THERMAL ENERGY STORAGE FOR EXPANDED USE OF DATA CENTER INDIRECT/DIRECT EVAPORATIVE COOLING

by

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ABSTRACT

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Computer cooling system design evolved over time with goals of increasing efficiency and decreasing cost. Early computers were essentially hand-built and very expensive. Reliable operation required aggressive cooling to maintain acceptable component temperatures and this was achieved with relatively low ventilation air temperatures. With time, the scale of operations increased to the point that operating cost began to strongly influence design decisions. Computer room air conditioners consumed substantial amounts of electrical power, in some situations almost as much power as the computer equipment. One cost saving idea used outside air when the ambient temperature fell below the normal cooling supply air from the computer room air conditioner. This modification acquired the term “free-cooling”. Substantial cost savings from free-cooling led to the desire to expand its use to higher temperatures. Continuing to expand on this approach, some facilities ventured into evaporative cooling which proved highly successful in locations with an amenable climate. Water's latent heat of evaporation cooled the air using

very little electrical power. While evaporative coolers use much less power than direct expansion units of computer room air conditioners, they have more-restrictive limitations on the allowable climate conditions of temperatures and humidity. Also, by their nature, evaporative systems use considerable quantities of water. Cooling system designers continue seeking improvements in the on-going efforts to reduce operating costs.

Temperature/humidity limits for evaporative coolers are a consequence of the upper temperature limit for data center cooling supply air and thermodynamic limits of water evaporation to cool the air. Evaporative systems' cooling capacity reach a minimum during the hottest part of a 24-hour cycle. Water consumption reaches a peak at this condition as well. Designed with the necessary cooling capacity at this hot condition, the systems have excess capacity during the cooler portions of the day.

Thermal energy storage offers potential to address the two negatives of evaporative cooling, restrictive limitations and high water consumption, by time-shifting cooling capacity. Thermal energy storage enables time-shifting cooling capacity from coolest portion of the 24-hour cycle when the evaporative cooler has excess capacity. Stored cooling can augment the evaporative cooler's performance at times of challenging cooling demand during the hottest portion of the 24-hour cycle. With additional cooling from thermal energy storage the data center cooling supply air temperature can be maintained in hotter environments. Cooling from a thermal energy storage system also enables the reduction of water consumption. Thermal energy storage with free cooling, when no water is used, can provide cooling later to offset water consumption.

For thermal energy storage, phase change materials offer economic and performance advantages. The latent heat of phase change can store energy using much less material than sensible heat storage. The near-constant temperature energy exchange of phase change can improve the system thermal performance relative to energy storage with changing temperature. A commercially available thermal energy storage medium comes in the form of a water-based slurry of micro-encapsulated organic wax. The small, micron-size capsules in water overcome one of the major engineering challenges with many phase change materials, low heat transfer during the liquid to solid phase transition with low thermal conductivity material. While conductivity may be low, the maximum conduction distance is the capsule radius which is also small.

This study investigates the benefits of thermal energy storage (TES) integrated with an indirect/direct evaporative cooler in a data center application. Concepts for integration of the TES with the cooler are developed, evaluated, and compared. Performance of the most promising candidate concept is evaluated for extended temperature operation and water conservation potential at three representative geographic locations. Capital costs for TES to be integrated with an indirect/direct evaporative cooler are estimated. Finally, operating benefits in the form of reduced operating costs are combined to determine an overall cost benefit.

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CHAPTER 1

INTRODUCTION

The developed world today depends increasingly on electronic information and communication. As digital computers increased the creation of information and the storage of electronic information, access to the information became an issue. Communication of the trove of stored information started with links between a few computers. The communication network has grown to a worldwide network that has enabled new capabilities: email, world wide web, mission critical e-business functions, e-commerce, social media, entertainment, telephony, file sharing, blogging, smart-phones, cloud computing, big data, high performance computing.

Many organizations consolidate their electronic information in a facility called a data center. A data center houses equipment for data storage, data processing, and communication. This data equipment includes a number of computers to manage the storage, processing, and communication of data. These computers carry the designation of server, serving the data on demand. Small data centers may contain a few servers. With large scale consolidation, some data centers have thousands of servers.

Compute servers receive two inputs, data for processing or storage and electrical power to operate. Two products result from server operation, electronic data for storage or transmission and waste heat from the operation of the electrical components. Data centers with large numbers of compute servers produce large quantities of heat. The servers' electronic components are temperature sensitive so the data centers deal with the waste heat by providing some form of cooling.

