Hybrid Cooling Towers Water Savings Calculations and Measurements

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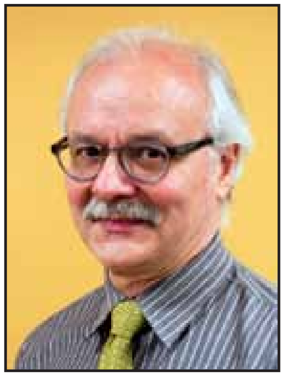

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Abstract

In many areas of the globe, water has become a critical natural resource. To address this, a variety of non-conventional cooling towers have been developed to consume less water, while still meeting the cooling capacity of a traditional evaporative tower. The variety in the design and operation of these hybrid cooling towers results in a wide spectrum of water savings potential, which is further amplified when considering the climate in which the hybrid cooling tower will be installed. Currently, there is no generally-accepted method for estimating the expected water savings from a specific hybrid design at a specific site; nor is there a generally-accepted method for verifying that the estimated water savings was realized. This paper proposes methods for both standardization of hybrid water savings calculations and verification of water savings by field measurements.

Introduction

It has become apparent that water, like energy, is a critical natural resource, not only in the arid southwest of the US, but throughout the world. The electric power generation industry is one of the largest users of water, with the main water consumption occurring in cooling. In the industry, both once-through and closed-cycle cooling (cooling towers) are used. Once-through cooling involves withdrawing cool water from a source – typically a river – and returning the same quantity of warm water back. It is generally accepted that very little water is consumed in this cooling process, as the only loss is from a slight increase in evaporation rate from the warmer discharged water. However, the thermal pollution of the source water can be an environmental issue. In addition, the newest regulations of the Clean Water Act §316 (b) imposes requirements on the water intake to minimize the impact to aquatic fish and shellfish, as well as mammals such as manatees. Cooling towers draw significantly less water into the facility than a once-through system but also 'consume' more water through evaporation.

There is often a tradeoff between using water to save energy and using energy to save water. Nowhere is that interdependence seen more clearly than in electric energy generation using steam-powered turbines, where the efficiency of the Rankine-cycle is dependent on the final condensing temperature of the steam. As the steam condensing temperature rises, so does the back-pressure to the turbine, which lowers the turbine efficiency. For years, water based cooling that use either once-through cooling or closed-cycle cooling was the only choice for a power plant. In the late 1960's, the first Air-Cooled Condenser (ACC) was installed in North America. Since then, ACC installations have increased because they almost completely eliminate water usage by the generating plant. However, ACCs can only cool using the dry bulb temperature. This limitation forces ACCs to be larger, more expensive, and provide lower overall plant efficiencies than equivalent wet cooling towers.

As a result, the cooling industry has responded with many new and innovative hybrid cooling tower designs that attempt to strike a balance between water and energy use. However, because these technologies are so different, they have been difficult to reliably compare.

Hybrid Cooling Systems

The cooling industry has developed a variety of hybrid cooling systems that conserve water. These hybrids combine dry cooling with evaporative cooling, providing less water usage than a conventional wet cooling tower, without sacrificing Rankine-cycle steam condensing temperature. A synopsis of current hybrid cooling tower technologies is shown below. For more information on these technologies, please reference TP14-19 [LI1].

Parallel Condensers – This design consists of a smaller air-cooled condenser connected in parallel to a water-cooled condenser and a cooling tower. The system is self-balancing as the steam will travel to the lowest pressure unit. The wet cooling tower is typically sized to handle the full design load for when the ambient dry bulb is high.

Parallel Path Wet-Dry – This design is considered primarily for plume abatement. Finned coils, located in the plenum walls, are fed a portion of the hot circulating water. Ambient air is drawn through these coils and sensibly heated. This warm, dry air is then mixed in the plenum with the humid air from the evaporative heat exchanger, which reduces the cooling tower's exhaust relative humidity and plume. Since some cooling is done by the coils, some water will be saved; however, the evaporative section must be large enough to handle the entire cooling load. This system is generally much taller than a conventional cooling tower.

Series Path Wet-Dry – This design is considered for higher water savings. Finned coils, located across the plenum, above the fill and nozzles and below the fan, are fed all of the hot circulating water, which is then sprayed over the evaporative heat exchanger. Ambient air first passes through the evaporative heat exchanger, then is heated sensibly through the finned coils, which minimizes plume. Since cooling is done by the coils, water will be saved. An added advantage in hot, dry climates, is that the dry bulb temperature of the evaporative exhaust may be lower than the ambient dry bulb, which allows for additional dry cooling, thus more water savings.

Side-Stream Air Cooling – This is another form of parallel path wet-dry technology. A side-stream dry cooler is fed a portion of the hot circulating water, which is then mixed and sent to an evaporative tower. Since some cooling is done by the coils, some water will be saved. The evaporative tower needs to be full sized, and there is no plume abatement.

Air-to-Air in Plenum – This design places an air-to-air heat exchanger above the wet cooling tower. The exhaust air from the cooling tower passes on one side of the heat exchanger while ambient air passes on the other. In some conditions, the cool ambient air condenses a portion of the evaporative exhaust air, returning the condensate to the basin. Since no cooling of the hot circulating water is done by the air-to-air heat exchanger, the wet tower must be full sized. This design is usually much taller than a conventional wet tower.

Wet/Wet-Dry – This design places sparsely finned coils side by side with a traditional evaporative section in the same airstream. On hot days, the coils are used as splash fill (no fluid in the coils), with a portion of the circulating water flow sprayed over them. In cool weather, however, the circulating water flow can be redirected to flow through the coils, allowing for dry cooling. The dry exhaust would then mix with the evaporative exhaust in the plenum, providing plume abatement. Since a portion of the cooling is done by the coils, some water will be saved. Since the dry coils are on the same level as the evaporative section, this design is much shorter than other hybrid designs, and because the dry coils also double as splash fill, the footprint is only slightly larger than a conventional cooling tower.

Each of these hybrid cooling tower designs have different water savings potentials, costs, parasitic energy demands, reliability considerations, and space issues. A potential purchaser must evaluate using their site requirements and experience. And if that is not complicated enough, there is no accepted methodology for calculating the key requirement, how much water will be saved. This leaves each equipment supplier providing their own assumptions and calculations, and a potential purchaser buried in unverifiable information. The result is that the estimated water savings between two differing technologies is not comparable.

This leaves two quandaries:

1. How can a potential purchaser evaluate the water savings of different technologies when there is no industry-accepted method for doing so?
2. How can a potential purchaser verify that the cooling equipment supplied meets the advertised water savings when there is no industry-accepted method for doing so?

This paper proposes a method for standardization of hybrid water savings calculations and an additional method for verification of the water savings by field measurements. This insures that all cooling equipment and technologies can be assessed on a level field.

General Approach

The authors' proposed approach to estimating water savings is to compare the water use of a hybrid cooling tower to that of a calculated benchmark cooling tower. The calculated water used by this benchmark will be the basis of comparison for all hybrid systems from all potential equipment providers for a given location and thermal load.

Typically, wet cooling towers for power plants operate under almost a constant load and circulating water flow, which means that as the ambient conditions change throughout the year, the cold and hot water fluctuate together to maintain a constant cooling range across the cooling tower. If energy savings is important, cooling towers may be operated at a set cold water temperature while fans are modulated to minimize power usage. Similarly, hybrid cooling towers typically operate at a constant cold water temperature set point in order to take maximum advantage of water saving modes.

