ACHIEVING NEAR-WATER-COOLED POWER PLANT PERFORMANCE WITH AIR-COOLED CONDENSERS

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ABSTRACT

Power plants using air-cooled condensers suffer a 5 – 10% plant-level efficiency penalty compared to plants with once-through cooling systems or wet cooling towers. In this study, a model of a representative air-cooled condenser (ACC) system is developed to explore the potential to mitigate this penalty through techniques that reduce the air-side thermal resistance, and by raising the air mass flow rate. The ACC unit model is coupled to a representative baseload steam-cycle power plant model. It is found that water-cooled power-plant efficiency levels can be approached by using enhanced ACCs with a combination of significantly increased air flow rates (+68%), reduced air-side thermal resistances (–66%), and air-side pressure losses near conventional levels (+24%). Emerging heat-transfer enhancement technologies are evaluated for the potential to meet these performance objectives. The impact of ambient conditions on ACC operation is also examined, and two hybrid wet/dry cooling system technologies are explored to improve performance at high ambient temperatures. Results from this investigation provide guidance for the adoption and enhancement of air-cooled condensers in power plants.

KEYWORDS

Air-Cooled Condenser, Steam Condenser, Power Plant, Dry Cooling, Hybrid Cooling, Water Conservation

NOMENCLATURE

A	heat transfer area	$[\text{m}^2]$
A_{tot}	total air-side heat transfer area (tube and fins)	$[\text{m}^2]$
A_{tube}	tube outside heat transfer area	$[\text{m}^2]$
D_H	hydraulic diameter	$[\text{m}]$
h	convective heat transfer coefficient	$[\text{W m}^{-2} \text{K}^{-1}]$
h_{eff}	effective heat transfer coefficient	$[\text{W m}^{-2} \text{K}^{-1}]$
ITD	initial temperature difference across condenser	$[\text{°C}]$
K_{dj}	jetting loss coefficient	$[-]$
K_{do}	downstream loss coefficient	$[-]$
K_o	outlet loss coefficient	$[-]$
$K_{\text{θt}}$	total loss coefficient	$[-]$
K_{up}	upstream loss coefficient	$[-]$
$\dot{m}_{\text{air}}$	air mass flow rate through air-cooled condenser cell	$[\text{kg s}^{-1}]$
$\dot{m}_{\text{steam}}$	steam mass flow rate through air-cooled condenser cell	$[\text{kg s}^{-1}]$
P	fan power consumption	$[\text{W}]$
P_s	heat-transfer-area specific fan power consumption	$[\text{W m}^{-2}]$
$T_{\text{air,in}}$	air inlet temperature to the condenser	$[\text{°C}]$
$T_{\text{air,out}}$	air outlet temperature from the condenser	$[\text{°C}]$
T_{amb}	ambient air temperature	$[\text{°C}]$
$T_{\text{steam,in}}$	steam inlet temperature to condenser	$[\text{°C}]$
U_s	heat-transfer-area specific volumetric air flow rate	$[\text{m s}^{-1}]$
V	volumetric flow rate	$[\text{m}^3 \text{s}^{-1}]$

1. INTRODUCTION

1.1 Water Resources and the Role of Power Plant Condensers

Population growth and increasing energy intensity throughout much of the world are placing increasing strain on limited freshwater resources that are needed for residential use, power generation, industry, and agriculture. These factors and ecological considerations have led to increasing pressure on thermoelectric power generation utilities to reduce water withdrawals and consumption, even as demand for electricity increases. These conflicting demands are particularly difficult to satisfy because areas with the highest population growth, increasing water usage, and increased electricity demand coincide with areas with scarce water supplies (EIA, 2013; Kenny *et al.*, 2009; USCB, 2012). Additionally, seasonal periods of peak power-generation demand often coincide with drought conditions.

Power plants currently account for 41% of US freshwater withdrawals (Kenny *et al.*, 2009), over 90% of which is employed for condenser cooling (EPRI, 2011). Historically, simple and low-cost water-to-steam once-through condensers were widely employed, and currently account for 43% of the US generation fleet. However, thermal pollution from the high return water temperatures (typically 10°C above intake temperatures) and the water withdrawal rates required to achieve even such high return temperatures ($75 - 150 \text{ m}^3 \text{ MWh}^{-1}$, $1 \text{ m}^3 \text{ MWh}^{-1} = 0.26 \text{ kgal MWh}^{-1}$) have led to EPA restrictions on new construction (EPA, 2002). Wet cooling towers represent an alternative condenser technology with comparatively low water withdrawal requirements ($2 - 28 \text{ m}^3 \text{ MWh}^{-1}$), and are currently installed in 42% of US thermoelectric power plants (DOE and NETL, 2010; Kenny *et al.*, 2009). While thermal pollution of watersheds is a less significant concern for wet cooling tower installations, the increased water consumption through evaporation ($2.3 \text{ m}^3 \text{ MWh}^{-1}$ compared to $0.8 \text{ m}^3 \text{ MWh}^{-1}$ in once through systems) may be untenable in areas with water availability concerns. Cooling ponds (14% of U.S. plants) also operate

through evaporative cooling, and have similar water consumption rates. Although fresh water supplies can be conserved by increasing plant efficiency, recycling water supplies, using waste water, and other avenues, dry cooling systems have the potential to almost eliminate power plant water usage. Air-cooled condenser (ACC) technology is not yet widely employed in the U.S. (only in 1% of U.S. plants), but is expected to see increased adoption due to competing water demands and water conservation regulations.