Cooling approaches adapted as technology evolved and business imperatives changed. Reliable operation of early computer hardware was the main business imperative and this required maintaining relatively low ventilation air temperatures. Cooling air needed to be no more than 15°C (60°F). Vapor cycle systems using ammonia or chlorofluorocarbons provided a reliable source of cold air for Computer Room Air Conditioners (CRACs). These recirculate the air. All of the generated heat transfers to the air and must be removed by the CRAC.

With time, reliability improved and the scale of operations increased to the point that another business imperative, cost, rose in importance. Computer room air conditioners consumed consequential electrical power removing the total heat load, in some situations almost as much power as the computer equipment. An obvious opportunity for savings was to shut off the computer room air conditioners when the ambient temperature was below 15°C and use the outside air for cooling, termed free-cooling. Substantial cost savings from free-cooling led to the desire to expand its use to higher temperatures. Data centers accommodated this by increasing the temperature of the cooling air as long as the compute-server hardware operated reliably.

Cost of operation continued as a business imperative so cooling approaches evolved to reduce operating costs. Dry, low-humidity regions successfully use evaporative cooling in conditioning occupied spaces for so this was adapted for data center cooling. The simplest evaporative cooling designs force ambient air through a wet medium. Water evaporates from the wet medium, cooling the air and increasing the humidity. This cooled, humidified air provides the data center cooling. Increased humidity from this system raise concerns about corrosion which would adversely affect reliability so, in some cases, an indirect design is used.

Indirect cooling designs create a separate air flow where supplied water evaporates. The air is humidified and cooled then expelled. Importantly, excess water flow is likewise cooled, creating a supply of chilled water. The chilled water feeds a heat exchanger and cools the air supply for the data center without adding humidity. Indirect evaporative cooling cannot provide air temperatures as low as the direct evaporative cooling as a consequence of inefficiency with the heat exchanger. The indirect/direct evaporative cooler is a hybrid design that combines the two resulting in better capability to provide cooled air. As with the indirect evaporative system, a separate air and water supply produce chilled water which cools air for the data center using a heat exchanger. This air cooled in the heat exchanger is further cooled as it passes through a wet medium and thence to cool the data center.

Data center business still embraces the imperative to reduce costs. Evaporative cooling works best in dry, low-humidity climates, providing very cost-effective cooling. The local data center climate dictates limits for evaporative cooling. The operating cost advantage of evaporative cooling strengthens the impetus to expand the limits to data centers in warmer, more-humid locales. One approach to accommodate expanded limits goes back to vapor cycle systems. A supplemental vapor cycle direct expansion system augments the evaporative cooler. These direct expansion systems operate for a few hours of high ambient temperature and humidity when the indirect/direct evaporative cooler lack sufficient capacity. In these systems the direct expansion system utilization is very low.

Data centers operate continuously, 24 hours a day. The indirect/direct evaporative cooler capacity falls short only a portion of the twenty four hours in the day. During the remainder of the day, the indirect/direct evaporative cooler has excess capacity potential. Thermal energy storage offers an option to capture excess cooling capacity during cooler parts of the day then use this stored cooling during the warm, humid portion of the day. Thermal energy storage reduces the climate limitations allowing greater use of economical evaporative cooling.

In addition to assisting on the hottest days, a thermal energy storage system offers other benefits. Some locales experience twenty-four hour cycles that allow free cooling part of the time then the ambient warms to point that evaporative cooling is required. Thermal energy storage can offset some of this water consumption. During the free-cooling portion of the day, cooling capacity in excess of the data center demand is stored. When free-cooling is not suitable and additional cooling is required, the thermal energy storage system provides that cooling instead of water evaporation.

Thermal energy storage combined with an indirect/direct evaporative cooler can expand the operating limits to higher temperatures as well as reducing water consumption.

CHAPTER 2

LITERATURE REVIEW OF THERMAL ENERGY STORAGE

Thermal energy storage with phase change materials has a long history, starting with ice for cold storage and cooling. Ice continues as an important thermal energy storage. On-going development has found additional applications for thermal energy storage and phase change materials continue to play a role.

2.1 Thermal Energy Storage History

During the space race of the 1960's and 1970's, the United States National Aeronautics and Space Administration (NASA) sponsored numerous studies to support the technical advance of this period. In 1971, NASA released Technical Report CR-61363, Phase Change Material Handbook [1]. This thorough report summarizes seven material categories with phase change properties: paraffins, non-paraffin organics, salt hydrates, metallics, fused salt eutectic mixtures, miscellaneous, and solid-solid. Paraffins, the preferred material for this study, exhibit phase change temperatures ranging from 10°F (-12°C) to 159.8°F (71°C). The handbook lists potential space and terrestrial applications for phase change thermal energy storage. Terrestrial applications include temperature control of sensitive devices, solar energy storage, and cooking. Data centers are not mentioned.

