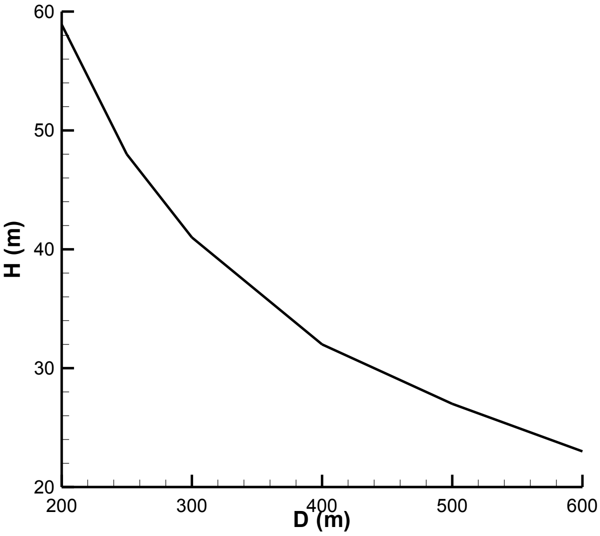

Figure 5: Required tower height for given sunroof diameters; other tower geometrical constrains are as those of Zou (2014).

Figure 6 uses the same design parameters as those of Zou et al. (2012) for the tower size to indicate the effectiveness of solar after-heating. Note that the driving force is proportional to the air temperature rise. The air temperature increases as a result of heat transfer from the cycle fluid to the air and then the heat gain from the sunroof. While the former has to be augmented to meet the heat rejection need, any increase in the latter can compensate for the tower height. As such, the designer needs to keep a close eye on the comparison between the two heating scenarios to come up with an optimal design. In generating Fig. 6, different velocity values are assumed, regardless of tower height, to give an idea of the effectiveness of solar after-heating. Expectedly, solar enhancement is more pronounced at lower air flow rates as a result of less efficient heat removal from the heat exchangers as opposed to less flow rate-dependent heat gain of the air stream under the sunroof. For instance, at low air flow rates, with a short sunroof diameter of about 200 m (where the base tower diameter is 100 m), the air temperature rise due to solar heating is about half of that caused by the heat exchangers. Even with that, one expects a total air temperature difference of about 1.5 times that of heat exchangers only (according to Eq. (5) above). With everything else unaltered, which does not necessarily have to be the case, the corresponding tower height can now be reduced by about 30%. Nonetheless, higher flow rates, say when the air speed is about 3 m/s, like the case studied by Zou et al. (2012), the enthalpy rise as a result of solar heating is only about 20% that caused by the heat exchangers. One notes that the above numbers given as reduction in height are only accurate if the NDDCT uses the same heat exchangers, here single row, that does not increase the air temperature by as much as a multi-row heat exchanger does; yet again more rows of the heat exchangers bring in additional flow resistance and make it hard for a comparison of this nature.

Interestingly, these are results for a single row foamed design. As the foams do not have to be bundled densely to lead to high heat transfer rate, see Hooman (2014), the second row can be spaced so widely that the pressure drop be only double that of a single tube which is still lower than the pressure drop of a two-row dense bundle due to local blockage in the heat exchanger.

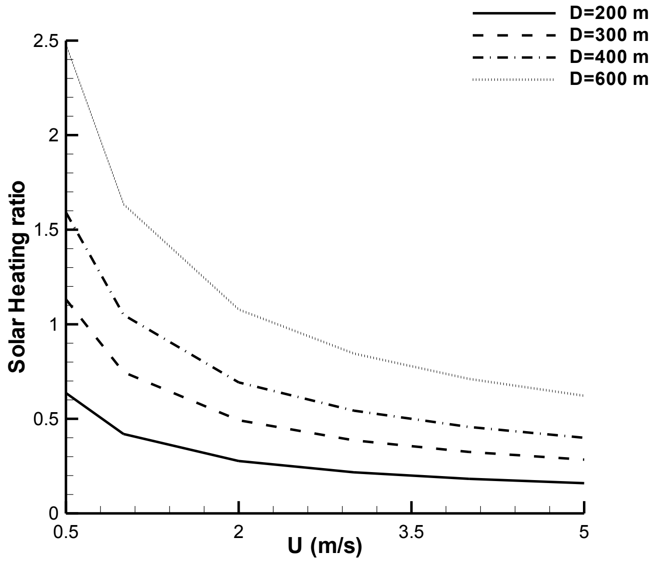

Figure 6: Solar heating ratio defined as $\Delta T_s/\Delta T_{hx}$ for different air velocity values.

4. CONCLUSION

A theoretical analysis was conducted to investigate the heat removal from a geothermal power plant using air-cooled condensers. An advanced design, using metal foam heat exchangers, is suggested. Being different from existing finned-tube heat exchangers, the optimal design for such foamed heat exchangers is completely different from the conventional bundles lending themselves well to a SENDDCT. Optimal layout for a SENDDCT with a widely-spaced single row foamed-tube bundle is obtained and the results are compared with those existing in the literature for multi-row finned tube bundles applied in SENDDCTs. It was noted that shorter towers with large base diameters can be realized using sparse arrangement of the foam-wrapped tubes in the tower. Increase in the heat transfer and decrease in the tower height are then quantified and discussed. Results comparable to a case study are presented but the formulation is generic to cover other cases of interest and potentially form the condenser for geothermal power plants constructed in sunny arid areas in the world.

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