1.2 Air-Cooled Condenser Description

In the US, direct-coupled mechanical-draft air-cooled condensers have been utilized in all current dry cooling systems (EPRI, 2005). In these systems, steam exiting the turbine is routed to the air-cooled condenser through a series of large horizontal ducts running along the top of A-frame condensers. Each row of A-frame condensers consists of a number of cells. A single ACC cell has finned tubes arranged in parallel along the inclined walls of the A-frame unit. Many current ACC cell designs utilize a single row of finned tubes, with each tube consisting of a rectangular carbon steel channel with aluminum fins (EPRI, 2005; Kröger, 1998). Steam enters the ACC cell through the large steam duct, condenses as it flows down the inclined tubes forming the walls of the A-frame, and is then collected in a condensate line at the bottom. A typical ACC cell has a footprint of 12×12 m, with finned tubes 9 to 12 m long and an apex angle of 60 degrees (EPRI, 2005; Kröger, 1998). Each finned tube has approximate dimensions of 25×190 mm, with 25 mm tall fins (EPRI, 2005; Yang *et al.*, 2012b). Air is driven through the tube banks and fins by large axial-flow fans approximately 9 m in diameter (EPRI, 2005; Kröger, 1998). ACC condensers are generally placed 20-50 m above ground level, and are enclosed by wind walls to reduce the impact of wind and potential air recirculation (EPRI, 2005; Kröger,

Figure 1. Schematics of (a) a full ACC system, (b) an ACC cell, (c) an individual steam tube.

1998). Schematics of an ACC assembly, ACC cell, and an individual finned steam tube are presented in Figure 1.

1.3 Air-Cooled Condenser Challenges

Despite the reduced water usage in dry cooling systems, limited market penetration has been achieved in the US due to substantial tradeoffs in terms of cost and performance. Air-cooled condensers require substantially higher capital investment than wet-cooled condensers because they incorporate larger heat exchangers, have huge fin areas, and necessitate additional support structures (Putman *et al.*, 2002). Overall, installation and operational costs for ACC systems are currently 3.5 – 5 times as much as for wet cooling systems (2008). Typical levelized power production costs for plants with ACCs are $\$3\text{--}6 \text{ MWh}^{-1}$ higher (up to $\sim 15\%$) than for plants utilizing wet cooling (Zhai and Rubin, 2010). However, expected increases in water usage costs could quickly eliminate this gap. Zhai and Rubin (2010) estimated that increasing water costs from a baseline of $\$0.26 \text{ m}^{-3}$ to $\$1.64 \text{ m}^{-3}$ would result in equivalent costs, and (2008) found that, depending on conditions, an increase in water cost to $\$0.53 \text{ m}^{-3}$ – $\$1.06 \text{ m}^{-3}$ would be sufficient to eliminate this gap.

ACCs suffer a performance penalty relative to wet cooling systems due to the poor thermal transport properties of air, which is exacerbated by their greater relative performance degradation at elevated ambient temperatures. ACC air-side heat transfer coefficients (typically $\sim 35 \text{ W m}^{-2} \text{ K}^{-1}$) are generally much lower than values for the water or evaporative-sides of wet cooled condensers. Air also has a much lower thermal capacity than water. At atmospheric pressure, air has a volumetric specific heat of $1.1 \text{ kJ m}^{-3} \text{ K}^{-1}$, while water has a specific heat of $4,200 \text{ kJ m}^{-3} \text{ K}^{-1}$ and a latent heat of vaporization of $2,252,000 \text{ kJ m}^{-3}$. Thus, it is necessary to supply substantially more air than water to provide the same thermal capacity for heat removal from the condenser, which is accompanied by a large parasitic fan power requirement. Blanco-Marigorta *et al.* (2011) found that with a steam condensation temperature of 37°C , this results in an exergetic efficiency of just 26% for an air cooled condenser, compared to 63% for a wet condenser. This thermal capacity difference results in ACCs requiring a higher initial temperature difference ($\text{ITD} = T_{\text{steam,in}} - T_{\text{air,in}}$) than water cooling systems with cost-effective designs. Raising the ITD by increasing the steam condensation pressure results in a greater available air thermal capacity and increases the temperature difference driving the heat transfer process. However, this also results in increased steam turbine backpressure, and thus reduces the steam cycle efficiency and power output. Based on results from the present modeling effort, a 3 K increase in ITD leads to a 1.1% reduction in power generation. In order to match the advantages of plants cooled by wet condensers, the steam turbine backpressure must be reduced through the development of high performance and economical ACCs.

In addition to this performance penalty under design conditions, ACC performance can be sensitive to operating conditions, such as ambient temperature, wind, rain, hail, or solar radiation (Kröger, 1998). High ambient temperatures increase the steam condensation pressure, lowering

the power plant output. Periods with high ambient temperatures typically coincide with peak electricity demand, penalizing power production when requirements are highest. Although ACC sizes could be increased to compensate, this would substantially increase costs. Zhai and Rubin (2010) found that an increase in ambient air temperature from 15 to 25°C increases the required dry cooling system size by 40% to reach a fixed power generation output, thereby increasing the capital and levelized costs by over 35%. Wind also has the potential to reduce ACC heat transfer rates by reducing fan performance or causing hot plume recirculation (Duvenhage and Kröger, 1996; Owen and Kröger, 2013). Very high wind speeds can even lead to air backflow through the fans (Spuy *et al.*, 2009). These effects are typically limited by enclosing the ACC cells with a wind wall (EPRI, 2005), although other methods, such as flow guiding devices, have been proposed (Yang *et al.*, 2012a). Even with such devices, wind effects can significantly degrade ACC performance (Wilber and Zammit, 2005).