2.2 Thermal Energy Storage Research

The basic science of thermal energy storage using phase change continues through the present day. Investigators study the characteristics and thermal exchange with phase change materials [2]-[9]. The phase change materials have been arranged in packed beds of spherical containers [10] and rectangular blocks [11]. Attempts to compensate for the low thermal conductivity of many phase change materials include the addition of high conductivity particles [12], [13] and addition of phase change material to a porous, high conductivity carbon foam [14].

Many concepts applying phase change materials for thermal energy storage appear to work well in the solid-to-liquid transition then struggle with the reverse liquid-to-solid transition. During the solid-to-liquid transition, melt liquid can be removed or the liquid may sustain high bouyancy-driven convective heat transfer. Either situation aids heat transfer to the remaining solid. In the reverse heat flow situation, liquid-to-solid, solid material often cannot be removed and conductive heat transfer is low. Build-up of solid material on the heat transfer surface becomes an increasingly thick, low conductivity barrier to heat transfer.

2.3 Thermal Energy Storage Applications

Performance of various thermal energy storage concepts and devices have been reported. Barba and Spiga reported a concept using solar energy for domestic hot water [15]. M. Cheralathan et.al. studied using off-peak electrical power to store cooling for later use [16]. Tabrizi and Sadrameli modeled a thermal storage regenerator for industrial processes [17]. Several researchers suggested and studied phase change materials for the regulation of electronic components subject to cyclic operation [1], [18], [19]. Storage of thermal energy for building space heating requirements are another possible use for phase change materials [20], [21].

The range of possible applications and the wide variety of phase change materials comprise a subject for summary and review. E. Osterman, et.al. published a review [22] in 2012 followed by Adeel Waqas and Zia Ud Zin publishing a review [23] in 2013. In 2014 the American Society of Mechanical Engineers formed Performance Test Codes Committee 53 on Energy

Storage to provide uniform test methods and procedures for the determination of the performance of energy storage systems used in electric power applications. Amy Fleischer published her book on thermal energy storage [24] in 2015, listing numerous actual and potential applications including electronic component thermal stabilization, space systems, solar utility power, textiles, domestic hot water, and building thermal management.

The building environment stabilization research with thermal energy storage has concentrated on occupied spaces where conditions are controlled for human comfort. Data center environments are subject to variations in cooling demand due to diurnal changes as well as equipment utilization and could benefit from thermal energy storage. Two research groups, Eric Fournier at The Fortress International Group [25] and Yefu Wang et. al. [26], considered ice for data center thermal energy storage. Both concluded that total power consumption was not reduced. The only benefit occurs when there exists a large price difference between day-time and night-time electric power rates. Intel published a report by Doug Garday and Jens Housley [27] with regard to a 48,000 gallon chilled water system to maintain cooling during electrical power outage. Lance Basgall's Kansas State University thesis [28] compared chilled water and ice systems where he highlighted issues with “recharging”. These chilled water- and ice-based thermal energy storage systems do not reduce total power consumption and have lengthy recharge times.

Luttrell, Guhe, and Agonafer published a 2015 study considering thermal energy storage with evaporatively cooled data centers [29] using micro-encapsulated organic wax pellets in a water slurry as a thermal energy storage system. This study concluded that modular thermal energy storage can easily integrate with current indirect/direct evaporative coolers and has the potential to operate at higher temperature-humidity conditions and reduce overall water consumption. These same authors published a 2016 study [30] which examined the effects of phase change temperature on performance. They found an upper limit for hot environment expansion and estimated potential reductions in water consumption.

CHAPTER 3

EVAPORATIVE COOLER ANALYSIS

Analysis of evaporative cooling systems requires evaluations of changing air-water vapor mixture properties, psychrometrics, and application of conservation of energy principals for each component which affects the state of the air-water vapor mixture. The two following sections elaborate on these two aspects of the analysis.

3.1 Psychrometric Analysis

Calculations of fluid and thermal properties of air with humidity use relationships from ASHRAE's Fundamentals handbook [31], many of which are also found in thermodynamics text books. Here the system of units is Imperial. Most common psychrometric charts state that indicated properties apply at standard atmospheric pressure. Occasionally a reference provides pressure corrections. Pressure has a noticeable effect on humid air properties. The model used in this study accounts for effects of pressure change on the air-water vapor mixture properties.

