A Simple Physical Model for Consumptive Water Use of Thermal Power Plants with Once-Through Cooling

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Abstract

Consumptive use of water in once-through cooling (OTC) of thermal power plants has been correctly attributed to enhanced evaporation from heated discharge water. However, complex models of the thermal plumes are not commonly explicated in full, and the physical concepts that they contain are challenging to comprehend. I present a simple physical model that builds on the conclusion from hydrodynamic models that bed conduction is a small fraction of ultimate heat loss. Simple thermodynamics of the rates of heat loss as latent heat, sensible heat, and thermal radiation then generates simple and robust estimates to the total evaporative loss. The evaporative loss fraction is confirmed as dependent almost solely upon water temperature and not upon relative humidity.

Keywords

once-through cooling
consumptive water use
evaporative fraction
thermal power plants
simple model
relative humidity

1. Introduction

Water use in the US includes major components for cooling at thermal electric power plants. Estimates place this use as 39% to 41% of total freshwater withdrawals [6][11]and approximately 3.3% of total consumptive use [11]. Thermal power plants use three different types of cooling [8] for waste heat rejection: 1) wet towers: closed cycle circulation of water, with evaporation of water in a cooling tower as the main fraction of heat rejection; 2) dry towers: closed cycle circulation of water, with the condenser transferring heat to dry air, incurring no extra water use for cooling; and 3) once-through cooling: copious water is drawn over the condenser, transferring heat to it, with the warmed water being rejected to the same body of water. While it seems intuitive that once-through plants should generate little evaporative loss because the water is returned in the liquid phase, various reports [3][6] put the evaporative loss at 60% or more of the loss from a wet tower of the same cooling capacity. In essence, warming the water increases the evaporation rate from the whole water body to which the flow is returned.

2. A simple, comprehensible model

The great majority of reports or publications offer the claim that once-through cooling (used by 43% of generating capacity: [8] has such significant evaporative losses, but without presenting the physical models to support the claim. It would appear that the modeling is complicated, given fluid mixing in three dimensions and a variety of heat-transfer processes. However, I show here that the physics is rather simple, to a good working approximation. The core of the argument is that the warmed water has only three major ways to lose its heat content – evaporation that carries latent heat, sensible heat flux, and thermal radiation emission; heat conduction into the bed of the water body is considered to be minor when streams are deep (e.g, 12% for a stream as shallow as 3 m in common operating conditions: [2]. If the warmed water is diluted with other water, the ratio of the fluxes from the three loss routes is changed only moderately, because the heat loss rates are relatively close to linear in behavior. Furthermore, one can see that the fraction of heat loss as latent heat is higher in warmer climates, but virtually independent of relative humidity (here, and also Williams and Tomasko, 2009)[11].

Models of the enhanced evaporation from water discharged from once-through cooling systems have been published, such as by Czernuszenko [2], Lowe et al., (2009) [5], and Williams and Tomasko [11]. However, some did not consider radiative cooling, a modest route for heat loss, and results of the last-mentioned study are inconsistent with energy balance. Here I present an alternative, simple physical model that is comprehensible and intrinsically more accurate.

Downstream of a once-through cooling installation, water temperature rises by an increment ΔT(x,y,z) across a thermal plume in three dimensions. Ignoring heat conduction into the bed, the temperature rise is ultimately presented at the surface as an increment δT(x,y). Because the heat loss processes are closely linear functions of temperature within a small range of the unperturbed temperature, I may write the expressions for vertical flux densities of sensible heat, H, latent heat, LE, and thermal infrared emission, R, as

$$H = H_0 + \left(\frac{\partial H}{\partial T}\right)\delta T \tag{1}$$

$$LE = LE_0 + \left(\frac{\partial LE}{\partial T}\right)\delta T \tag{2}$$

$$R = R_0 + \left(\frac{\partial R}{\partial T}\right)\delta T \tag{3}$$

I then write these energy flux densities in terms of the boundary-layer conductance, g_b (in molar units, mol m^{-2} s^{-1}), the molar heat capacity of air, $C_{P,m}$, water surface temperature, T, ambient air temperature, T_a, the saturated partial pressure of water vapor at the water surface, $e_{w,s}$, the partial pressure of water vapor in ambient air, e_a, total air pressure, P_a, the molar heat of vaporization of water, λ, the thermal emissivity of water, ε (about 0.96, angle-averaged), and the Stefan-Boltzmann constant, σ. I obtain

$$H = g_b C_{P,m}(T_w - T_a) \qquad \frac{\partial H}{\partial T} = g_b C_{P,m} \tag{4}$$

$$LE = \lambda g_b \frac{(e_{w,s} - e_a)}{P_a} \qquad \frac{\partial LE}{\partial T} = \lambda g_b \left(\frac{\partial e_{w,s}}{\partial T}\right) / P_a \tag{5}$$