1.4 Present Study

Given the aforementioned limitations, the U.S. Environmental Protection Agency does not currently consider dry cooling systems to be the best technology available for minimizing the environmental impact of power plant cooling systems (EPA, 2002). The performance penalty associated with dry cooling systems leads to increased primary energy consumption to generate the same level of power output, increasing associated greenhouse gas emissions. The EPA also recognizes that ACC performance is significantly reduced during periods of high ambient temperature, leading to power production, grid reliability, and emission concerns. Meanwhile, the higher cost of dry cooling systems provides an incentive to extend the life of existing plants, which otherwise may be replaced by new facilities with lower environmental impact. Thus,

while air cooled condensers have the potential to almost eliminate power plant water usage, additional cooling system costs and power production penalties remain significant barriers to widespread adoption. The present study examines the potential to improve ACC performance by reducing the air-side thermal resistance through the use of enhanced heat transfer techniques and by increasing air flow rates. This study quantifies the level of improvement needed for ACCs to match the performance of wet-cooled condensers, and explores the potential of hybrid wet/dry cooling technologies. A representative baseload steam power-plant model is developed to estimate the effect of condenser-level improvements on plant-level efficiency and generation gains.

2. APPROACH

A model of a representative A-frame air-cooled condenser cell was developed in Engineering Equation Solver (Klein, 2013) (Figure 1). The unit has an overall width of 12.2 m with angled tubes 10.7 m long connected to the steam duct at an apex angle of 60 degrees. Each angled tube is an elongated, nearly rectangular, carbon steel steam-flow channel with cross-sectional dimensions of 191×25.4 mm, and a wall thickness of 1.3 mm. An array of aluminum fins is located on each side of the angled tubes with a fin pitch of 0.43 fins per mm (11 fpi), fin thickness of 0.25 mm, and fin height of 25.4 mm. The total steam-side mass flow rate is 7 kg s^{-1} , while the baseline air mass flow rate is 645 kg s^{-1} , although this was varied during the study. The baseline air temperature and pressure are 30°C and 101.3 kPa, respectively, and a relative humidity of 0.25 is assumed.

The heat transfer coefficient in the air-side fin channels was estimated using data from Kakaç *et al.* (1987), as implemented in Engineering Equation Solver (Klein, 2013). For louvered fins,

the correlation developed by Kim and Bullard (2002) was used. The heat transfer coefficient of the steam flow is assumed to be constant at $15 \text{ kW m}^{-2} \text{ K}^{-1}$ (Heyns, 2008). The actual value may vary along the steam tubes and under the considered operating conditions, but the net effect on condenser performance would be small as the air- and fin-side heat transfer resistances are dominant (~96% of the total).

Under conditions with laminar air flow expected across smooth fins, the pressure drop was evaluated using the friction factor correlation from Kakaç *et al.* (1987), corrected for entrance effects using the correlations available in Shah and London (1978). For turbulent conditions, the friction factor correlation by Li *et al.* (2011) was employed. For louvered fins, a correlation by Park and Jacobi (2008) was used to predict the pressure drop.

To find the total pressure loss associated with air flow through the A-frame condenser, the approach of Kröger (1998) was adopted. In addition to the channel-flow frictional pressure losses through the heat exchanger fins, this model accounts for the minor losses due to the upstream support structure and screen, the velocity profile through the fan, the transition from the fan to the angled heat exchanger tubes, the downstream support structure, and the inlet and exit losses from the heat exchanger tubes. K-factors were determined for each of these losses using experimentally derived correlations or representative values from the literature, as available. Additionally, the model accounts for the buoyancy of the heated air flow across the ACC tubes.

The impact of flow resistances associated with the upstream support structure and screen were included in an upstream loss coefficient, K_{up} , estimated at 0.30 for this configuration (Kröger, 1998; Venter and Kröger, 1991). The outlet loss coefficient, K_o , as defined by Kröger, represents the resistance associated with the velocity profile entering the fan. This value was evaluated using the approach of van Aarde and Kröger (1993), and was found to be 7.96 under

baseline operating conditions. The flow loss associated with the transition from the fan to the angled heat exchanger tubes was included in the jetting loss coefficient, K_{dj} , calculated to be 2.63 under baseline operating conditions using the approach of van Aarde and Kröger (1993). The downstream loss coefficient, K_{do} , accounts for the flow resistances associated with obstructions downstream of the fan inside the A-frame, such as the walkway and support structure, and was estimated to be 0.10 for this configuration (Kröger, 1998; Venter and Kröger, 1991). The minor losses associated with the inlet and exit resistances of the finned heat exchanger tubes, including the impact of the angled flow relative to the heat exchanger bundle, were accounted for in a total loss coefficient, $K_{\theta t}$, based on the approach of van Aarde and Kröger (1993). In this study, the fin loss, jetting loss, and outlet loss components of $K_{\theta t}$ were reported separately. Under baseline operating conditions, this results in a total loss coefficient of 3.51. Each loss coefficient was combined with the appropriate density and velocity to calculate the contribution to pressure drop. Finally, the pressure change associated with the buoyancy of heated air rising across the condenser is calculated to be 4.7 Pa under baseline operating conditions ($T_{air,in} = 30^{\circ}\text{C}$, $T_{air,out} = 52^{\circ}\text{C}$). The values for pressure drop under baseline operating conditions are reported in Table 1. Note that the most significant contributions are due to the fan losses and pressure

Table 1. Pressure Drop Contributions at Baseline Operating Conditions

Parameter	Loss Coefficient [-]	Pressure Drop [Pa]
Upstream losses, K_{up}	0.30	0.4
Fan profile, K_o	7.96	21.9
Jetting losses, K_{dj}	2.63	7.0
Downstream losses, K_{do}	0.10	4.5
Fin entrance/exit losses, $K_{\theta t}$	3.51	21.1
Frictional smooth fin channel-flow losses	-	28.8
Frictional louvered fin channel-flow losses	-	61.4
Property change	-	-4.7

drops associated with the heat transfer fins.