For the sake of commonality and simplicity, this paper proposes that the hybrid mode of operation be used in all evaporation calculations. (Constant circulating water flow, and constant hot and cold water temperatures.) It will be shown later in this paper that this operating mode provides an accurate estimate of annual water usage even though the operating mode may differ from actual.

In summary, the proposed benchmark water consumption method includes these assumptions:

1. Constant heat loading all year long
2. Constant hot water and cold water temperatures
3. Site specific set of climate data (Typically ambient wet bulb, dry bulb and barometric pressures that reflect average annual weather conditions.)

With these assumptions, the annual water usage for a benchmark wet cooling tower can be calculated using a specific methodology, and should accurately reflect the loading and climate conditions at a specific location.

The hybrid equipment supplier will use the same assumptions outlined above to calculate any proposed hybrid technology's water usage over a year. Equation (1) shows that the difference between the hybrid and benchmark water usages, divided by the benchmark water usage, equals the estimated hybrid water savings as a percentage.

$$Percent\ water\ savings = P_{ws} = \frac{(Q_{\mathrm{EBe}} - Q_{EHy})}{Q_{\mathrm{EBe}}} \cdot 100\ [\%] \qquad \text{Eq. (1)}$$

$Q_{\mathrm{EBe}} = Volume\ flow\ of\ evaporation\ (Benchmark)\ (l/s)$

$Q_{\mathrm{EHy}} = Volume\ flow\ of\ evaporation\ (Hybrid)\ (l/s)$

Note: blowdown is not included in the calculation. While all evaporative sites will require blowdown, the blowdown is assumed to be a fixed percentage of the make-up water for both the benchmark wet tower and the hybrid and thus does not affect the comparison.

The final, essential step is to validate the field performance of the hybrid cooling tower after commissioning. Unlike plume evaluation and thermal performance, there is no accepted methodology for measuring water consumption under specific conditions. A simple method of evaluating water consumption will be proposed, along with a more complex and more accurate method. The intent is that all parties involved in a hybrid cooling tower installation would agree to one of the two methods to verify water savings.

Benchmark Water Usage for a Wet cooling Tower

To create the benchmark water consumption of an evaporative cooling tower, an equation can be derived using a generalized equation of heat balance between air and water, and then manipulated to provide the exhaust air properties of enthalpy and humidity ratio. How this equation is derived and used is shown in Appendix A, but the main formula is Equation (2).

$$h_{ae} = h_{ai} + c_{pw}T_{cw} \cdot (w_e - w_i) + c_{pw}\Delta T_w \cdot \frac{L}{G} \qquad \text{Eq. (2)}$$

$h_{ae} = Enthalpy\ of\ exhaust\ air,\ \ h_{ai} = Enthalpy\ of\ inlet\ air\ (J/kg)$

$c_{pw} = Specific\ heat\ of\ water\ (At\ average\ water\ temperature)\ (J/kg \cdot {}^\circ C)$

$T_{cw} = Temperature\ of\ cold\ water,\ \ \Delta T_w = Change\ in\ water\ temperature({}^\circ\ C)$

$w_e = Humidity\ ratio\ of\ exhaust\ air,\ \ w_i = Humidity\ ratio\ of\ inlet\ air$

$L = Mass\ flow\ of\ hot\ water,\ \ G = Mass\ flow\ of\ dry\ air\ (kg/s)$

From Equation (2), the known variables are enthalpy and humidity ratio of the inlet air (from climate data), the water temperatures (assumption that hot and cold water temperatures are set as constant), the mass flow of hot water (assumption of fixed 100% water flow) and the specific heat of water. Only the mass flow of dry air and the exhaust air conditions are still unknowns. The following assumptions can be made to reach a unique solution. Note: The effects and practicality of the final two assumptions will be addressed in Section 3.

1. Set the L/G ratio equal to a constant value of 1.0
2. Assume the exhaust air is 100% saturated

Knowing the exhaust air conditions will enable the evaporation rate to be calculated. Doing so for a set of climate data will yield an annualized water consumption that can be used as a benchmark for evaporative cooling towers. Appendix A explains in detail how this can be done.

Hybrid Cooling Tower Water Usage

Because of the numerous hybrid cooling tower offerings noted in Section 1.1, and the many variations of each, it is unlikely a standard evaluation can be created that works for all hybrid technologies. With that in mind, hybrid cooling tower water consumption calculations will be provided by the equipment supplier using their own (and often proprietary) rating methods. Because proprietary calculations are by nature obscured, the following provisions are proposed in order to ensure water consumption calculations are estimated on an equal basis.

1. The equipment supplier must estimate water consumption using the same climate data and assumptions (100% loading, constant hot and cold water temperature) that is used to calculate the wet benchmark. *Note: This is critical. The same unit evaluated using different climate data can predict vastly different water savings.*
2. The equipment supplier must provide evaporation curves that can be used to verify their hybrid cooling tower water consumption predictions.

The evaporation curves should show the evaporation rate vs. entering air wet bulb temperature for various entering air relative humidities. Since the wet benchmark cooling tower assumes constant load and constant range, evaporation curves that provide a direct comparison must (at minimum) depict the full design water flow and full design range at the appropriate altitude (or barometric pressure). An example of appropriate evaporation curves can be seen in Figure (1).

It is important that enough information be presented in the curves to evaluate every point in the appropriate climate data set. The annualized hybrid cooling tower water consumption can then be approximated by extracting the evaporation rate for each climate data point and then correcting for the number of hours or months used in the data set.

While a single evaporation graph that shows 100% water flow and cooling range is sufficient for verifying water savings compared to the benchmark, it is recommended that the equipment supplier provides more information in order to verify actual operating water consumption. The actual water consumption can be checked against the predictions using one of two proposed field tests outlined in Section 4.

For use with the field tests, it is recommended that the equipment supplier provide a set of at least nine evaporation curves, one for each of the combinations of 80%, 100%, and 120% design ranges, and 90%, 100%, and 110% design water flow (Similar to the plume characteristic performance curves from ATC-150). Since these curves are designed to estimate water consumption for actual operating conditions, it is recommended that they be calculated using full fan speed and letting the cold water temperature vary.

Note: It is possible that hybrid cooling towers can have different modes of operation, each designed to conserve more water or energy. For these cases, it is possible that evaporation curves would need to be generated for each mode of operation to verify all operating conditions.

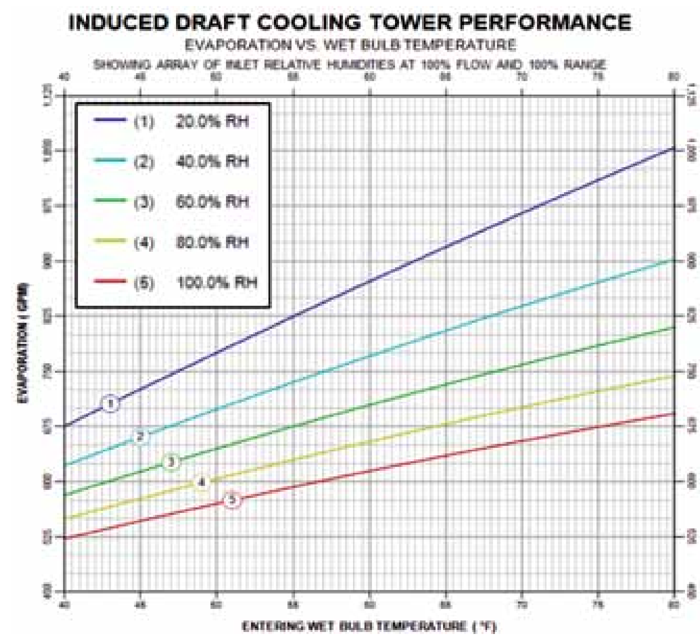

Figure 1: Example of a generic cooling tower's evaporation at 100% water flow and cooling range. This example is based on a hot water temperature of 37.77 °C (100°F), a cold water temperature of 29.44 °C (85°F), and a volumetric water flow of 3,785.41 l/s (60,000 GPM). Sea Level.