$$R = \epsilon \sigma {T_{w,abs}}^4 \qquad \frac{\partial R}{\partial T} = 4 \epsilon \sigma {T_{w,abs}}^3 \tag{6}$$

where $T_{w,abs}$ is the Kelvin temperature. From the formula for ∂LE/∂T, one can see that the extra evaporation from the water body over the unperturbed case is independent of air humidity; e_a does not enter into the formula. In practice, then, one need know only the environmental variables g_b, T_w, and P_a and the physical constants $C_{P,m}$, λ, ε, and σ. The boundary-layer conductance, g_b, can be calculated from other formulations, such as conductance formulated in velocity units of m s^{-1}, $g_{b,v}$, using the molar density of air, $P_a/(RT_{a,abs})$. Molar units are commonly used in physiology (Ball, 1987) and some micrometeorology. The value of $g_{b,v}$ is generally calculated using a function of windspeed, u, and is equivalent to a knowledge of the wind function, ψ, as used by Williams and Tomasko [11] and in the U. S. Geological Survey Branched Lagrangian model [4], as I show below.

The fraction of total energy input from cooling the condenser that goes into evaporation, or evaporative fraction, f_e, is then

$$f_e = \frac{\left(\frac{\partial LE}{\partial T}\right) \delta T}{\left(\frac{\partial LE}{\partial T}\right) \delta T + \left(\frac{\partial H}{\partial T}\right) \delta T + \left(\frac{\partial R}{\partial T}\right) \delta T} = \frac{\left(\frac{\partial LE}{\partial T}\right)}{\left(\frac{\partial LE}{\partial T}\right) + \left(\frac{\partial H}{\partial T}\right) + \left(\frac{\partial R}{\partial T}\right)} \tag{7}$$

This fraction must be evaluated at each location (x,y). However, for small temperature ranges over the reach of the plume, the function f_e changes only moderately. It may be evaluated at usable accuracy using a reasonable weighted average over the reach. In a linear relaxation process, which the water cooling approximates, the mean temperature offset, δT, at which cooling occurs is just δT/2, as one can show with simple calculus. One then does not need to know the detailed pattern of the temperature offset, δT, in the thermal plume in order to get an acceptable answer. If the small nonlinearities in the fluxes as a function of temperature are included, one can add second derivatives in the formula above, but the effect is very small and the complication is generally not informative.

A quite accurate approximation to the saturated water vapor pressure, $e_{w,s}$, is [7]

$$e_{w,s} = 610.8\ Pa\ exp\left(\frac{17.269\,T}{237.2+T}\right) \qquad \frac{\partial e_{w,s}}{\partial T} = \frac{17.269*237.2}{(237.2+T)^2} e_{w,s} \tag{8}$$

3. Comparison To other estimates

I now proceed to a numerical estimate, using environmental conditions that I infer for the example of Calvert Cliffs power plant on Chesapeake Bay from Williams and Tomasko [11]. They computed the evaporation rate, E, which is related to LE as LE = λE. They also used the wind function, ψ, which is simply $E/(e_{w,s}-e_a)$, with E in the units of cm d^{-1}, equivalent of 0.0064 mol m^{-2} s^{-1}. Using some algebra, one can show that g_b = 0.0064 mol m^{-2} s^{-1} * P_a (kPa). Assuming P_a is close to 100 kPa at sea level, one gets g_b = 0.64 ψ in their conditions. The formula, ψ = 0.301 + 0.113 V, gives ψ = 0.866 in its native units and g_b = 0.554 mol m^{-2} s^{-1}, with a windspeed of 5 m s^{-1}. The authors did not specify a water temperature, which must vary seasonally, but the results are not highly insensitive to this variable over the plausible range of water temperatures. Using an unperturbed (inlet) water temperature, $T_w{}^0$= 15°C as an annual average[9], and a temperature rise of 5.6°C, I use the average T_w as 17.8°C, thereby obtaining these results:

$$e_{w,s} = 2039\ Pa \qquad \frac{\partial e_{w,s}}{\partial T} = 128\ Pa\ K^{-1} \qquad \frac{\partial LE}{\partial T} = 32.0\ W\ m^{-2}\ s^{-1} \tag{9}$$

$$\frac{\partial H}{\partial T} = 16.0\ W\ m^{-2}\ s^{-1} \tag{10}$$

$$\frac{\partial R}{\partial T} = 5.4\ W\ m^{-2}\ s^{-1} \tag{11}$$

$$f_e = \frac{32.0}{32.0 + 16.0 + 5.4} = 0.60 \tag{12}$$

If the inlet temperature is higher, such as the 25°C in Chesapeake Bay in the summer, the mean plume temperature is T_w = 27.8°C, the corresponding results are