To assess the impact of the air-cooled condenser on overall power plant operation, the A-frame condenser model was coupled to a Rankine cycle model representative of a 500 MW power output baseload steam power plant. In the plant cycle model, the boiler pressure was fixed at 16.5 MPa, and superheated steam was delivered to the turbine at 550°C. The condenser temperature and pressure varied depending on the performance of the dry cooling system and ambient conditions, with the condenser exit quality being fixed at zero. Isentropic efficiencies for the steam turbine, condensate pump, and ACC fans are specified based on representative values from the literature:

- Steam turbine: 85% (Srinivas *et al.*, 2008)
- Condensate pump: 95% (Ajundi, 2009)
- ACC fans: 55% (Bredell *et al.*, 2006)

The overall plant efficiency was calculated accounting for generated turbine work, condensate pump power, heat input to the boiler, and required ACC fan power.

3. RESULTS AND DISCUSSION

3.1 Baseline ACC and Plant Model Results

In the baseline ACC model, 0.0219 kg s⁻¹ of saturated vapor steam is assumed to flow into each rectangular finned tube. For the 10.7 m long steam tube with internal cross-sectional dimensions of 189 × 23 mm, the assumed condensation heat transfer coefficient of 15,000 W m⁻² K⁻¹ yields an internal heat transfer resistance of 1.55 × 10⁻⁵ K W⁻¹ per tube. The relatively thin (1.3 mm) carbon steel tube walls only contribute a minimal heat transfer resistance of 4.93 × 10⁻⁶ K W⁻¹. The corrugated aluminum full-tube width air-side fins with height 25.4 mm, thickness

0.25 mm, and pitch of 2.3 mm lead to an overall per-tube heat transfer area of 74.5 m². Assuming adiabatic fin tips and a uniform air-side convection coefficient of 37.4 W m⁻² K⁻¹, the overall fin conduction efficiency is found to be 75%. The dominant portion of the condenser heat is removed through the fins (95%). In this heat transfer path, conduction through the fins accounts for 25% of the thermal resistance (1.1×10^{-4} K W⁻¹), and convection over the fins accounts for 75% (3.3×10^{-4} K W⁻¹). Air flow directly over the bare tube walls removes the remaining 5% of the condenser heat, with a thermal resistance of 8.2×10^{-3} K W⁻¹. Thus, for the baseline modeled ACC configuration, the combined fin and air-side thermal resistances account for 96% of the total. Frictional pressure drop is neglected for the steam flow; therefore, heat transfer is calculated between the air flow and condensing steam at a uniform saturation temperature. At the baseline per-tube air and steam mass flow rates ($\dot{m}_{\text{air}} = 2.01$ kg s⁻¹, $\dot{m}_{\text{steam}} = 0.0219$ kg s⁻¹) and air inlet temperature of 30°C, a condenser pressure of 25.1 kPa ($T_{\text{sat}} = 65.1^\circ\text{C}$) is required for complete condensation. This corresponds to an ITD of 35.1 K, and 45.4 kW of heat transfer per steam tube. The air exits the condenser at 52.1°C, indicating a temperature approach of 13.0 K.

Using the minor loss and fin resistance model described in Section 2, a total air-side pressure drop of 79.2 Pa is predicted for the baseline ACC configuration and flow rate. Of this pressure drop, 28.8 Pa (36.4%) results from fin channel-flow resistance, 21.1 Pa (26.6%) is due to fin entrance/exit effects, and other minor losses account for the remainder (29.3 Pa, 37.0%). Assuming a representative fan isentropic efficiency of 55%, this would require 0.13 kW of fan input power per kg s⁻¹ of air flow (11.5 kW per kg s⁻¹ of steam flow). For the 500 MW nominal output steam power plant modeled in this study, the total required fan input power would be 5.5 MW.

In the modeled baseload 500 MW steam power plant, 473 kg s^{-1} of condensate flows from the ACC to the pump. Assuming a pump isentropic efficiency of 95%, 8.3 MW of mechanical power is required to raise the water to the high side pressure of 16.5 MPa (350°C). The pressurized water is boiled at the high side pressure and superheated to 550°C, requiring 1487 MW of thermal input. The steam flow enters the turbine, and is expanded to saturated vapor at the condenser pressure of 25.1 kPa (65.1°C). Assuming an isentropic efficiency of 85%, the turbine power output is 513.8 MW. Accounting for the pump (8.3 MW) and condenser fan (5.5 MW) power consumptions, the net delivered electrical power is 500 MW and the overall plant efficiency is 33.6%. In the following sections, the effects of varying the ACC configuration and operating parameters on ACC and overall plant performance are explored.

Figure 2. Predicted efficiency of a 500MW power plant with dry and wet cooling systems at varying water withdrawal rates.

3.2 Impact of Air-Side Convective Heat Transfer Coefficient and Flow Rate

An initial comparison of representative air-cooling, once-through liquid cooling, and evaporative cooling tower condenser systems for the modeled 500 MW baseload power plant is presented in Figure 2. Under baseline operating conditions, with the ambient air at 30°C, the wet-cooling systems generate ~6% more power than the dry cooling system. This is primarily due to the high initial temperature difference required by the dry cooling system (~35°C), which increases the steam turbine backpressure. This high ITD is necessary (1) to provide sufficient driving temperature difference for heat transfer given the high air-side thermal resistance, and (2) to provide the required temperature rise in the air stream. Decreasing the ITD requires both of these factors to be addressed.

The low air-side convection coefficients are partially compensated for by increased heat transfer area in the fins. Increasing fin area is only advantageous up to the point where the benefits from heat transfer enhancement are offset by increased pressure drop and capital costs. Thus, it is desirable to employ heat-transfer enhancement techniques that increase the air-side

Figure 3. Steam-tube heat-transfer resistances for the baseline and enhanced air-side convection cases. Bar heights represent thermal resistance through paths (combined for the parallel air-side paths: bare tube convection and fin transport). Bar widths represent fraction of heat transfer through paths (e.g., 5% through bare tube and 95% through fins in baseline case).

convection coefficient without requiring additional fin material or substantially increasing pressure drop. A number of emerging convection enhancement technologies are presented and assessed for the potential to achieve these objectives in Section 3.5. However, the convection phenomena and fin configuration cannot be fully decoupled because fin efficiency decreases with increasing convection coefficient. For example, tripling the air-side convection coefficient to $112 \text{ W m}^{-2} \text{ K}^{-1}$ would only yield a 52% reduction in total heat transfer resistance because the fin efficiency would decrease from 0.75 to 0.53 (–29%). This is indicated schematically in Figure 3 where the relative fin conduction resistance increases from 24% to 44% for the dominant portion (>90%) of heat rejection transported through the fins. Enhanced heat transfer design can thus be a challenging process, but is not the main objective of this study, which is rather to focus on

overall ACC and power plant operation, and identify performance targets and potential pathways for future investigations.