Sensitivity Studies

In order to validate the assumptions made to simplify and standardize the evaporation calculations, studies were done to determine their effect on the resulting evaporation estimates.

Sensitivity of Annual Water Use to 'L/G Ratio'

A significant assumption was made by setting the L/G ratio to a value of 1.0. This value was selected to simplify the benchmark equations, as well as to represent a reasonable average value of existing wet cooling tower applications. When evaluating the effect of L/G on evaporation, it is important to remember that the load requirements of the condenser are specified for a specific water flow rate, which means changes to the L/G are really just changes in air flow rate.

While heat and mass transfer processes in a cooling tower are complex, a psychrometric chart can be used to model the air conditions within a cooling tower. The air can also be analyzed by separating the heat and mass transfer into two stages.

In the first stage, the inlet air is adiabatically saturated by following the wet bulb line to the saturation curve. This is illustrated by Curve 1 in Figure (2). Since an adiabatic path is assumed, no enthalpy is transferred from the water.

The second step involves the saturated air moving up the saturation curve (Curve 2 in Figure (2)). This is where enthalpy is transferred directly from the hot water to the air. The air will follow the saturation curve until the enthalpy lost from the water equals the enthalpy added to the air.

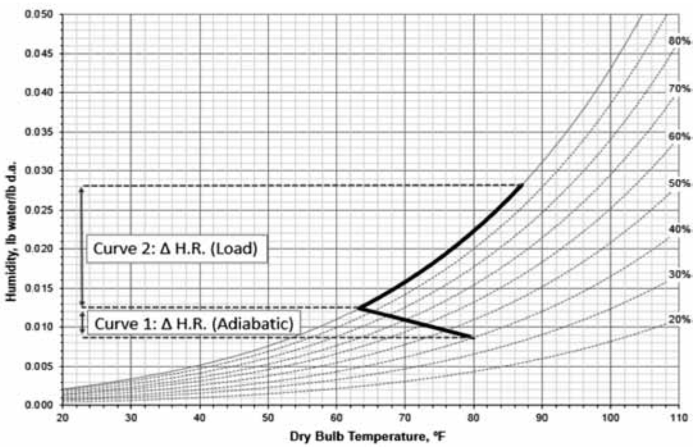

Figure 2: Water usage in a generic wet cooling tower.

Since the enthalpy and humidity ratios are given per mass of air, the lower the ratio of air to water, (high L/G), the less water is needed to adiabatically saturate the air (Curve 1 in Figure (2)). However, lowering the ratio of air to water will also raise the exhaust air temperatures, forcing the exhaust air higher up the saturation curve (Curve 2 in Figure (2)).

When the air is cooler, toward the bottom-left of Figure 2, the saturation curve is flatter, which means more energy is transferred to the air via sensible cooling, heating the air, rather than using evaporation. For warmer temperatures, located in the upper right of Figure 2, the saturation curve is much steeper, which means less energy is transferred to sensibly heat the air than is transferred latently.

Knowing that the total amount of water evaporated in a cooling tower is equal to the mass flow of dry air multiplied by the difference in humidity ratio of the inlet and exhaust air (Equation A.3), it is clear that if the tower is operating where the saturation curve is steeper, more evaporation is expected. So the effect of the L/G is then dependent on where on the saturation curve the cooling tower is operating.

Figure (3) displays the effect of various L/G ratios on the total annual evaporation in four different climates. The results are normalized to the evaporation calculated for an L/G of 1.0 for each location.

At very low ratios, the amount of water required to adiabatically saturate the air dominates, and total calculated water usage artificially goes up due to the saturation assumption. At mid to high L/G ratios, the adiabatic cooling becomes less important and the effect of L/G on evaporation becomes minor. Although the outliers of Alaska (Arctic) and Arizona (Arid) show significant dependence on L/G, moderate climates begin leveling off at around an L/G value of 1.0. While not perfect, a value of 1.0 seems to provide a reasonable average benchmark.

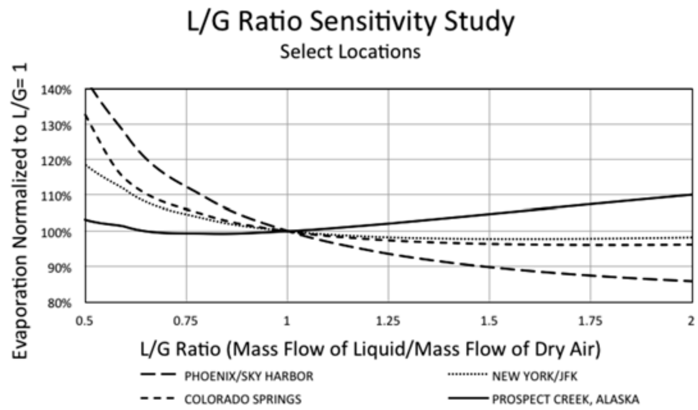

Figure 3: Annual Water Usage as a Function of L/G. Design conditions are the same as in Figure (1).

Note: It is worth mentioning that assuming a constant L/G with constant hot and cold water temperatures is not entirely accurate. As the wet bulb decreases from design and the water flow remains the same, an evaporative cooling tower will naturally provide colder water. In order to keep the cold water at the desired temperature, the fans would need to slow down to draw less air to reduce the amount of cooling. With the mass flow of air decreasing, but the mass flow of water remaining constant, the L/G ratio increases. However, as Figure (3) suggests, increasing L/G past the value of 1.0 does not have a significant impact on the evaporation calculations of most climates.

Sensitivity of Annual Water Use to 'Exhaust Air Saturation Assumption'

Another simplifying assumption in the benchmark water consumption calculations is that the exhaust air is fully saturated. For many inlet air conditions, that is a valid assumption, but when the inlet air is very dry (low relative humidity), this assumption may pose issues. Low L/G ratios (high air flows for a fixed water flow) will also tend to yield unsaturated air.

Using a proprietary program, exhaust air saturation was calculated for each of the weather data sets used to generate Figure (3). As can be seen in Figure (4), when the L/G value is lower, and the climate is more hot and arid, the evaporation can differ significantly if saturation is assumed.

Evap. (Millions of Gal)

	L/G = 0.5	L/G = 1	L/G = 1.5	L/G = 2
PHOENIX (SAT)	691.1	456.1	410.0	391.4
PHOENIX (NS)	403.1	401.4	397.9	391.2
NEW YORK (SAT)	410.9	331.6	324.4	325.7
NEW YORK (NS)	345.5	327.2	324.3	325.7
COLORADO (SAT)	446.5	345.4	333.2	332.4
COLORADO (NS)	352.4	333.7	331.1	332.3
ALASKA (SAT)	262.1	244.3	256.0	269.1
ALASKA (NS)	232.5	241.3	255.8	269.1

Figure 4: Annual water evaporation as a function of L/G, for saturated (SAT), and non-saturated (NS) exhaust air. Design conditions are the same as in Figure (1).