Solutions to psychrometric problems use a number of relationships between important properties. Some key properties are three temperatures; dry bulb, wet bulb, and dew point. Dry bulb temperature is the equilibrium temperature typically found using a regular thermometer. Wet bulb temperature is found with a special thermometer using a wet material that gives a temperature influenced by the latent heat of evaporation of the water, in thermodynamic terms “adiabatic saturation”. Dew point is the temperature at which water starts to condense as the temperature is lowered at a constant pressure. These temperature measurements allow determination of the amount of water vapor present in the air.

One of the basic assumptions regards a mixture of gases. The Dalton model of partial pressures has proven accurate for air-water vapor mixtures. The Dalton model assumes each gas constituent of the mixture exists throughout the volume at the temperature of the mixture. With regard to pressure, each constituent gas contributes a partial pressure. The sum of the partial pressures of the constituents equals the pressure of the mixture. The ideal gas relationship applies to both the air and water vapor using the appropriate partial pressure. If the mixture contacts solid or liquid water, the water vapor attains an equilibrium condition called "saturated air" where the partial pressure of the water vapor equals the water saturation pressure at the mixture temperature.

Mixtures with lower water vapor partial pressures a the same temperature are not saturated and the water vapor is considered superheated. The ratio of the actual water vapor partial pressure, P_v, to the water vapor partial pressure of saturated air, P_g, is called relative humidity, φ. With the ideal gas assumption this ratio also applies to the densities, ρ.

$$\varphi = P_v / P_g = \rho_v / \rho_g \qquad \text{E1}$$

The ideal gas assumption and partial pressures applied to the mixture allows calculation of the mass for each constituent. The ratio of the mass of water vapor, m_v, divided by the mass of the air alone, m_a, is termed the humidity ratio, ω.

$$\omega = m_v / m_a \qquad \text{E2}$$

The ideal gas equation, applied to the humidity ratio, leads to another formulation in terms of partial pressures of the water vapor and air with a leading coefficient that is the ratio of molecular weights, MW_{water} = 18.01528 g/mol and MW_{air} = 28.9645 g/mol.

$$MW_{water} / MW_{air} = 18.01528 / 28.9645 = 0.62198 \approx 0.622 \qquad \text{E3a}$$

$$\omega = 0.622\, P_v / P_a \qquad \text{E3b}$$

These results of E1 and E3b can combine to provide a relationship between relative humidity, φ, and humidity ratio, ω.

$$\varphi = \omega P_a / (0.622 P_g) = 1.608 \omega (P_a / P_g) \quad \text{E4}$$

Another important parameter is enthalpy of the air-water vapor mixture. Since both constituents are treated as ideal gases, enthalpy becomes a linear function of the dry bulb temperature using the constant pressure specific heat. The water vapor enthalpy also includes the latent heat of vaporization at 32°F / 0°C, h_{fg}, as a constant value.

$$h_a = C_{pa} * T_{db} \quad \text{E5}$$

$$h_v = C_{pv} * T_{db} + h_{fg,32°F} \quad \text{E6}$$

The enthalpy of the air-water vapor mixture, h_t, multiplies the enthalpy of each constituent by the fraction of each component. Using the humidity ratio, ω, the enthalpy of the mixture is calculated as:

$$h_t = C_{pa} * T_{db} + \omega (C_{pv} * T_{db} + h_{fg,32°F}) \quad \text{E7}$$

In the special condition of "saturated air", the wet bulb temperature equals the dry bulb temperature (and the dew point temperature), so the saturated air enthalpy, h*, would be:

$$h^* = C_{pa} * T_{wb} + \omega (C_{pv} * T_{wb} + h_{fg,32°F}) \quad \text{E8}$$

The adiabatic saturation process allows an initial air-water vapor mixture to reach the "saturated air" condition mentioned earlier. Conservation of energy then leads to the conclusion that the enthalpy of the initial mixture, h_t, plus the enthalpy of the added/evaporated water, $(\omega^* - \omega) h_w$, will equal the saturated air enthalpy, h*. Substitution of the relationships listed in E1 through E8 into the conservation of energy equation provides a relationship called the "Carrier equation" which can be solved in terms of the following properties: dry bulb temperature, wet bulb temperature, the pressure of the mixture, the partial pressure of the water vapor, and the partial pressure of saturated water vapor at the wet bulb temperature. Using appropriate constants in standard English units, the Carrier equation is:

$$P_v = P_{sat,wb} - \frac{(P - P_{sat,wb})(T_{db} - T_{wb})}{2830 - 1.44\,T_{wb}} \qquad \text{E9}$$