Even with no heat transfer resistance across the ACC, the ITD must still be sufficient to provide for the required rise in air temperature due to the heat of condensation, i.e., 22°C for the baseline air and steam flow rates of 645 kg s⁻¹ and 7 kg s⁻¹, respectively. Thus, approaching the typical ITD value of 20°C achieved using wet cooled condensers with ACCs requires both reduced air-side heat transfer resistance and increased air flow rate. The impact of varying both of these parameters on ITD is presented in Figure 4. Doubling the air-side convective heat transfer coefficient (from 37.4 W m⁻² K⁻¹ to 74.8 W m⁻² K⁻¹) decreases the ITD from 35.1°C to 27.6°C, while tripling this heat transfer coefficient decreases the ITD further to 25.3°C. Meanwhile, doubling the mass flow rate ratio ($\dot{m}_{\text{air}} / \dot{m}_{\text{steam}}$) reduces the ITD to 28.2°C, while tripling it reduces the ITD to 26.1°C. In both cases, there are diminishing returns for further improvements in one parameter independently of the other; it is necessary to increase both parameters simultaneously to substantially reduce the ITD. For instance, if both the air-side mass flow rate and heat transfer coefficients are doubled, the ITD decreases to 19.9°C, while tripling both reduces it further to 14.7°C. The impact of this variation of heat transfer coefficient and mass flow rate on power production is presented in Figure 5. In the representative air-cooled condenser unit, each degree decrease in ITD relative to baseline conditions increases electricity generation by approximately 0.4%. Doubling the air-side convective heat transfer coefficient increases power production by 3.3%, while tripling the air-side convective heat transfer coefficient only increases power production by 4.3%. Meanwhile, doubling and tripling the air-side mass flow rate result in power production increases of 3.0% and 4.0%, respectively. However, combining increases in both parameters again yields the greatest gains in power

production. Doubling both parameters results in increased power production by 6.7%, while tripling both parameters increases power production by 9.1%. Thus, although reducing air-side thermal resistance or increasing the air-side mass flow rate can increase the electrical generation of the turbine, increases in both parameters are necessary to match the performance of wet-cooled condensers.

Figure 4. Impact of convective heat transfer coefficient and mass flow rate ratio on initial temperature difference.

Figure 5. Impact of convective heat transfer coefficient and mass flow rate ratio on power production.

3.3 Pressure Drop

Although increasing the air-side heat transfer coefficient and mass flow rate result in increased turbine output, this does not always increase the net power plant output due to increased air-side pressure drop, and therefore fan power requirements. Increasing the heat transfer coefficient typically requires the addition of surface enhancements or other features that increase pressure drop. Similarly, increasing the mass flow rate increases the frictional pressure drop in the heat exchanger fins and the minor losses associated with flow through the overall ACC cell. Pressure drop contributions from the fins and minor loss terms for varying air flow rates are presented in Figure 6. At the lowest mass flow rate considered, fin channel-flow resistance pressure drop accounts for 47% (14.2 Pa / 30.1 Pa) of the total pressure drop, but decreases to 18% (174 Pa / 957 Pa) at the highest $\dot{m}_{\text{air}} / \dot{m}_{\text{steam}}$ value considered. Thus, future low-ITD ACC designs may necessitate new housing geometries to reduce flow resistance at high air flow rates. For instance, the fin entrance and exit losses account for a substantial penalty due to the sharp angle of the heat exchanger tubes relative to the fans, but increasing this angle from 30 to 35 degrees is predicted to reduce these minor losses by 36% at the highest mass flow rate. Similarly, enhanced finned tube designs must also be tailored to yield acceptable pressure drops at greater air flow rates.

Figure 6. Contributions to total pressure drop for a range of mass flow rates.

3.4 Net Plant Efficiency

Increasing the cooling air flow rate only raises the net power plant efficiency up to a limit. As shown in Figure 7, the maximum net plant efficiency for the baseline case operating with smooth or louvered fins is reached at $\dot{m}_{\text{air}} / \dot{m}_{\text{steam}} \sim 105$. Although turbine output continues to increase beyond this point, the additional required fan power exceeds those gains. If possible, increasing the heat transfer coefficient by a factor of 2 or 3 without additional pressure drop would increase the net plant efficiency by 2.4% or 3.1%, respectively. However, the peak plant efficiency is still reached at approximately the same mass flow rate. As indicated in Figure 7, even such performance gains are not sufficient to match the performance of a wet cooling tower. To achieve similar plant efficiencies, the ACC pressure drop must also be reduced. For example, consider a case where the air-side pressure drop varies linearly from the baseline value at a $\dot{m}_{\text{air}} / \dot{m}_{\text{steam}} = 92$ to twice that value at $\dot{m}_{\text{air}} / \dot{m}_{\text{steam}} = 350$. If the air-side convective heat transfer

Figure 7. Net efficiency of power plant with dry cooling system for a range of mass flow rates.

coefficient was also tripled from the baseline value, then the peak net power plant efficiency would reach 35.0% at a mass flow rate ratio of 155 (+68% above the baseline value) and air-side pressure drop of 98 Pa (+24%), matching the performance with a wet cooling tower. Thus, to achieve plant performance similar to water-cooled facilities, ACC air-side heat transfer coefficients and flow rates must both be substantially increased with minimal additional fan power requirements.