In moderate climates, the difference in annual water consumption with an L/G of 1.0 is less than 5%; however, in hot, dry climates there is a greater error. A possible further refinement of the benchmark method would be to assume 100% saturation if the inlet relative humidity is greater than 50%, and a common lower exhaust relative humidity otherwise. As long as each of the equipment suppliers are held to the same methodology, the final comparisons will be valid.

Sensitivity of Annual Water Use to 'Climate Data set'

Both hybrid and traditional cooling towers operate differently in varying climates. This cannot be clearer than in Figure (4). At an L/G of 2.0, a cooling tower located in Phoenix evaporates nearly 50% more water than the same cooling tower located in Alaska. This only highlights the importance of making sure all equipment is evaluated on the same climate data basis.

It is also important to realize that not all climate data sets are given in the same format. The most detailed data set may provide the average hourly climate on a monthly basis (24 hours x 12 months = 288 data points). Others reduce the number of data points by different averaging methods. Figure (5) shows how these averages trend over the course of a year.

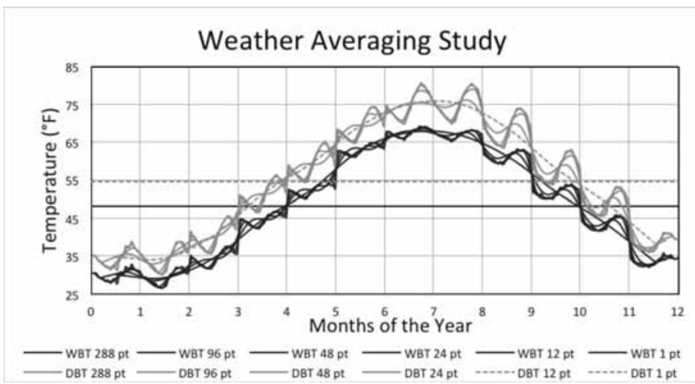

Figure 5: Weather data for New York/JFK, showing the different averaging cases.

Figure (6) shows the sensitivity of the water consumption calculations on the fineness of the dataset. The benchmark evaporative cooling tower evaporation was calculated using each of the datasets from Figure (5).

Figure (6) shows that there is very little error introduced in the water consumption calculations by using a simplified climate dataset as long as there are at least 12 data-points (monthly averages).

	Evaporation (Millions of Gallons)	% Deviation (Baseline 288 pt)
288 Point	331.57	–
96 Point	331.57	0.00%
48 Point	331.57	0.00%
24 Point	331.59	0.01%
12 Point	331.59	0.01%
1 Point	333.12	0.47%

Figure 6: Calculated evaporation, using climate data that is averaged to varying degrees. Design conditions are the same as in Figure (1). L/G assumed to be 1.0.

Note: This is different from estimating the parasitic fan power used by a wet cooling tower. Because of the cube of power requirement for air flow, use of coarse climate data will underestimate the parasitic power needed by an evaporative cooling system. If comparison of both energy and water usage is desired between a hybrid and wet tower, the finest data set available should be used.

Also Note: Hybrid systems may require more detailed data sets in order to take advantage of lower nightly or winter temperatures where dry modes can be implemented more frequently to increase water savings.

Field Testing

Two different methods are suggested for testing evaporation in the field. Choosing the appropriate method depends on many factors such as location, basin setup and whether or not the system is once through or closed-cycle. The best method can be determined from the procedures below.

Short Method

The first proposed test method will run in conjunction with an ATC-105 acceptance test, keeping the same protocol on allowable variances of design conditions and parameters of operation. However, with this evaporation test, the balance of water flow and air flow between tower cells is critical. The water flow to each cell must be within ± 5% of the average water flow per cell. The measured fan power must also be within ± 5% of the average fan power for each cell. A balanced tower will allow each cell to be considered identical for the evaporation measurement. During testing, any additional water source interacting with the circulating water loop (make-up water, other water usages, etc.) must be shut off. Since the blowdown will be turned off for the duration of the test, the tower should be pre-bled prior to the test to prevent over-cycling. Check with your water treatment professional on how this should be accomplished.

The tester must have accurate drawings of the cooling tower basin showing any obstructions such as piers, columns or screens present in the basin. The drawing should depict any slopes in the basin walls or bottom. This will allow the tester to choose whether this method is valid, and then select an ideal location to take measurements.

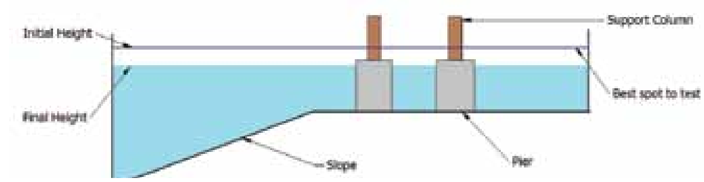

Figure 7: This shows some of the complication with measurements.

Note: When the water level lowers, the obstruction area can change. This can happen if the water level starts above the top of the piers and dips below, or if the water level drops to expose a sloped region during the test. This method is not appropriate for these cases, due to the complexity of correcting for these area changes.

While running near design conditions, an initial and final measurement should be taken over a fixed amount of time. The height differential of the water level, along with area calculated from the basin drawings, will give a change in volume over time. This is the volumetric flow of evaporation.

The length of the test should be sufficient for a large enough change in basin water level. The uncertainty of the test decreases as both the length of the test and water height differential increase. To demonstrate, a test with 9.2 inch basin water level change over 60 minutes has the following uncertainty: Assuming the time is correct but there is a quarter inch margin of error in the basin water level measurement, the final evaporation measurement would have a 4% uncertainty. This example is shown in Example B.2 in the Appendix B.

The short method will not work for all cooling tower layouts. It cannot be applied to once-through cooling towers as there is no standing basin water level. There should be as-built drawings of the basin. Rough measurements of the tower size could introduce error into the final measurement. Depending on the complexity of the basin, the short method may not be accurate enough for evaporation guarantees.

Long Method

For cooling towers unable to take advantage of the short method, there is a longer, more complex, method of evaporation testing. The long method is intended to be run during an ATC-150 plume abatement test. Similar to the short method, the balance of water flow and air flow to the cooling tower cells is critical for extrapolating evaporation to the entire tower. Each cell should be within ± 5 % of the average water flow and fan power measurements. This method allows for normal operation of the tower. Outside water sources may continue to run with the circulating water loop.

The long method uses psychrometric calculations of inlet and exhaust air streams to find a calculated evaporation flow after all the measurements have been performed on a single cell (or two cells) of the cooling tower. Following ATC-150, measurements are taken in the fan exhaust plane with multiple instruments along with the inlet air measurements. It is required to have a minimum of 20 different points of measurements following two perpendicular diameters forming equal annular area segments. Figure (8) demonstrates the instrument layout.

