Proceedings of ES2008
Energy Sustainability 2008
August 10-14, 2008, Jacksonville, Florida USA

ES2008-54335

AN ESTIMATION OF THE PERFORMANCE LIMITS OF DRY COOLING ON TROUGH-TYPE SOLAR THERMAL PLANTS

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ABSTRACT

The southwestern US is an ideal location for solar power plants due to its abundant solar resource, while there is a difficulty in implementing wet cooling systems due to the shortage of water in this region. Dry cooling could be an excellent solution for this, if it could achieve a high efficiency and low cost as wet cooling. Some dry cooling systems are currently in operation, and investigations of their performance have been reported in the literature.

This paper looks into the limits to the power production implicit in dry cooling, assuming that improvements might be made to the system components. Use of higher performance heat transfer surfaces is one such possible improvement.

We have developed a model of a fairly typical, but simplified, solar trough plant, and simulated thermodynamic performance of this with the software Gatecycle. We have examined the power generation and cycle efficiency of the plant for the Las Vegas vicinity with conventional wet cooling and conventional dry cooling cases considered separately using this software. TMY2 data are used for this location for this purpose. Similarly, the same studies are carried out for “ideal” cooling systems as a comparison. We assumed that in the ideal dry cooling system, the condensing temperature is the ambient dry bulb temperature, and in the ideal wet cooling system, it is the ambient wet bulb temperature. It turned out that the ideal dry cooling system would significantly outperform the conventional wet cooling system, indicating the possibility of the dry cooling system being able to achieve increased performance levels with component improvements.

INTRODUCTION

As one of the major condensing options, dry cooling technology has earned a significant place in the power generation industry since it emerged several decades ago. A detailed understanding of dry cooling systems is very important on either design or any improvement to them.

The advantages of dry cooling technology have been appreciated more since environmental problems are being viewed with more importance. Industries are investing in dry cooling rather than in cheaper wet cooling systems at some locations mostly to conserve the water resource (Johnson and Maulbetsch, 1979) (Hintzen and Benzing, 1999). Infinite availability of air as the cooling medium, less pollution, free choice of location, and the simplified approval procedure are other issues that impact the choice of this cooling option. (Hintzen and Benzing, 1999) The air-cooled system performs very well at low ambient air temperature; however, it becomes synonymous with lower efficiency and lower plant output when the ambient temperature goes high. Wet cooling usually demonstrates lower heat rate performance.

There have been many reports on the efforts that have been made to improve the performance. For example, as early as the 1970s, an economic optimization option was given that a dry tower plant would relegate a portion of its generation capability to other plants within the network during summer months. (Leung, 1973) Later a new concept of power plant “heat-sink system” which employed the combination of a conventional wet-tower and a conventional dry-tower to reduce wet cooling-tower makeup-water requirements in water-short areas was considered. (Larinoff and Forster, 1975) And a system that employed ammonia as the heat rejection fluid rather than water was proposed. (McHale et al., 1979) Later a concept was proposed which aimed to achieve highest possible thermal efficiency at high temperature by precooling a portion of the air flow with water and causing only this portion to act on the coldest part of the heat exchange surface. (Oplatka, 1981) In 1986, the Electric Power Research Institute studied an advanced ammonia-based cooling system for power plants which demonstrated safety and reliability, responding smoothly to power plant fluctuations, with a much lower cost. (Electric Power Research Institute, 1986) Others showed the effectiveness of finned heat-pipes that are ammonia-filled and lined with capillary-wick material applied to dry cooling systems. (Azad and Karimeddini, 1986) All of the previous investigations have tried to improve performance of existing dry cooling systems by either preprocessing the working fluid or making up the loss with the assistance of other systems. However, work has seldom been done, to our knowledge, in the detailed study on dry cooling systems themselves.

Many factors or operating parameters affect the performance of dry cooling systems. Among these is the dry bulb temperature which is the major environmental factor that affects the condensing performance of dry cooling systems. The dry bulb temperature changes constantly. For a solar trough power plant with a dry cooling system, weather has a major impact on plant performance. Abundant sunshine in summer could provide more energy for the power plant, but associated high ambient temperatures may decrease the power output of the turbine. Appropriate changes to the heat transfer surface geometry used in air-cooled condensers could result in improved condensing efficiency.