3.5 Potential to Improve ACC Performance through Heat Transfer Enhancement

Results from the previous section indicate that air-to-steam flow rate ratios must be dramatically increased (+68%), air-side thermal resistance significantly reduced (−66%), and air-side pressure losses maintained close to conventional levels (+24%) in order to match wet-cooled condenser power-plant efficiencies. Such improvements may not be feasible with conventional

plain-finned-tube ACC technologies and package sizes. A relatively mature air-side heat-transfer enhancement approach (multi-louvered fins) was considered in this study, but the high associated pressure drops were found to negate any net plant performance improvements. Thus, it appears that innovative transport enhancement techniques must be developed in order to approach wet-cooled condenser performance levels.

In this section, a number of heat transfer enhancement techniques are presented and assessed for the potential to yield significant ACC performance improvements. This survey is not meant to be comprehensive, but rather, is intended to highlight the breadth of enhancement techniques currently being investigated. The outlined ACC performance objectives are certainly challenging. However, dramatic gains in comparatively mature technologies including gas turbine blade cooling (internal and film cooling (Han *et al.*, 2001)) and electronics thermal management (with modern chip heat fluxes exceeding 100 W cm^{-2} (Anandan and Ramalingam, 2008)) have been achieved through innovative enhancements. With sufficient impetus, similar gains could be achieved in air-cooled condensers.

3.5.1 Conventional enhancement

Multi-louvered, ribbed, and slit-fin transport enhancement techniques are commonly employed in compact finned-tube heat exchangers operating at laminar air-side flow conditions, such as those considered in this study ($Re_{DH} = 1070$ for the baseline flow case). These technologies operate by frequently disrupting and “restarting” the air-side boundary layers, yielding the high heat transfer coefficients characteristic of developing flows (Dewan *et al.*, 2004). While these approaches can significantly reduce air-side thermal resistance (-37% thermal resistance, $+60\%$ h in this study), the resulting pressure drop increase ($+110\%$ in the considered fin channels) was found to negate net plant performance gains. However, the

relatively low additional manufacturing cost compared with plain fins (one stamping operation) should also be considered as an important benchmark when assessing other transport enhancement techniques.

3.5.2 Winglet Vortex Generators

Winglet vortex generators (VGs) represent an alternate air-side enhancement technique that can also be economically produced using a stamping process. In this approach, small tabs or *winglets* extend into the flow and generate longitudinal vortices that promote bulk mixing (Joardar and Jacobi, 2008). When implemented strategically, winglet vortex generator arrays can significantly improve heat transfer with minimal pressure drop penalties. Based on a survey of the literature, heat transfer coefficient enhancements of 20 – 60% can be achieved with only 10 – 20% increases in pressure loss (Fiebig *et al.*, 1993; He *et al.*, 2010; Joardar and Jacobi, 2005, 2008; Russel *et al.*, 1982; Sohal and O'Brien, 2001). In some cases, heat transfer enhancements can be achieved while simultaneously reducing pressure drop (–10 – –15%) (Torii *et al.*, 2002; Wu and Tao, 2007). However, such results have been specifically achieved by configuring winglet VGs to reduce flow separation downstream of round tubes in finned-tube heat exchangers. It is not clear whether similar gains can be achieved in plain channel flow. Further heat-transfer enhancements can be achieved by tuning winglet VG geometries. Gholami *et al.* (2014) found heat transfer improvements (5%) and pressure drop reductions (–15%) by using wavy winglet VGs instead of flat winglets. Overall, winglet VGs are a promising technology for ACC enhancement because they can yield significant heat transfer enhancements at small, or even negative, pressure drop penalties. Winglet VGs alone may not be sufficient to achieve wet-cooled condenser performance levels in ACCs, but could represent an important element of more sophisticated *compound* transport enhancement approaches.

3.5.3 Chaotic Mixing

Passive *chaotic mixing* heat transfer enhancement has received increased attention in recent years. In this approach, the flow geometry is configured so that pathlines of adjacent fluid particles diverge in the flow direction, leading to efficient mixing. Chaotic mixing configurations can lead to significant increases in heat transfer with minimal additional pressure drop penalties compared with similar flow geometries that only produce *regular mixing*. A number of investigators have compared convective heat transfer in conventional helically coiled tubes (regular mixing) with similar geometries in which the coil axes are rotated by 90° after each turn (chaotic mixing) (Acharya and Sen, 1992; Chagny *et al.*, 2000; Mokrani *et al.*, 1997; Peerhossaini *et al.*, 1993). In these studies, chaotic mixing yielded 15 – 30% increases in heat transfer rates with negligible changes in pressure drop. Zheng *et al.* (2013) conducted a simulation-based study of chaotic mixing in small-diameter zig-zag channels, which could conceivably be stamped into corrugated metal fins for ACC applications. Zheng *et al.* (2013) found heat transfer enhancements of $\sim 80\%$ and relatively high pressure drop increases of $\sim 140\%$ compared with equivalent size straight channels. However, overall performance improvements may be competitive with other passive heat-transfer enhancement techniques that do not take advantage of chaotic mixing. Based on these studies, carefully designed flow geometries can passively effect chaotic mixing, and improve heat-transfer performance compared with regular-mixing-based enhancement configurations at minimal additional pressure-drop penalties.