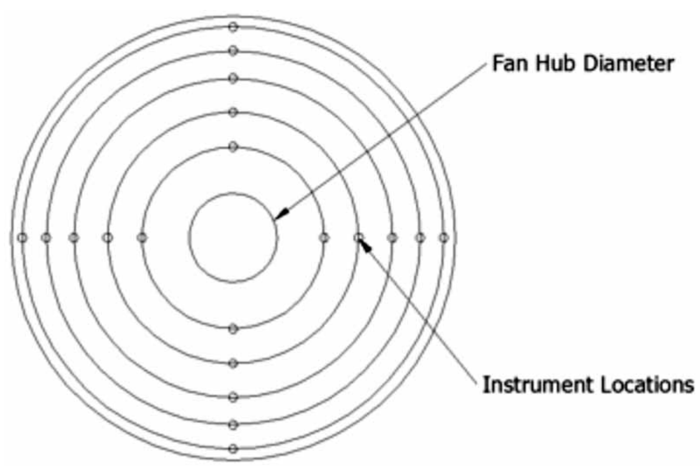

Figure 8 – Example of 20 instrument locations in fan exhaust plane during ATC-150 test.

The instruments measure the local air velocity, wet bulb temperature and dry bulb temperature at each location. The measured temperatures combined with barometric pressure can be used to calculate the air properties at each location. Following the method described in ATC-150 5.3.3.3, average air properties are found using a weighted average based on local air velocities. The mass flow of dry air can also be found by using measured air velocity and integrating over the area of the fan exhaust. It is possible to measure negative air flows near the hub of the fan, which should be omitted when finding averages and air flow. Utilizing the mass flow of dry air, inlet and exhaust humidity ratios, it is possible to compute the mass flow of evaporation.

One of the disadvantages of the long method is the complexity in taking measurements, which increases the time of testing as well as the cost to the owner of the equipment. As a result, when this test is required, it is more likely that an already equipped CTI licensed test agency will be involved. It is worth noting that the level of accuracy of propeller or vane anemometers is not as good as the other instrumentation used by CTI Licensed test agencies, such as pitot tubes and Resistance Temperature Detectors (RTDs). This may lead to greater than 5% uncertainty on air flow measurements. An S-type Pitot tube may be advisable.

Conclusion

As water, like energy, becomes a critical natural resource, it is necessary not only to develop new water conservation technologies, but also to specify methods for evaluation and field verification. This technical paper introduces two proposals.

1. The first proposal specifies how to compute the amount of water consumed on an annual basis, by using a benchmark wet cooling tower using climate data and a relatively simple method of calculation. The calculation uses simplifying assumptions, but sensitivity analyses show that the margin of error from these assumptions is reasonable. The benchmark water consumption will be used by all potential equipment suppliers. The water usage of any hybrid cooling system on an annual basis will be directly compared to the proposed benchmark to generate a meaningful percentage of water conservation.
2. The second proposal introduces two procedures for measuring the water consumed by an actual cooling system operating in the field. A short procedure, intended to be run during an ATC-105 thermal acceptance test, determines evaporation rate by the evaluation of the water level drop in the cold water basin during a measured amount of time. The long method is intended to be run during an ATC-150 plume abatement test. It measures effluent air temperatures and uses psychrometric equations to compute the average humidity ratio of the exhaust air. From that, the rate of evaporation can be determined.

The authors further propose to use this technical paper as the basis to create two new Cooling Technology Institute standards: one, a bid standard to specify how to calculate proposed hybrid cooling tower water savings; the second, a test standard to evaluate, compare and field-measure water usage. Included in the first standard, would be detailed requirements of what the utility needs to supply to potential bidders by way of climate, altitude, load, et cetera.

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[LI2] Libert, Jean-Pierre, et al. Wet/Dry Hybrid Cooling System Demonstration at The Water Research Center: Evapco Inc., 2014. [NN1] NNDC Climate Data Online. (n.d.). Retrieved from http://www7.ncdc.noaa.gov/CDO/cdoselect.cmd

Appendix A: Details of Benchmark Calculation

This appendix will detail how to calculate the benchmark wet cooling tower evaporation.

Basic Wet cooling Tower Variables

Figure (A.1) shows the water and air properties of a generic cooling tower.

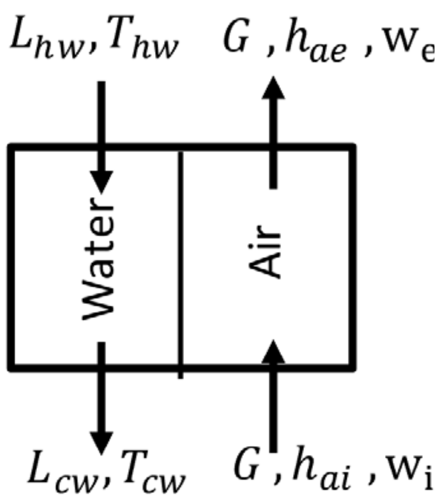

Figure A.1 – Overall heat balance for wet tower

L_{hw}=Mass flow of hot water, L_{cw}=Mass flow of cold water (kg/s)

$T_{(hw)}$=Temperature of hot water, T_{cw}=Temperature of cold water (°C)

G=Mass flow of dry air (kg/s)

h_{ae}=Enthalpy of exhaust air, h_{ai}=Enthalpy of inlet air (J/kg)

w_e=Humidity ratio of exhaust air, w_i=Humidity ratio of inlet air

Water enters the cooling tower with a given mass flow and temperature, and because of evaporation and heat transfer, it leaves at a different mass flow and temperature. The air enters the cooling tower at a given mass flow of dry air, humidity ratio, and enthalpy, and because of heat transfer and evaporation, it leaves at a different humidity ratio and enthalpy. In cooling tower applications, the mass flow of dry air remains constant for both the inlet and exhaust conditions, because no dry air mass is injected or withdrawn from the system, and the mass transfer due to evaporation involves adding vaporized liquid mass to the air, not dry mass.

Known Variables

Using the assumptions specified in Section 2, the following variables are known:

- Mass flow of hot water (L_{hw}): From volumetric water flow and hot water density
- Temperature of hot and cold water (T_{hw}, T_{cw}): Direct specification
- Inlet air enthalpy and humidity ratio (h_{ai}, w_i): From the inlet air wet bulb, assumed or specified air inlet relative humidity, and psychrometric calculations

Basic Wet cooling Tower Heat Balance

Starting with the concept that the heat entering the system is equal to the heat leaving the system, the following heat balance equation can be derived.

$$\dot{q}_{hw} + \dot{q}_{ai} = \dot{q}_{cw} + \dot{q}_{ae} \qquad \text{Eq. (A.1)}$$

$\dot{q}_{hw} = Heat\ flow\ of\ hot\ water,\ \dot{q}_{cw} = Heat\ flow\ of\ cold\ water\ (J/s)$

$\dot{q}_{ai} = Heat\ flow\ of\ inlet\ air,\ \dot{q}_{ae} = Heat\ flow\ of\ exhaust\ air\ (J/s)$

Equation (A.1) simply states that the heat flow entering the system via the water and air, must equal the heat flow leaving the system via the water and air. Equation (A.1) can then be expanded using the variables defined in Figure (A.1) and known equations for heat flow.

$$(L_{hw} \cdot c_{phw} \cdot T_{hw}) + (G \cdot h_{ai}) = (L_{cw} \cdot c_{pcw} \cdot T_{cw}) + (G \cdot h_{ae}) \qquad \text{Eq. (A.1)}$$

$c_{phw} = Specific\ heat\ of\ hot\ water\ (J/kg \cdot °C)$

$c_{pcw} = Specific\ heat\ of\ cold\ water\ (J/kg \cdot °C)$

Reducing Unknowns

To calculate evaporation, Equation (A.3) can be used.

$$Evaporation\ (Mass\ Flow) = L_E = G \cdot (w_e - w_i)\ [kg/s] \qquad \text{Eq. (A.3)}$$

However, to implement Equation (A.3), the exhaust air humidity ratio must first be determined. To include this variable, a relationship can be expressed between the mass flows of cold water, hot water and evaporation. This expression can be further expanded using Equation (A.3).

$$L_{cw} = [L_{hw} - L_E] = [L_{hw} - G \cdot (w_e - w_i)] \qquad \text{Eq. (A.4)}$$

By inserting Equation (A.4) into Equation (A.2), then assuming that the specific heat of water can be expressed using the average water temperature, Equation (A.2) can then be written as:

$$h_{ae} = h_{ai} + c_{pw}T_{cw} \cdot (w_e - w_i) + c_{pw}(T_{hw} - T_{cw}) \cdot \frac{L_{hw}}{G} \qquad \text{Eq. (A.5)}$$

Equation (A.5) now has 3 unknowns.