The aim of this paper is to investigate the influence of these factors on dry cooling systems which in turn affect the power output of the solar plant. Hourly performance and power output are calculated as well. In addition, a comparison with wet cooling systems is conducted.

NOMENCLATURE

a_c b_c	Coefficient for conventional wet cooling systems
a_i b_i	Coefficient for ideal wet cooling systems
T_d	Ambient dry bulb temperature, °C
φ	Relative humidity
$\eta_{d,c}$	Cycle efficiency of the power plant with the conventional dry cooling system
$\eta_{d,i}$	Cycle efficiency of the power plant with the ideal dry cooling system
$\eta_{w,c}$	Cycle efficiency of the power plant with the conventional wet cooling system
$\eta_{w,i}$	Cycle efficiency of the power plant with the ideal wet cooling system

ANALYSIS

The major factors that affect the performance of cooling towers are the ambient conditions. These include the dry bulb temperature for dry cooling systems and both dry bulb temperature and relative humidity for wet cooling systems. When the dry bulb temperature increases, the performance of the cooling towers decreases. We wished to find how the dry bulb temperature affects the performance of the cooling towers as well as how much the relative humidity makes wet cooling towers different from dry cooling towers. Gatecycle software was used to simulate a simple Rankine cycle with either type of cooling tower performing under the same ambient conditions. Gatecycle is a PC-based software application published by GE Company that performs detailed, steady-state design and off-design analyses of thermal power systems. Before using it on dry and wet cooling system simulation, verification is conducted on its reliability. A case for a Rankine cycle with dry cooling system was designed in Gatecycle and run under a certain ambient conditions. Meanwhile a separate Matlab code was written for this cycle to do the calculation. A very small difference was found between the results of the codes. This could have been due to the assumptions made in developing the Matlab code.

In Gatecycle, two simple Rankine cycles were established separately, one with a conventional air-cooled condenser and another with a conventional wet cooling tower. The computational modules for the turbine, pump, boiler, ACC (air-cooled condenser) and wet cooling tower were picked from the module pools in Gatecycle, and connected to form a Rankine cycle as shown in Figures 1 and 2. Each module in both models is run in a design mode to calculate its physical size from key specified performance parameters. These two models are designed with the same specified performance parameters and under the same ambient conditions, which is 0 °C for ambient dry bulb temperature and 10% for relative humidity. The key specified performance parameters are shown in Table 1.