$$e_{w,s} = 3738\ Pa \qquad \frac{\partial e_{w,s}}{\partial T} = 218\ Pa\ K^{-1} \qquad \frac{\partial LE}{\partial T} = 54.4\ W\ m^{-2}\ s^{-1} \tag{13}$$

$$\frac{\partial H}{\partial T} = 16.0\ W\ m^{-2}\ s^{-1} \tag{14}$$

(unchanged)

$$\frac{\partial R}{\partial T} = 5.9\ W\ m^{-2}\ s^{-1} \tag{15}$$

$$f_e = \frac{54.4}{54.4 + 16.0 + 5.9} = 0.71 \tag{16}$$

The evaporative fraction rises significantly with temperature, because the latent heat flux rises 6% to 7% per degree Celsius, while sensible heat losses are unchanged and radiative losses are small overall. The general pattern is show in Fig. 1.

The values for f_e can be converted to mass of water evaporated, knowing the total thermal input. Williams and Tomasko [11]quote a flow rate of 9500 $m^3\ min^{-1}$ or 158.3 $m^3 s^{-1}$. With a temperature rise of 5.6°C and the heat capacity of water as $4.2\text{x}10^6$ J m^{-3} K^{-1}, I calculate the thermal input as 3.72 GW. If a fraction f_e = 0.60 of this heat goes into evaporation, the loss as vapor can be calculated by dividing the total latent heat loss by the heat of vaporization of water, $2.5\text{x}10^9$ J m^{-3} (at 0°C). The result is 0.90 $m^3\ s^{-1}$, less than 1% of inflow and consistent with the estimate of 1% or less by the Electric Power Research Institute [3]. This rate is readily converted to common power engineering units of water use per unit electrical energy, as gal/MWh. One must back-calculate the electrical energy, approximately ½ the rejected heat, or find the operating specifications. The operators, CENG, report each of the two units at

Calvert Cliffs as having 875 MWe net output (http://www.cengllc.com/calvert-cliffs-nuclear-power-plant/). The consumptive water use is then

$$CWU = \left(\frac{waste\ heat\ rate}{electrical\ power}\right) f_e \left(\frac{vol. water\ evaporated}{unit\ heat\ dissipated}\right) \qquad (17)$$

$$= \left(\frac{3.72\ GW}{1.75GW}\right) 0.60 \left(\frac{1\ m^3\ evap.}{2.5x10^9\ J}\right)\left(\frac{3.6x10^9 J}{MWh}\right)\left(\frac{1\ gal}{3.78x10^{-3} m^3}\right) \qquad (18)$$

$$= 485\ gal/MWh \qquad (19)$$

This moderately exceeds the published range of estimates for nuclear power plants (400 gal/MWh: [3][6]. Those estimates, too, are derived from models, because it is impractical to measure differences in stream flows upstream and downstream to the required accuracy. The calculation may be compared with that presented by Williams and Tomasko [11]. They estimate a plume area of 1000 m by 300 m, for an area of $3x10^5$ m^2. Their estimate of enhanced evaporation over the plume area is 0.9 cm d^{-1}, for a total evaporation rate of 2700 m^3 d^{-1} or only 0.0312 m^3 s^{-1}, far different from my calculation. Their corresponding evaporation rate is then only 0.02% of inflow, quite unrealistic.

The same authors also did a calculation for the Dickerson Steam Electric Station on the Potomac River, with a reported much smaller once-through flow of 800 m^3 min^{-1} = 13.3 m^3 s^{-1}, a temperature rise δT = 10°C, and a much larger plume, 10,000 m by 300 m. The thermal input to the river is calculated as 560 MW. My calculations of enhanced evaporation at a mean T_w = 15 + 0.5*10 = 20°C give $e_{w,s}$ = 2339 Pa, $\partial e_{w,s}$ = 145 Pa K^{-1}, $\partial LE/\partial T$ = 36.1 W m^{-2} K^{-1}, $\partial H/\partial T$ = 16.0 W m^{-2} K^{-1}, $\partial R/\partial T$ = 5.5 W m^{-2} K^{-1}, and, finally, f_e = 0.63. This converts to a latent heat loss rate of 350 MW. The other authors obtained an enhanced evaporation rate of 1.65 cm d^{-1} over the plume area, or 28,900 m^3 d^{-1} = 0.335 m^3 s^{-1}. It is equivalent to a latent heat loss rate of 838 MW, which exceeds the thermal input and is in error.