Three important curve fits aid the analytic solution of psychrometric conditions. These come from regressions of the data for vapor pressure versus temperature and specific enthalpy versus temperature. A curve fit of water vapor pressure as a function of temperature (degrees Rankine) uses the following equation E10:

$$P_{wv} = e^{(C1/T)} + C2 + C3(T) + C4(T^2) + C5(T^3) + C6\log(T) \qquad \text{E10}$$

C1:	-10440.397
C2:	-11.29465
C3:	-0.027022355
C4:	1.28904E-05
C5:	-2.47807E-09
C6:	6.5459673

A fourth order curve fit of specific enthalpy as a function of wet bulb temperature (degrees Rankine) uses the following equation E11:

$$h = C1 + C2(T) + C3(T^2) + C4(T^3) + C5(T^4) \qquad \text{E11}$$

C1	1.7044953911
C2	0.2196431835
C3	3.4975468427E-3
C4	-3.071955914E-5

C5 4.3355232263E-7

A fourth order curve fit of wet bulb temperature (degrees Rankine) as a function of specific enthalpy uses the following equation E12:

$$T = C1 + C2(h) + C3(h^2) + C4(h^3) + C5(h^4) \qquad \text{E12}$$

C1	-4.494078401
C2	3.8537914714
C3	-7.34956829E-2
C4	8.752025E-4
C5	-4.5497E-6

Thermodynamic properties listed below are calculated for each air-water vapor mixture flow location:

air temperature - dry bulb	partial pressure water vapor at the dry bulb temperature
air temperature - wet bulb	partial pressure water vapor at the wet bulb temperature
pressure	partial pressure water
enthalpy (intensive)	density of dry air at the dry bulb temperature
volumetric air flow	density of the air-water vapor mixture
air-water mixture mass flow	humidity ratio

Given three measurements of an air-water vapor mixture, the other properties are found using the equations E1 through E2. For example, ambient conditions provide the dry bulb temperature, wet bulb temperature and pressure and the other properties are found as follows.

1. Get vapor pressure of water at dry bulb temperature, E10

2. Get vapor pressure of water at wet bulb temperature, E10

3. Get partial pressure of water vapor, Carrier equation E9

4. Get partial pressure of dry air, P - P_v

5. Calculate absolute humidity from the partial pressures of water and air-stream, E3b

6. Get density of the water vapor using ideal gas assumption and P_v from (3)

7. Get density of dry air using ideal gas assumption and P_a from (4)

8. Add densities from (6) and (7) to get the air-water vapor mixture density

9. Calculate the specific enthalpy from wet bulb temperature, E11

Assuming a volumetric flow:

10. Calculate the mass flow from volumetric flow and mixture density

3.2 Energy Balance of Cooler Components

Properties at points in the evaporative cooler system are found starting with inlet “ambient” conditions and finding all of the associated, necessary air-water vapor mixture properties. As the air-water vapor mixture passes through a component, the component characteristics will modify some of the air-water vapor mixture properties. Conservation of mass and conservation of energy provide relationships for resolving remaining mixture properties.

3.3 Fan Component

A fan increases the pressure and increases the dry bulb temperature due to compression work and energy losses in the fan. A simple model assumes the fan produces a pressure increase with some energy losses. The energy equation can be simplified as follows:

$$Q = E_2 - E_1 + W$$
$$0 = \dot{m}_2 h_2 - \dot{m}_1 h_1 + W$$

The fan inlet condition includes all of the 12 properties listed above. The inlet conditions and fan pressure increase provide the outlet side pressure. The isentropic compression relationship gives the isentropic outlet temperature and the isentropic work can be calculated from the energy equation. The fan efficiency allows calculation of the actual work from whence the actual outlet dry bulb temperature is determined. Conservation of mass leads to the conclusion that the humidity ratio does not change. Having the outlet side pressure, dry bulb temperature and humidity ratio allows finding the remaining properties using the method listed at the end of section 3.1.

3.4 Heat Addition Component

A heat addition component models the data center behavior. The component should define the pressure loss and heat load added to the air stream. The energy equation can be simplified as follows:

$$Q = E_2 - E_1 + W$$
$$Q_{data\ center} = \dot{m}_2 h_2 - \dot{m}_1 h_1$$
$$Q_{data\ center} = \dot{m}_1 (h_2 - h_1)$$

The data center air inlet condition includes all of the 12 properties listed earlier. The inlet conditions and defined pressure loss provide the outlet side pressure. The energy equation allows calculation of the outlet dry bulb temperature. Conservation of mass leads to the conclusion that the humidity ratio does not change. Having the outlet side pressure, dry bulb temperature and humidity ratio allows finding the remaining properties.