3.5.4 Jet-Impingement Enhancement

Air-jet impingement has been demonstrated to deliver high flux cooling for electronics thermal management applications, and could be a promising heat-transfer enhancement approach for ACCs. Effective convection coefficients can approach values typically associated with liquid cooling ($h \sim 2000 \text{ W m}^{-2} \text{ K}^{-1}$ (Brignoni and Garimella, 1999; El-Sheikh and Garimella, 2000)), compared to $37.4 \text{ W m}^{-2} \text{ K}^{-1}$ in the baseline ACC design. However, fan-power requirements for given flow rates have conventionally been much higher than in plain finned-tube heat exchangers. Based on a survey of representative studies (Brignoni and Garimella, 1999; El-Sheikh and Garimella, 2000; Garimella and Schroeder, 2001; Leland *et al.*, 2002), the most efficient design delivered an average convection coefficient of $\sim 1700 \text{ W m}^{-2} \text{ K}^{-1}$ at a heat-transfer-surface specific flow rate ($U_s = V / A$) of 7.0 m s^{-1} (Brignoni and Garimella, 1999). This air-jet impingement heat sink required a driving pressure of 140 Pa, leading to a surface-specific fan power ($P_s = P / A$) of 1780 W m^{-2} , assuming a fan efficiency of 55%. In comparison, for the baseline ACC design, total fin entrance/exit and channel flow pressure drops of 49 Pa are predicted for a similar value of U_s (7.5 m s^{-1}), yielding a P_s value of 680 W m^{-2} ($\sim 60\%$ less). There could be alternate air-jet configurations that are better suited for ACC applications with relatively high heat-transfer coefficients (e.g., $100 - 200 \text{ W m}^{-2} \text{ K}^{-1}$), but acceptable fan-power requirements.

A major challenge in applying jet-impingement cooling to ACCs would arise in economically routing and pumping air-flow to the distributed heat transfer surfaces over the full condenser. Still, the high jet-impingement cooling performance achieved in electronics thermal management applications provides motivation for future research into large-scale heat-exchanger and ACC implementations.

3.5.5 Mechanical mixing

Hidalgo *et al.* (2010) proposed a convection heat-transfer enhancement approach in which piezo-actuated oscillating reeds are inserted into air-flow channels. The motion of the reeds generates vortical flow structures and increases fluid mixing. The oscillating reeds were experimentally demonstrated to increase downstream heat transfer coefficients by 50 – 120% with 80 – 150% increases in pressure drop. These experimental studies were conducted in high-aspect-ratio (~ 11) channels that are representative of air-coupled heat exchanger geometries ($D_H = 4.6$ mm, similar to $D_H = 4.8$ mm in the baseline design). Direct power consumption by the reeds was small: $\sim 6\%$ of fan power consumption at a surface-area specific air flow rate of 0.48 m s^{-1} . In a follow-on study, Herrault *et al.* (2012) demonstrated a similar configuration in which reed oscillation was fully aeroelastically activated, eliminating the need for direct electrical input and simplifying system integration. Similar heat transfer coefficient enhancements ($\sim 100\%$) were demonstrated in this study with only $\sim 30\%$ increases in pressure drop relative to plain channels. While such mechanical mixing techniques are still in their infancy, these initial investigations have demonstrated substantial heat transfer enhancement with moderate pressure drop penalties in geometries relevant to ACC applications.

3.5.6 Electrohydrodynamic Enhancement

Electrohydrodynamic (EHD) techniques have been proposed as mechanically simple approaches for active convective heat transfer enhancement. In the *ionic wind* configuration, a low-current high-voltage circuit is formed from a wire electrode through the fluid medium to a second flat electrode on a heat-transfer surface. The transport of ionized gas molecules between these electrodes effects net flow through viscous mechanisms, and can distort the boundary layer, increasing heat transfer rates (Laohalertdecha *et al.*, 2007). Go *et al.* (2008) applied this

approach to air flow over a heated plate, and measured an average heat transfer coefficient enhancement of approximately 100% (locally exceeding 200%) with an electrical power consumption of $\sim 100 \text{ W m}^{-2}$. These results are promising, considering the baseline ACC surface-specific fan-power consumption of 680 W m^{-2} . However, Go *et al.* (2008) only considered external flow over a flat plate with unenhanced heat transfer coefficients of $10 - 15 \text{ W m}^{-2} \text{ K}^{-1}$. The potential for gains in higher baseline pressure-drop and heat-transfer-coefficient configurations, such as ACC fin-channel flows, is not yet known.

3.5.7 Acoustic Enhancement

Acoustic methods have been proposed for active convective transport enhancement. These techniques have primarily been applied to liquid and phase-change configurations, but have also been investigated for gas-phase heat transfer. In the “power ultrasound” frequency range (20 – 100 kHz), amplitudes can be sufficient to increase fluid mixing or generate bulk fluid flow through acoustic streaming. Single-phase convective heat transfer coefficient increases of 50 – 300% have been achieved through such approaches (Legay *et al.*, 2011). While ultrasonic techniques may hold promise for future ACC and air-coupled heat exchanger configurations, required power intensities and pressure drop changes are currently not well characterized. Additionally, scale-up to full air-coupled heat exchangers has not yet been investigated.

3.5.8 Summary of Heat Transfer Enhancement Techniques for ACC Applications

A wide variety of heat-transfer enhancement techniques are being developed that could lead to improvements in air-cooled power plant performance. Representative “state of the art” heat-transfer-coefficient and required motive-power increases for these approaches are summarized in Table 2. EHD and acoustic enhancements techniques have not yet been characterized in heat transfer configurations relevant to ACC applications, and are therefore excluded from Table 2. Some enhancement techniques, such as winglet VGs and chaotic mixing, can yield significant increases in convection coefficients (15 – 60%) at low pressure drop increases, but are not sufficient to approach water-cooled power-plant performance levels. Other approaches, such as multi-louvered fins, jet-impingement, and mechanical mixing can yield dramatic reductions in air-side thermal resistance, but at high fan-power requirements. In most of the considered air-side convection configurations, fan power scales as $P \propto V^3$. At present, none of these approaches have been demonstrated to operate near baseline fan-power requirements at the target 68% increase in flow rate. However, with further refinement, some of these emerging technologies could approach such performance criteria. Additionally, there may be the potential to achieve greater gains by integrating multiple approaches and harnessing the advantages of each, a method termed *third-generation enhancement* (Bergles, 2002). For example, perhaps an ACC

Table 2. Representative “state of the art” heat-transfer enhancement performance factors

Approach	Heat Transfer Coefficient Increase	Motive Power Increase
Smooth corrugated fins (baseline)	0%	0%
Multi-louvered fins (conventional enhancement)	60%	110%
Winglet vortex generators	20 – 60%	–15 – 20%
Chaotic mixing	15 – 30%	~5%
Jet impingement	2,000+ %	200+%
Mechanical mixing	100%	30%
Enhancement objective (at +68% flow rate)	200%	24%

design could incorporate boundary layer *restarting* with multi-louvered fins, mechanical bulk-flow mixing, and pressure drop reduction with winglet VGs.