4. Water consumption contrasts with wet towers

It is of interest to compare the evaporative fraction for once-through cooling with that for wet cooling towers. A typical tower [1] operates with a nominal inlet air temperature of 68°F = 20°C at 50% relative humidity ($e_{a,in}$ = 1170 Pa). The outlet air is at 88°F = 31.1°C at 98% relative humidity ($e_{a,out}$ = 4433 Pa). At a given flow rate, V, of inlet air, the sensible heat flow, H, is V *322 J mol^{-1} (dry-air part only; extra H is carried by water vapor, giving about 340 in the same units). The latent heat flow is the flow rate multiplied by the extra mole fraction of water vapor in air, multiplied by the molar heat of vaporization of water. At a total air pressure of 100 kPa, this is V * (4433-1170)/10^5 * $4.5x10^4$ J mol^{-1}, or V * 1546 J mol^{-1}. The evaporative fraction is then about 1546/(1546 + 340) = 0.82. This exceeds the fraction for once-through cooling, but only moderately.

The two examples of once-through cooling given here were in the humid Eastern US. In the arid or semi-arid Western US, base evaporation rates from rivers (or reservoirs) are higher than in the humid Eastern US, primarily from lower relative humidity. However, the evaporative fraction of heat rejection from thermal power plants with once-through cooling does not depend on relative humidity, as noted above. For similar windspeeds, the only significant controlling variable is water temperature, which is quite variable by location in the Western US. Note that very few Western power plants use once-through cooling [10].

5. Conclusions

I conclude that the simple estimates of enhanced evaporation by the method presented here may be useful in understanding the consumptive water use (CWU) by thermal power plants, without complex models. The method could be applied straightforwardly to calculating evaporative losses from water surfaces whose spatial pattern of temperature is resolved, e.g., by thermal infrared imaging [5]. The robust conclusion that relative humidity has little effect on CWU of once-through cooling offers support for more detailed predictions of shutdown of such plants in humid regions during periods of low river flows [12]. My results contradict the conclusion of Macknick et al. [6] that CWU of wet towers is more than double that of once-through cooling; the typical evaporative fractions for the two technologies may differ by factors of only 1.2 to 1.4. The results are closer to conclusions of the study by the Electric Power Research Institute [3], presenting factors that range from 1.0 to 1.8.

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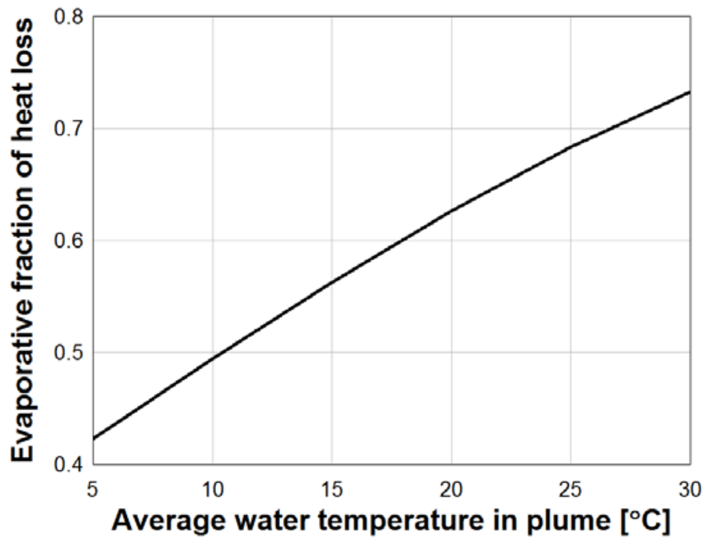

Fig. 1. Fraction of heat discharged in once-through cooling that is ultimately lost as latent heat of evaporation, as a function of mean water temperature of the discharge plume.

Nomenclature

C_p	Molar heat capacity of air at constant pressure (J mol^{-1} K^{-1})
CWU	Consumptive water use rate (gal MWh^{-1} or equivalent)
e_a	Partial pressure of water vapor in ambient air (Pa)
$e_{w,s}$	Partial pressure of water vapor at water temperature (Pa)
f_e	Fraction of added energy going to evaporation (dimensionless)
g_b	Boundary layer conductance of air, molar units (mol m^{-2} s^{-1})
H	Sensible heat flux density from surface(W m^{-2})
LE	Latent heat flux density from surface (W m^{-2})
P_a	Total air pressure (Pa)
R	Radiative heat flux density (outgoing) from surface (W m^{-2})
V	Rate of air flow at wet tower inlet (units cancel in calculations)
δT	Incremental rise in water temperature at any location (K or °C)
ε	Thermal emissivity of a water surface (dimensionless)
λ	Molar heat of vaporization of water (J mol^{-1})
σ	Stefan-Boltzmann constant (5.67×10^{-8} W m^{-2} K^{-4})

Highlights

I examine consumptive water use in once-through cooling of a thermal power plant
I offer a concise model for enhanced evaporation from the water body surface
Heat transfer to the bed of the water body can be neglected, to first order
Evaporative fraction is comparable to that in wet towers but independent of humidity
Consumptive water use is in the range of complex models that can obscure errors