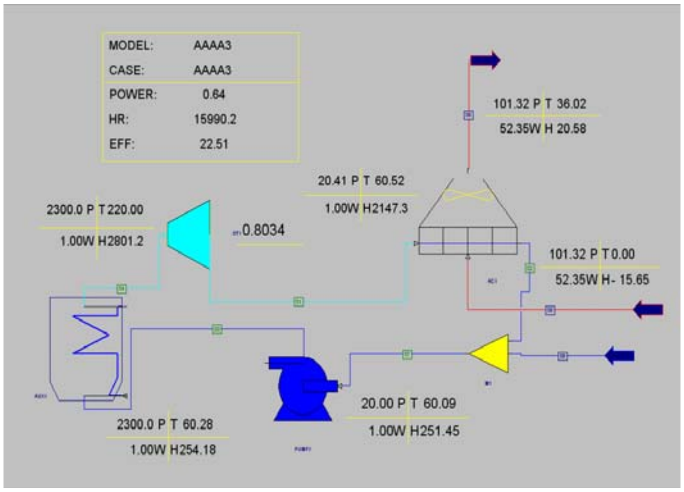

Figure 1 The model of Rankine cycle with dry cooling system in Gatecycle

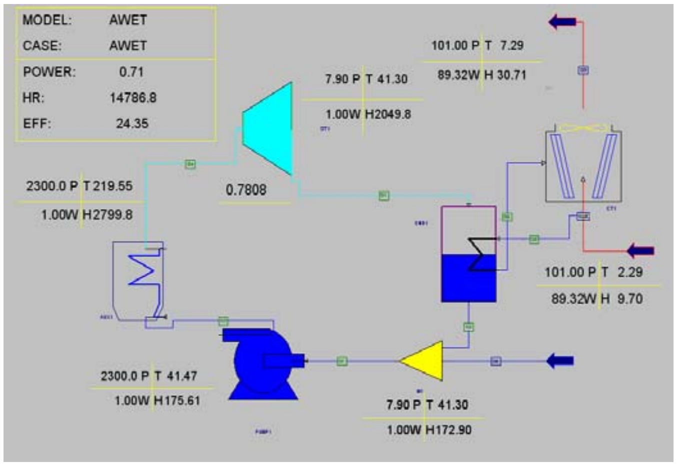

Figure 2 The model of Rankine cycle with wet cooling system in Gatecycle

The conventional ACC (air-cooled condenser) parameters set up are shown in the

Table 2. During its design process, the face velocity of inlet air is empirically selected to be a typical value for ACC, as is suggested in the Gatecycle help file, and other parameters in ACC are default values provided in Gatecycle.

Table 1 The key specified performance parameters

Rankine Cycle Parameters	Values
Mass flow rate of steam	1 kg/s
Boiler pressure	2300kpa
Condensing pressure at the design mode	11 kpa
Boiler efficiency	90%
Turbine efficiency	90%
Quality of the steam at the outlet of the boiler	0
Quality of the steam at the outlet of the air-cooled condenser	1
Condensing temperature at the design mode	48.1°C

Table 2 Parameters designed for conventional ACC (air-cooled condenser)

ACC Parameters	Values
Desired saturation pressure	11 kpa
Inlet air temperature at the design mode	0
Mass flow rate of air	140.75 kg/s
Total heat transfer surface area of ACC	984.69 m^2
Face velocity of inlet air	3 m/s
Length of ACC tubes	7.0778m
Number of tubes per row	19
Number of rows	3
Outer diameter of the tubes	25 mm
Inner diameter of the tubes	21 mm
Diameter of the fin	40 mm
Thickness of the fin	1.016 mm
Pitch of the fin	2.822 mm

Table 3 Parameters designed for the conventional wet cooling tower

Wet Cooling Tower Parameters	Values
Inlet air temperature at the design mode,	0 °C
Fixed cooling water temperature rise	11.111 °C
User input values for U	2.85 kJ/s-m^2-K
Pressure drop	no
Outer diameter of the tubes	22.2, 25.4mm
Velocity inside the tube	2.1336 m/s
Tube material	Stainless steel type 304
Tube gauge	16
Desired pressure	11 kpa

The conventional wet cooling counterpart is also built in the design mode in Gatecycle, and the parameter set up is shown in the Table 3. All the parameters use default values in Gatecycle, except that the desired pressure is set to be 20 kpa to match the desired requirement of the Rankine cycle system.

Once the design case has been created, this case can be referenced by the same icon running in off-design mode in another case. This enables you to analyze the performance of a "physically-based" equipment icon under off-design operating conditions. Studies of two models are carried out in the off-design mode, where the ambient conditions are allowed to vary, and the performance of the Rankine cycle is calculated. The study of the influence of ambient dry bulb temperature on the conventional dry cooling system mainly focused on the condensing temperature, and the condensing pressure inside ACC as well as the Rankine cycle efficiency. They are calculated in a series of cases by varying the dry bulb temperatures from 5 to 40 °C with values of the other parameters held constant. In comparison, the influence of the ambient conditions on the Rankine cycle efficiency of the wet cooling counterpart is studied as well.

In addition, for each cooling system, an ideal case is simulated in order to see how much different the conventional case performance is from it. The ideal cooling systems are set up by making the condensing temperature equal to the ambient dry bulb temperature for the dry system and the wet bulb temperature for the wet cooling system. In each dry cooling system case, performance of the Rankine cycle under different dry bulb temperatures was simulated, and in each wet cooling system case, the impact of both dry bulb temperature and relative humidity is included.