3.5 Heat Exchanger Component

Heat exchangers create energy exchange between two fluid streams which do not mix. This study's modeling approach uses effectiveness for performance calculations. The energy equation can be simplified as follows where subscripts “a” and “b” denote different fluids:

$$\begin{aligned}
&Q = E_{2a} + E_{2b} - E_{1a} - E_{1b} + W\\
&0 = \dot{m}_{2a} h_{2a} + \dot{m}_{2b} h_{2b} - \dot{m}_{1a} h_{1a} - \dot{m}_{1b} h_{1b}\\
&0 = \dot{m}_{1a}(h_{2a} - h_{1a}) + \dot{m}_{2b}(h_{2b} - h_{1b})\\
&0 = \dot{m}_{1a} C_{pa}(T_{2a} - T_{1a}) + \dot{m}_{2b} C_{pb}(T_{2b} - T_{1b})
\end{aligned}$$

The heat exchanger air side inlet condition includes all of the 12 properties listed above while the liquid water side includes only mass flow rate and temperature. The inlet conditions and pressure drop specifications provide the air outlet side pressure. The effectiveness relationship leads to the outlet dry bulb temperature of the air side and the water outlet temperature. Conservation of mass leads to the conclusion that the air stream humidity ratio does not change. Having the outlet side pressure, dry bulb temperature and humidity ratio allows finding the remaining properties.

3.6 Evaporative Media Component

Like heat exchangers, evaporative media involve two fluid streams however there is some mixing. The mixing and conservation of mass assumption gives the conclusion that the mass flows of the individual outlet streams are different from the inlet mass flows. This study's modeling approach uses saturation efficiency for calculating the amount of water evaporated from the water supply mass flow which transfers the air stream. The energy equation can be simplified as follows where subscripts “a” and “b” denote different fluids:

$$\begin{aligned}
&Q = E_{2a} + E_{2b} - E_{1a} - E_{1b} + W\\
&0 = \dot{m}_{2a} h_{2a} + \dot{m}_{2b} h_{2b} - \dot{m}_{1a} h_{1a} - \dot{m}_{1b} h_{1b}
\end{aligned}$$

The evaporative media air side inlet condition includes all of the 12 properties listed above while the liquid water side uses only mass flow rate and temperature. The inlet conditions and pressure drop specifications provide the air outlet side pressure. The saturation efficiency relationship leads to an air side outlet humidity parameter. In this model the result is the air outlet partial pressure of water vapor which leads to the outlet humidity ratio and calculation of the total mass of water evaporated. The mass of water evaporated along with the latent heat of evaporation defines the enthalpy change of the air side. Outlet temperature of any excess water flow depends on the component performance specification. Having the air outlet side pressure, partial pressure of water vapor, and enthalpy allows finding the remaining properties using the method listed at the end of section 3.1.

CHAPTER 4

APPROACH TO EVALUATING THERMAL ENERGY STORAGE SYSTEMS FOR EVAPORATIVE DATA CENTER COOLING

This study evaluates the practicality of thermal energy storage to extend the ambient temperature and humidity range and reduce water consumption of indirect/direct evaporative cooling for data centers. Thermal energy storage concepts herein use a slurry of micro-encapsulated phase change material in water, introduced in the publications by Luttrell, et.al. [29-30]. A data center cooling system incorporating thermal energy storage can time-shift cooling capacity. The constraint of contemporaneously matching cooling production to cooling demand is removed. Instead, a time-integrated cooling capacity takes advantage of periods where cooling capacity exceeds the cooling demand to augment other times when cooling capacity has little margin.

Figure 4-1 illustrates notional diurnal variations of the ambient temperature and cooling capacity of an evaporative cooling system, compared to a constant cooling demand. The evaporative cooler design meets the cooling demand at the most severe (highest) ambient temperature, and the cooler has excess capacity throughout most of the 24 hour cycle. Thermal energy storage permits time-shifting cooling capacity, storing cooling capacity when there is excess, then using this stored cooling to help meet the needs at other operating conditions. This enables improvements to the evaporative cooler operation which includes water use reduction.