- Exhaust air enthalpy and humidity ratio (h_{ae},w_e)
- Mass flow of dry air (G)

To simplify the benchmark calculation, the mass flow of dry air can be assumed to be equal to the mass flow of hot water. This gives an L/G ratio of 1.0. *(Note: the effects of this assumption is detailed in Section 3.1)* Thus the equation can be simplified further.

$$h_{ae} = h_{ai} + c_{pw}T_{cw} \cdot (w_e - w_i) + c_{pw}(T_{hw} - T_{cw}) \qquad \text{Eq. (A.6)}$$

There are still two unknowns, but assuming the exhaust air is saturated, it is possible to iterate to a solution.

The first step is to approximate the enthalpy of the exhaust air by assuming that there is no loss of water from evaporation. The enthalpy of the exhaust air can be estimated by adding the inlet air enthalpy to the enthalpy removed from the water. From this value, and assuming the exhaust air relative humidity is 100%, the exhaust air humidity ratio can be approximated from psychrometric equations.

The second step is to use Equation (A.6) as intended, since an approximation of the exhaust air humidity ratio is now known. The resulting exhaust air enthalpy will be revised higher than step one, which now compensates for the water lost due to evaporation. *(Note: This makes sense when looking at the energy balance Equations (A.1) and (A.2): if there is evaporation, the mass flow of the cold water (L_{cw}) will be less, which means the heat in cold* *water will be less, which means the heat in the balancing exhaust air must be greater.)*

The second step can then be repeated as needed until the humidity ratio and enthalpy values converge; typically, this only requires one to two iterations for reasonable precision.

Once the exhaust air humidity ratio is known, the evaporation rate can then be calculated. To find the mass flow of evaporation, simply multiply the mass flow of the dry air by the difference in the inlet and exhaust humidity ratio. The volume flow of evaporation will be that value divided by the density of the water at the average water temperature.

$$Evaporation\ (Volume\ Flow) = Q_E = \frac{L_E}{\rho_{w,avg}}\ [l/s] \qquad \text{Eq. (A.7)}$$

$\rho_{w,avg}$ = Density of Water (At Average Water Temperature) (kg/l)

Example A.1 (Single Point Analysis)

This example will evaluate the estimated evaporation for a single ambient condition for a wet benchmark cooling tower, assuming the following design conditions:

- Barometric pressure: 101.2445 kPa (29.8975 inHG)
- Inlet air conditions:
  - Wet bulb temperature: $25.5\bar{5}$°C (78°F)
  - Relative Humidity: 50%
  - Dry bulb temperature: 34.3°C (93.75°F)
- Water conditions:
  - Hot water temperature (T_{hw}): $37.7\bar{7}$°C (100°F)
  - Average water temperature ($T_{(w,avg)}$): $33.6\bar{1}$°C (92.5°F)
  - Cold water temperature (T_{cw}): $29.4\bar{4}$°C (85°F)
  - Water flow (Q_{hw}): 3,785.41 l/s (60,000 GPM)

Inlet Air Properties

By using a psychrometric chart, or corresponding equations, the relevant inlet air properties can be determined.

Note: *if using a psychrometric chart, be aware that discrepancies in results can occur from barometric pressure differences. For accurate results, psychrometric equations are recommended. These equations can be found in referenced book [AS1].*

- Humidity ratio (w_i): 0.01717 (mass moisture/mass dry air)
- Enthalpy (h_{ai}): 78.513 kJ/kg (41.433 BTU/lb)

Water Properties

In order to calculate the relevant water properties, the following equations published by Detlev Kröger [Kr1] were used: *(Note: Equations valid for temperatures from 273.15K to 380K.)*

$$\rho_w = \{1.49343\cdot10^{-3} - 3.7164\cdot10^{-6}\cdot(T+273.15) + 7.09782\cdot10^{-9}\cdot(T+273.15)^2 - 1.90321\cdot10^{-20}\cdot(T+273.15)^6\}^{-1} \qquad \text{Eq. (A.8)}$$

ρ_w = Water density (kg/m³), T = Water temperature (°C)

$$c_{pw} = 8.15599\cdot10^3 - 2.80627\cdot10\cdot(T+273.15) + 5.11283\cdot10^{-2}(T+273.15)^2 - 2.17582\cdot10^{-13}\cdot(T+273.15)^6 \qquad \text{Eq. (A.9)}$$

c_{pw} = Water specific heat (J/kg · K), T = Water temperature (°C)

Also, the mass flow of hot water can be calculated as:

$$L_{hw} = Q_{hw}\cdot\rho_{hw} \qquad \text{Eq. (A.10)}$$

L_{hw} = Mass flow of hot water (kg/s)

Q_{hw} = Volume flow of hot water (l/s)

ρ_{hw} = Water density of hot water(kg/l)

By using equations (A.8), (A.9) and (A.10), the following water properties can be determined:

- Water density (ρ_{hw}): 993.156 kg/m³ (62.00 lb/ft³)
- Water density ($\rho_{(w,avg)}$): 994.584 kg/m³ (62.09 lb/ft³)
- Specific Heat (c_{pw}): 4177.4 J/kg-K (0.9978 BTU/lb-°F)
- Mass flow of hot water (L_{hw}):

$$L_{hw} = \left(3{,}785.41\frac{l}{s}\right)\cdot\left(993.156\frac{kg}{m^3}\right)\cdot\left(\frac{1}{1000}\frac{m^3}{l}\right) = 3{,}759.51\frac{kg}{s} = 497{,}298\frac{lb}{min}$$

Exhaust Air Properties – First Iteration

By using Equation (A.6) and assuming no evaporation, a first approximation of the exit air properties can be reached.