RESULTS AND DISCUSSION

A series of tests of Rankine cycle and ACC performance are run in Gatecycle under different dry bulb temperatures. The influence of ambient dry bulb temperature on the condensing temperature for a conventional dry cooling system is plotted in Figure 3. Likewise the effect of ambient temperature on condenser pressure is shown in Figure 4. Finally the effect of ambient temperature on Rankine cycle efficiency is shown in Figure 5.

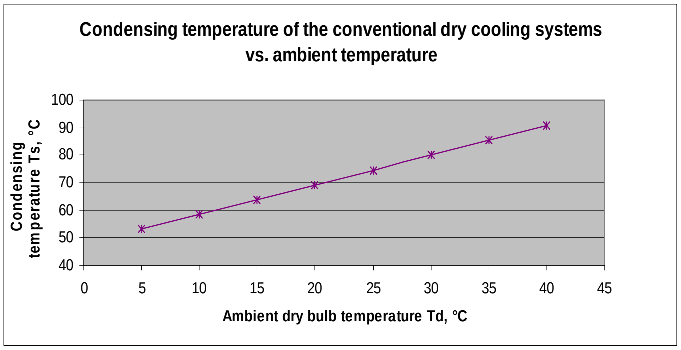

Figure 3 The temperature inside condenser vs. ambient temperature for a conventional dry cooling system is shown

Inspection of Figure 3 shows that the condensing temperature is greatly affected by the ambient dry bulb temperature as expected. It is obvious that the condensing temperature is linearly increasing as the ambient temperature increases. Since the condensing pressure is a function of condensing temperature only when the steam is at saturation state, this results in an increasing condensing pressure at the same time, which is shown in Figure 4.

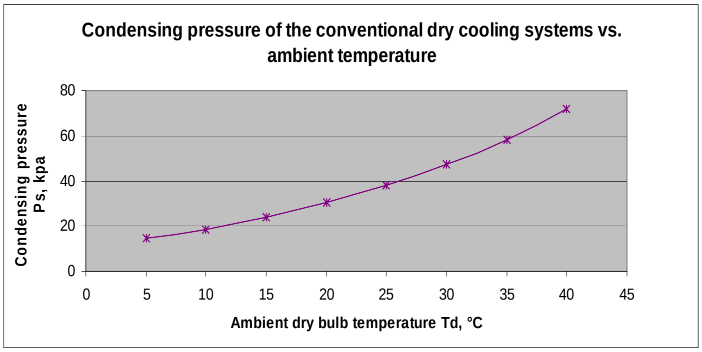

Figure 4 Pressure inside condenser vs. ambient temperature for a conventional dry cooling system

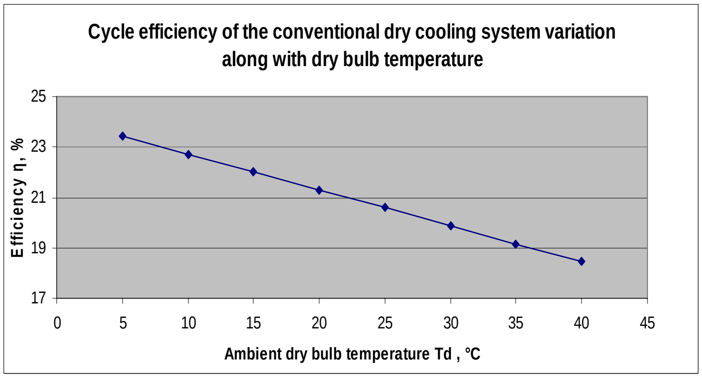

Figure 5 The cycle efficiency vs. ambient temperature for a power plant with the conventional dry cooling system

Figure 5 indicates a great decrease in Rankine cycle efficiency when the ambient dry bulb temperature goes up. As shown here, during winter time when the ambient temperature goes down to 5 °C, the efficiency of the power plant could reach 23.4%, while during summer when the ambient temperature could reach as high as 45 °C in Las Vegas, the efficiency of the power plant could decrease 21.4% compared to the performance in winter. This is the efficiency of a simple Rankine cycle. For a more complete power plant system with improvements on dry cooling systems, the efficiency would be different and the sensitiveness to the ambient temperature might decrease, as is described in the literature noted earlier in this paper.