The objectives of the evaporative cooling/thermal energy storage study are:

- Concept development and comparative evaluation of indirect/direct evaporative coolers with integrated thermal energy storage

- Assessment of the thermal energy storage system expanded operating range compared to existing indirect/direct evaporative coolers
- Assessment of the thermal energy storage system reduction in water use compared to existing indirect/direct evaporative coolers
- Assess the thermal energy storage system cost benefits compared to existing indirect/direct evaporative coolers

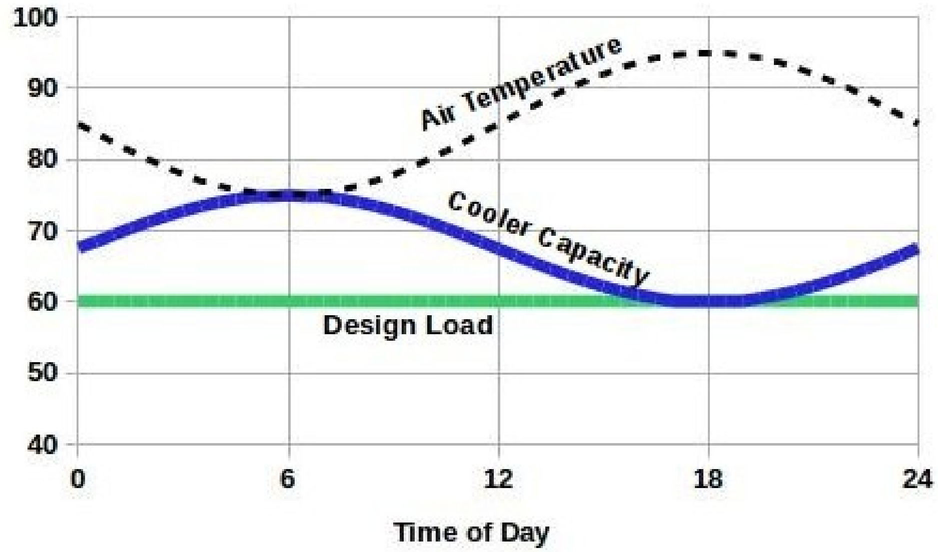

Figure 4-1. Diurnal Variation of Evaporative Cooler Capacity

Comparative evaluation of evaporative coolers with integrated thermal energy storage involves both subjective and objective aspects. Subjective evaluations depend on experience and opinion of experienced experts. Objective evaluation requires a method of determining the performance of the system. For an objective comparative evaluation, simulation is preferred over the time and expense of constructing prototype hardware. Evaluation of the concepts needs a flexible, scalable model of a thermal energy storage system and a flexible model of an indirect/direct evaporative cooler capable of modification to integrate a thermal energy storage system at any point in the evaporative cooler.

4.1 Indirect/Direct Evaporative Cooler Simulation

The indirect/direct evaporative cooler model determines fluid conditions at the inlets and exits of the major components using a one-dimensional, thermodynamic/fluid dynamic approach. It is recognized that many components have two-dimensional variations of fluid properties across their inlets and outlets however, the one-dimensional approach assumes that mean values adequately represent the thermodynamic/fluid dynamic state for a system-level evaluation. Humid air properties, that is “psychrometrics”, are calculated using relationships and coefficients from Chapter 3. The selected modeling environment is Octave*, a high level language for numerical computing.

As the design basis, a notional baseline indirect/direct evaporative cooler is assumed to provide a data center air flow of 10,000 cubic feet per minute (283.2 cubic meters per minute). With a twenty degree Fahrenheit (11.1°C) temperature rise through the data center, the baseline cooling capacity is 67.7 kilowatts. For maximum thermodynamic efficiency, the cooling tower has a maximum air flow comparable to the data center air flow, 10,000 cubic feet per minute (283.2 cubic meters per minute). Four environment conditions specify the operating condition of the indirect/direct evaporative cooler. These four conditions are: 1) ambient pressure, 2) dry bulb temperature, 3) wet bulb temperature, and 4) make-up water temperature. Several additional parameters are specified to define operating conditions to be similar to an existing commercial

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* GNU Octave version 4.0.0 manual: a high-level interactive language for numerical computations.
John W. Eaton, David Bateman, Søren Hauberg, and Rik Wehbring (2015).
URL http://www.gnu.org/software/octave/doc/interpreter/

indirect/direct evaporative cooler. These performance parameters include: evaporative media effectiveness and pressure loss, heat exchanger effectiveness and pressure loss, data center pressure loss, and a fan inefficiency parameter. Appendix A provides details on these parameters.