- Humidity ratio of exhaust air (w_e): $w_{e,0}=w_i=0.01717$ (mass moisture/mass dry air)
- Relative humidity of exhaust air: 100%
- Enthalpy of exhaust air (h_{ae}):

$$h_{ae,1}(SI) = \left(78.513\frac{kJ}{kg}\right) + \left(4177.4\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\frac{kJ}{J}\right)\cdot 29.4\bar{4}°C\cdot(0.01717 - 0.01717) + \left(4177.4\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\frac{kJ}{J}\right)\cdot(37.7\bar{7}°C - 29.4\bar{4}°C)$$

$$h_{ae,1}(SI) = 113.325\ kJ/kg$$

$$h_{ae,1}(IP) = \left(41.433\frac{BTU}{lb}\right) + \left(0.9978\frac{BTU}{lb\cdot°F}\right)\cdot 85°F\cdot(0.01717 - 0.01717) + \left(0.9978\frac{BTU}{lb\cdot°F}\right)\cdot(100°F - 85°F)$$

$$h_{ae,1}(IP) = 56.4\ BTU/lb$$

Note: *units of specific heat has been altered to reflect °C, which does not change the value since specific heat is calculated in the incremental unit of per degree Kelvin, or ΔK, which is the same increment as Δ°C*

Exhaust Air Properties – Second Iteration

By using a psychrometric chart, or corresponding equations, and knowing the relative humidity of the exhaust air (100%) and enthalpy of the exhaust air (113.325 kJ/kg) or (56.4 BTU/lb), the following properties can be determined:

- Humidity ratio of exhaust air (w_e): $_{we,1}$=0.03154 (mass moisture/mass dry air)
- Relative humidity of exhaust air: 100%
- Enthalpy of exhaust air (h_{ae}):

$$h_{ae,1}(SI) = \left(78.513\frac{kJ}{kg}\right) + \left(4177.4\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\frac{kJ}{J}\right)\cdot 29.4\bar{4}°C\cdot(0.03154 - 0.01717) + \left(4177.4\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\frac{kJ}{J}\right)\cdot(37.7\bar{7}°C - 29.4\bar{4}°C)$$

$$h_{ae,2}(SI) = 115.09\ kJ/kg$$

$$h_{ae,1}(IP) = \left(41.433\frac{BTU}{lb}\right) + \left(0.9978\frac{BTU}{lb\cdot°F}\right)\cdot 85°F\cdot(0.03154 - 0.01717) + \left(0.9978\frac{BTU}{lb\cdot°F}\right)\cdot(100°F - 85°F)$$

$$h_{ae,2}(IP) = 57.62\ BTU/lb$$

Exhaust Air Properties – Third Iteration

By using a psychrometric chart, or corresponding equations, and knowing the relative humidity of the exhaust air (100%) and enthalpy of the exhaust air (115.09 kJ/kg) or (57.62 BTU/lb), the following properties can be determined:

- Humidity ratio of exhaust air (w_e): $w_{(e,2)}$=0.03211 (mass moisture/mass dry air)
- Relative humidity of exhaust air: 100%
- Enthalpy of exhaust air (h_{ae}):

$$h_{ae,1}(SI) = \left(78.513\,\frac{kJ}{kg}\right) + \left(4177.4\,\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\,\frac{kJ}{J}\right)\cdot 29.4\bar{4}°C\cdot(0.03211\ -0.01717\ ) + \left(4177.4\,\frac{J}{kg\cdot°C}\right)\cdot\left(\frac{1}{1000}\,\frac{kJ}{J}\right)\cdot(37.7\bar{7}°C\ -29.4\bar{4}°C)$$

$$h_{ae,2}(SI) = 115.16\ kJ/kg$$

$$h_{ae,1}(IP) = \left(41.433\,\frac{BTU}{lb}\right) + \left(0.9978\,\frac{BTU}{lb\cdot°F}\right)\cdot 85°F\cdot(0.03211\ -0.01717\ ) + \left(0.9978\,\frac{BTU}{lb\cdot°F}\right)\cdot(100°F\ -85°F)$$

$$h_{ae,2}(IP) = 57.67\ BTU/lb$$

Since the value of the exhaust air enthalpy has changed by less than 0.1% of the previous iteration, iterating further will provide no practical accuracy improvement.

The humidity ratio of the exhaust air (w_e) after the third iteration is then:

- $w_{(e,3)}$=0.03213 (mass moisture/mass dry air)

Evaporation Calculation:

To calculate the evaporation using Equation (A.3), the mass flow of dry air (G) needs to be known. Since the benchmark wet cooling tower assumes an L/G value of 1, the following is true:

- The mass flow of exhaust air (G): = ($L_{(hw)}$) = 3,759.51 kg/s (497,298 lb/min)

Using Equation (A.3), the mass flow of evaporation (L_E) is then:

$$L_E = \left(3{,}759.51\,\frac{kg}{s}\right)\cdot(0.03213\ -0.01717) = 56.26\,\frac{kg}{s} = 7{,}441.6\,\frac{lb}{min}$$

Using Equation (A.7), the volumetric flow of evaporation (Q_E) is then:

$$Q_E = \frac{56.26\,\frac{kg}{s}}{994.584\,\frac{kg}{m^3}}\left(\frac{1000}{1}\,\frac{l}{m^3}\right) = 56.6\,\frac{l}{s} = 897\ GPM$$

Example A.2 (Yearly Evaporation Calculation)

To estimate the benchmark wet cooling tower evaporation, a weather data profile must be provided. Per the weather averaging study done in Section 3.3, the minimal data required is a single monthly average for two of the following variables:

- Inlet Wet bulb
- Inlet Dry bulb
- Inlet Relative Humidity

Taking annual weather data for New York/JFK from [NN1] and averaging values for each month, the following table of temperatures specify the temperature profile for the year.

	Wet bulb	Dry bulb	Wet bulb	Dry bulb
January	-1.142 °C	1.442 °C	29.945 °F	34.595 °F
February	-1.543 °C	1.300 °C	29.222 °F	34.340 °F
March	1.339 °C	4.646 °C	34.410 °F	40.363 °F
April	6.959 °C	10.667 °C	44.527 °F	51.200 °F
May	11.312 °C	15.188 °C	52.362 °F	59.338 °F
June	17.132 °C	20.667 °C	62.837 °F	69.200 °F
July	19.710 °C	23.850 °C	67.478 °F	74.930 °F
August	19.367 °C	23.946 °C	66.861 °F	75.103 °F
September	16.195 °C	20.329 °C	61.152 °F	68.593 °F
October	10.959 °C	14.392 °C	51.725 °F	57.905 °F
November	6.438 °C	9.500 °C	43.588 °F	49.100 °F
December	0.909 °C	3.617 °C	33.637 °F	38.510 °F

Figure A.2 – Monthly wet bulb and dry bulb profile for New York/JFK

Using the weather data from Figure (A.2), and assuming the same barometric pressure and water conditions from Example (A.1), the evaporation rate for each month can be calculated by following the same process as Example (A.1), but varying the inlet wet bulb and dry bulb.

Note: The barometric pressure and water conditions are considered by the benchmark cooling tower analysis to be constant for year round operation. This means that the cooling tower will cool the specified water flow from the specified hot water temperature to the specified cold water temperature all year round.