For each dry cooling system configurations, several cases are analyzed in using in Gatecycle under different dry bulb temperatures, and for each wet cooling system considered, a series of tests are run under both different dry bulb temperatures and relative humidity. The trend of Rankine cycle efficiency varying along with the ambient dry bulb temperature and relative humidity is plotted for each case. This is shown in Figure 6, where cycle efficiency of the conventional wet cooling system variation with ambient temperature and relative humidity can be noted.

Figure 6 shows a similar efficiency trend of a Rankine cycle with the conventional wet cooling system under the influence of ambient temperature, no matter the value of the relative humidity. Figure 6 also indicates that relative humidity has a larger impact on Rankine cycle efficiency when the ambient temperature is higher. Also, the higher the relative humidity is, the lower the Rankine cycle efficiency will be, all other factors being the same. But even under the highest relative humidity here, the Rankine cycle efficiency with a wet cooling system is still higher than that of the power plant with a dry cooling system.

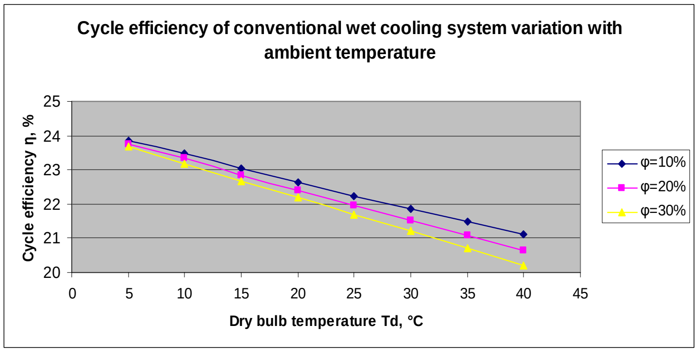

Figure 6 Cycle efficiency of the conventional wet cooling system variation with ambient temperature and relative humidity

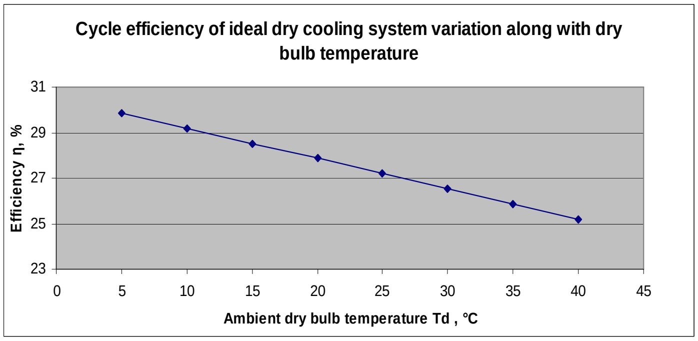

Figure 7 Cycle efficiency of the ideal dry cooling system variation with ambient temperature

Figure 7 shows a similar trend for the ideal dry cooling system. However, Rankine cycle efficiency is much higher in a power plant with the ideal dry cooling system than that with the conventional dry cooling system. Note that this yields the same results in Figure 8 for the wet cooling counterparts. It is found that the power plant with an ideal dry cooling system could provide a comparable efficiency to that of the ideal wet cooling system. This gives us hope that a dry cooling system could perform nearly as well as the wet cooling system, even during severely hot days. Keep in mind that round-tube-and-round-fin surface geometry is assumed for the dry cooling system. A more advanced surface could be further studied to see how it might improve cycle performance.