3.6 Ambient Temperature

Even with significantly reduced thermal resistance and increased air flow rates, the performance of ACCs is highly sensitive to the ambient air temperature. As shown in Figure 8, varying the ambient temperature by -10°C to $+10^{\circ}\text{C}$ leads to respective changes in turbine power production of $+4.2\%$ to -4.1% . This trend is approximately linear throughout the considered operating conditions. Similar power production trends are observed with increased air flow rates and heat transfer coefficients, as indicated in Figure 8. Thus, plant performance gains from advanced ACC technology could be offset during periods of elevated ambient temperature, when power demand is greatest. In contrast, the temperatures of large bodies of water sourced for liquid cooled condensers are generally more stable than ambient air temperatures, leading to reduced performance penalties under such conditions. Similarly, the increased moisture capacity of high-temperature air allows evaporatively cooled condensers to operate effectively, even at elevated ambient temperatures.

Figure 8.Impact of ambient temperature on turbine power production.

3.7 Water Savings with Hybrid Systems

To compensate for the reduced power output of dry cooling systems under elevated ambient temperatures, hybrid wet/dry cooling systems can be employed. Figure 9 compares the water savings achieved with two hybrid cooling methods: evaporative spray cooling and parallel wet/dry cooling, under the baseline operating conditions evaluated in the air-cooled condenser model. Evaporative spray cooling reduces the inlet air temperature to the condenser using a liquid water spray. It enables the ACC system to operate at full capacity for short periods with high ambient temperatures, requiring minimal additional capital investment. Although this approach still consumes less water than a standard cooling tower with $<5\text{K}$ inlet air evaporative subcooling, at high ambient temperatures it has very high water consumption ($>5 \text{ m}^3\text{MWh}^{-1}$), and the maximum inlet cooling available is limited by the ambient relative humidity. In addition, unevaporated spray droplets impacting the surface risk causing scaling or corrosion damage to the finned tubes (EPRI, 2005). In the parallel wet/dry cooling paradigm, a wet cooling tower is

Figure 9. Water savings from two hybrid cooling system options.

installed in parallel with the air-cooled condenser. The two condensers can be scaled so that either the wet cooling tower is only operated at peak ambient temperatures, or that both condensers provide a portion of the cooling over a wide range of ambient conditions. The sample parallel system in Figure 9 is designed so that the air-cooled condenser provides 50% of the cooling load under baseline conditions ($T_{\text{amb}} = 30^{\circ}\text{C}$). At low ambient temperatures, the ACC can provide additional cooling, reaching a maximum of 89% of the required cooling at 0°C , but it provides reduced cooling at high ambient temperatures, reaching a minimum of 22% of the total cooling load at 50°C . This parallel cooling system has the advantage of reduced water usage throughout the entire possible ambient temperature spectrum, but requires substantial capital investment.

CONCLUSIONS

A model of a representative air-cooled condenser unit coupled to a baseload steam power plant was developed to examine the performance of air-cooled condensers under varying

operating conditions. It was found that wet-cooling systems generate ~6% more power than the dry ACC system, which was sized to have an ITD of 35°C at an ambient temperature of 30°C. Doubling the air-side convective heat transfer coefficient and mass flow rate could increase power production by 3.3% and 3.0%, respectively, provided that additional required fan power is minimal. It was shown that wet-cooled power-plant efficiency levels could be achieved with enhanced ACCs if air flow rates are significantly increased (+68%), convection resistances significantly reduced (−66%), and pressure losses maintained close to conventional levels (+24%). A variety of emerging convective heat-transfer enhancement approaches were assessed for the potential to achieve these performance levels in equivalent system sizes. At present, there are no individual mature enhancement technologies that have been demonstrated to meet these objectives. However, through further refinement, or possibly by integrating multiple complementary enhancement techniques, such performance improvements may be realized.

ACC power plants were shown to be particularly sensitive to ambient temperatures, and a 10°C increase in T_{amb} was found to **reduce power production by 4.2%**. This sensitivity could be mitigated by using a hybrid wet/dry system, but this would reduce the water savings of the system. A hybrid evaporative spray cooling system that only uses water at high ambient temperatures would consume less water than a standard cooling tower, while still achieving acceptable performance. A parallel wet/dry hybrid system was also investigated. This system was shown to have water usage than a wet cooling tower-only system for the entire range of ambient temperatures considered, with estimated savings of 22 – 89% over ambient temperatures of 0 to 50°C.

This investigation highlighted the potential to meet rapidly growing power demands with reduced water consumption through the use of air-cooled condensers. However, major

engineering challenges must be overcome to achieve similar power plant performances to those obtained with wet cooling technologies, particularly during periods of elevated ambient temperatures. Emerging heat transfer enhancement technologies may lead to significant improvements in dry cooling. Hybrid cooling technologies represent promising alternatives, but fouling concerns for spray cooling and the high capital costs of parallel wet/dry condensers must be taken into consideration.

ACKNOWLEDGEMENTS

The research reported in this work was supported, in part, by the National Science Foundation under grant 1264886. The authors are grateful for this support.

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