	$h_{ae,3}$ (kJ/kg)	$w_{e,3}$	L_E (kg/s)	Q_E (l/s)
January	43.312	0.01100	32.318	32.494
February	42.629	0.01083	32.490	32.666
March	47.724	0.01214	34.941	35.132
April	58.645	0.01507	38.951	39.163
May	68.555	0.01787	41.678	41.905
June	84.454	0.02256	44.160	44.400
July	92.735	0.02509	46.351	46.604
August	91.594	0.02474	46.889	47.144
September	81.665	0.02173	44.647	44.890
October	67.680	0.01762	40.787	41.009
November	57.527	0.01476	37.632	37.837
December	46.933	0.01193	33.736	33.920

Figure A.3 – Monthly evaporation rate calculation in SI

	$h_{ae,3}$ (BTU/lb)	$w_{e,3}$	L_E (lb/min)	Q_E (GPM)
January	26.720	0.01100	4,274.9	515.04
February	26.441	0.01083	4,297.6	517.77
March	28.510	0.01214	4,621.9	556.85
April	33.240	0.01507	5,152.4	620.75
May	37.525	0.01787	5,513.1	664.21
June	44.383	0.02256	5,841.3	703.76
July	47.964	0.02509	6,131.2	738.68
August	47.478	0.02474	6,202.3	747.25
September	43.188	0.02173	5,905.7	711.52
October	37.141	0.01762	5,395.1	650.00
November	32.748	0.01476	4,977.8	599.72
December	28.159	0.01193	4,462.5	537.64

Figure A.4 – Monthly evaporation rate calculation in IP

Taking the evaporation rates calculated in Figures (A.3) and (A.4) and multiplying by the operation time per month will result in the total monthly evaporation.

	Days	Hours	Evaporated Liters (In Millions)	Evaporated Gallons (In Millions)
January	31	744	87.032	22.991
February	28	672	79.027	20.877
March	31	744	94.096	24.858
April	30	720	101.511	26.816
May	31	744	112.238	29.650
June	30	720	115.086	30.402
July	31	744	124.823	32.975
August	31	744	126.271	33.357
September	30	720	116.354	30.737
October	31	744	109.837	29.016
November	30	720	98.073	25.908
December	31	744	90.851	24.000
Total	**365**	**8760**	**1255.199**	**331.589**

Figure A.5 – Monthly total evaporated water

Appendix B: Field Test Procedures for Evaporation

This appendix will cover the procedures to find an accurate evaporation flow of a cooling tower in operation. Two methods are explained that can be run either in tandem with an ATC-105 acceptance test or an ATC-150 plume abatement test. Additional conditions are applied to the standards for higher accuracy in the evaporation result. The evaporation curves from the manufacturer should be available for planning the test appropriately, and evaporation guarantee evaluation.

Short Method

This method is intended to be run during an ATC-105 acceptance test. In addition to the performance curve and design conditions, an accurate drawing of the basin will be required for volume calculations.

The same variations are allowable on design conditions per ATC-105. For accurate evaporation results, other conditions should be monitored and can affect the validity of the test. The tower must be balanced and each cell water flow should be within ± 5% of the average cell flow. The fans must be running at full speed and be within ± 5% of the average fan power. During the evaporation test time frame, flows such as the make-up and blow-down must be turned off. This includes any other flows that are entering or exiting the circulating water loop.

The tester must have accurate drawings of the cooling tower basin and any obstructions such as piers present in the basin. The drawing should depict all obstructions as well as any gradients in the basin. This will allow the tester to choose an ideal location to take measurements. Measurements should be taken in the spot where the water level is above any slopes for simplicity of volume calculations.

Measuring the evaporation involves taking an initial and final basin water height measurement. The time between the initial and final measurement must be recorded to find the evaporation flow. As an example, an evaporation test could run for 60 minutes. The timeframe selected should take into account the volume of water in the cooling tower, so that issues with a low water level can be avoided. The volume of evaporation per unit time can then be found using:

$$Q_E = \frac{L_{basin} * W_{basin} * (H_1 - H_0)}{t} \quad \text{Eq. (B.1)}$$

$Q_E = Volumetric\ flow\ of\ evaporation\ (m^3/hr)$

$L_{basin} = Basin\ length,\ W_{basin} = Basin\ width\ (m)$

$H_0 = Initial\ Height\ of\ water,\ H_1 = Final\ Height\ of\ water\ (m)$

$t = Time\ of\ test\ (hr)$

The short method will not work for all cooling tower set ups. The method cannot be applied to once-through cooling towers as there is no standing basin water level. There should be up-to-date drawings of the basin or rough measurements of the tower size could introduce error into the final measurement. Depending on the complexity of the basin, the short method may not be recommended for evaporation guarantees.

Example B.1 (SI)

A 4 cell cooling tower with 16.5m x 18.3m cells is being tested with a flow of 16,000 m3/h. The basin is an open rectangle with 67.8m x 20.1m. To simplify the calculation, there is no slope to the basin, and the pillar obstruction is insignificant. During a 60 minute evaporation test, the basin water level drops 235mm.

$$Q_E = 67.8m * 20.1m * 0.235m\ /1\ hour = 320\frac{m^3}{h}$$

The evaporation flow is 320 m3/h which comes out to 2.00% of the water flow. If there was an error of 4% on the basin water level change (± 7 mm on each individual measurement), the flow would be 320 ± 9.3 m3/h. The evaporation of the total flow would then be 2.00 ± .08 %.

Example B.2 (IP)

There is a 4 cell cooling tower with 54 foot x 60 foot cells cooling 70,000 GPM. The basin is an open rectangle with dimension 222 foot x 66 foot. An evaporation test is run for 60 minutes with a 9.2 inch basin water level change.

$$Q_E = 222\ ft * 66\ ft * \left(\frac{9.2}{12}\right) ft * 7.48\ \frac{gal}{ft^3} /60\ min = 1400\ GPM$$

The volumetric flow of evaporation is found to be 1400 GPM which is 2.00% of the tower water flow. If there was an error of 4% on the basin water level change (± ¼ inch on each individual measurement), the flow would be 1400 ± 56 GPM. The evaporation of the total flow would then be 2.00 ± .08 %.

Long Method

The longer and more complex method of testing for evaporation is done in conjunction with an ATC-150 plume abatement test. The design conditions variations allowable by ATC-150 must be followed. The cell must be balanced within ± 5% of the average water flow per cell. The fan power of the each cell must be within ± 5% of the average fan power. There are no restrictions on the entering and exiting flows of the water loop in this test.

Measurements are taken of the effluent air during testing. The test measures the wet bulb temperature, dry bulb temperature and local air velocity of each measuring position. It is required to have 20 measuring stations setup in the exhaust plane of the fan stack. They are aligned in equal area location along two perpendicular diameters. The data is recorded and used to find the humidity ratio of the exhaust air flow. A weighted average based on the local air velocities is used over the 20 stations. The method used can be found in ATC-150 5.3.3.3. The exhaust mass air flow is integrated over the local mass air flow and fan stack area. Negative air flows may be measured near the fan hub and should not be used in finding average conditions or mass air flow.

The mass flow of dry air can be found by integrating the local air velocities over the fan discharge area. The calculated evaporation only applies to the cell that was tested. With the assumption that each cell is identical in operation, the evaporation can be multiplied by the total number of cells for evaporation in the cooling tower.

$$Evaporation\ (Mass\ Flow) = L_E = G \cdot (w_e - w_i)\ [kg/s] \quad \text{Eq. (B.2)}$$

$$Evaporation\ (Volume\ Flow) = Q_E = \frac{L_E}{\rho_{w,avg}}\ [l/s] \quad \text{Eq. (B.3)}$$

$L_E = Mass\ flow\ of\ evaporation\ (\frac{kg}{s})\ \ G = Mass\ flow\ of\ dry\ air\ (\frac{kg}{s})$

$w_i = Inlet\ air\ humidity\ ratio\ \ w_e = Exhaust\ air\ humidity\ ratio$

$Q_E = Volume\ flow\ of\ evaporation\ (l/s)$

$\rho_{w,avg} = Density\ of\ water\ (At\ average\ water\ temperature)\ (kg/l)$