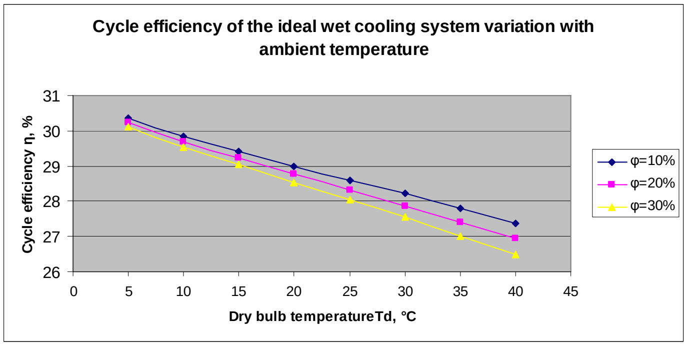

Figure 8 Cycle efficiency of the ideal wet cooling system variation with ambient temperature

In the conventional and ideal dry cooling system cases, the ambient dry bulb temperature is the only variable that affects cycle performance. With this in mind, the relation between the Rankine cycle efficiency and ambient temperature could be easily obtained from Figure 5 and 7:

For the conventional dry cooling system case:

$$\eta_{d,c} = -\ 0.1468\ T_d + 22.526 \tag{1}$$

and for the ideal dry cooling system case:

$$\eta_{d,i} = -\ 0.1324\ T_d + 30.503 \tag{2}$$

In wet cooling systems, both dry bulb temperature and relative humidity are considered at the same time. The thermal efficiency is assumed to be a linear function of dry bulb temperature, in which the coefficients are assumed to be a function of relative humidity.

$$\eta_{w,c} = a_c\,(\varphi)\ T_d + b_c\,(\varphi) \tag{3}$$

$$\eta_{w,i} = a_i\,(\varphi)\ T_d + b_i\,(\varphi) \tag{4}$$

Then for the conventional wet cooling system, the thermal efficiency as a function of both dry bulb temperature and relative humidity is obtained from Figure 6:

$$a_c(\varphi) = -\ 0.102\ \varphi - 0.0684 \tag{5}$$

$$b_c(\varphi) = -\ 0.305\ \varphi + 24.26 \tag{6}$$

Similarly for the ideal wet cooling system:

$$a_i\,(\varphi) = -\ 0.095\ \varphi - 0.074 \tag{7}$$

$$b_i\,(\varphi) = -\ 0.480\ \varphi + 30.75 \tag{8}$$

Thus, the year round thermal efficiency variation of the power plant with wet cooling system is calculated using the hourly dry bulb temperature and relative humidity provided in TMY2 data for Las Vegas. And the total power generated in a whole year using the direct normal solar data files for this location is calculated and is shown in Table 4.

Table 4 Comparison of yearly power generation of power plants with different conventional condenser systems

Power Plant Cases	Power Generation in a Whole Year (MWh)
With regular dry cooling system	38.2
With regular wet cooling system	43.4
With ideal dry cooling system	54.2
With ideal wet cooling system	55.8

CONCLUSIONS

A study of the performance of dry cooling systems under different ambient conditions as well as their wet cooling counterpart leads to the following conclusions:

1. The power plant efficiency increases as the ambient temperature decreases for both cases with dry cooling systems and wet cooling systems. The difference is that dry bulb temperature is the only factor that affects the efficiency for the dry cooling system, while the wet cooling system will be affected by both dry bulb temperature and relative humidity, and the higher the relative humidity, the lower the efficiency.
2. When the ambient temperature is low, a dry cooling system performs nearly as well as a wet cooling system, however, when the temperature goes higher, dry cooling systems will result in a lower efficiency than the wet counterparts.
3. An ideal dry cooling cycle is defined, where the ambient dry bulb temperature and condensing temperature are the same. Likewise, an ideal wet cooling cycle is defined, where the ambient wet bulb temperature and the condensing temperature are the same
4. The difference between the cycle performance with the ideal dry cooling system and with the ideal wet cooling system is small, which gives us hope that improvements can be made to dry cooling systems to increase their performance.

ACKNOWLEDGMENTS

Funding supplied by the National Renewable Energy Laboratory under the project “A Study Related to Developing Solar Power Systems in Nevada” is greatly appreciated.

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