The baseline model of the indirect/direct evaporative cooler determines liquid water and air-water mixture conditions at twelve locations. Four locations are associated with the liquid water circulation from the cooling tower sump, through the direct side heat exchanger, through the cooling tower evaporative media, and back to the sump. At the four locations of the water loop the model calculates the water temperature and flow rate only. The sump receives a water flow to make-up for evaporation and its temperature is an input parameter for the simulation. Properties are calculated at three air flow locations related to the cooling tower and five air flow locations related to the cooler direct side. Complexity of the cooling tower thermodynamics results in using two internal locations, one of which is an intermediate "psuedo-state" which accommodates non-adiabatic saturation. All twelve of the locations are described below:

1. Water leaving the sump (temperature and flow only)
2. Water leaving the direct side heat exchanger (temperature and flow only)
3. Water entering the cooling tower (temperature and flow only)
4. Water leaving the cooling tower (temperature and flow only)
5. Ambient air entering the cooling tower
6. Cooling tower air psuedo-state
7. Cooling tower evaporative media outlet

8. Ambient air entering the direct side

9. Air exiting the direct side heat exchanger

10. Air exiting the direct side evaporative media

11. Air exiting the direct side fan

12. Air exiting the data center

Some cooler components are not simulated. The conditions are not calculated for the outlet of the cooling tower fan as this does not affect the data center cooling performance. Pressure loss and associated property changes are not calculated for the direct side filter. This pressure loss is comparatively small and has little effect on the performance of the cooler. Property changes are not calculated that are attributable to the water pump power losses as the effect is small for a sub-cooled liquid.

4.2 Thermal Energy Storage Simulation

Thermal energy storage simulation, inherently transient, involves the time-integrated accounting of energy flows as the state-of-charge changes. In this case the periods cover one or more diurnal periods. The longest thermodynamic transients of the baseline indirect/direct evaporative cooler last a few minutes (for the water sump and evaporative media), therefore the performance of the cooler is treated as reaching steady state conditions fast enough that transient effects are negligible in relation to the thermal energy storage system. Modeling of the time integration of the thermal energy storage system initially used an implementation of the Runge-Kutta method which is a standard function in the modeling environment.

Like other thermal energy storage systems, temperature stratification plays an important role in maximizing the performance. Temperature stratification of the material in the storage tank intentionally minimizes mixing as a result of fluid flow in and out. The purpose is to maintain the lowest possible supply temperature for cooling and maintain the highest possible supply temperature during recharging (solidifying) the thermal energy storage phase change material. Discretizing the thermal energy storage into multiple energy storage lumps simulates the stratification. To maintain stratification when the thermal energy storage system switches between charging and discharging modes the controls must reverse flow through the system.

Energy storage resulting from the phase change uses an approach of modifying the specific heat versus temperature relationship. Analysis of one representative material in slurry form shows a specific heat of 0.86 Btu/lbm-F (3.60 kJ/kg-K) to be suitable while the material does not change phase. Increasing the specific heat to 30.95 Btu/lbm-F (129.6 kJ/kg-K) over a 1°F (0.56°C) band produces the desired enthalpy change. Discontinuities occur in the specific heat versus temperature relationship at the lower and upper end of the phase change temperature band. Some integration routines experience instability when inputs are discontinuous. A check of this found the selected integration routine handles the discontinuities without issue.

Early analyses using the discretized thermal energy storage and modified specific heat versus temperature relationship made it clear that using phase change thermal energy storage outside the phase change temperature should be avoided. Keeping the phase change temperature near the operating temperature at the interface minimizes entropy loss. If the system operated away from the phase change temperature, recovering back to the phase change temperature offset any benefit from sub-cooling or super-heating the material. As a consequence the thermal energy storage model simplified from the Runge-Kutta method to just tracking thermal energy capacity operating at the phase change temperature. Simple additions and subtractions of thermal energy allow tracking the thermal energy storage system state-of-charge.

All of these analyses assume the data center operates at a constant full load. A steady lower load will probably have little effect on the outcome of this study. A variable load profile for the data center will affect the results. For example, a reduced data center load at night could allow a greater energy storage to offset more day time loads. Or, a higher load at night could reduce the energy storage, but this may be sufficient for a lower day time load. Load profiles vary significantly so each potential installation must consider the anticipated profile.

CHAPTER 5

THERMAL ENERGY STORAGE INTEGRATED WITH

AN INDIRECT/DIRECT EVAPORATIVE COOLER

5.1 Baseline Indirect/Direct Evaporative Cooler

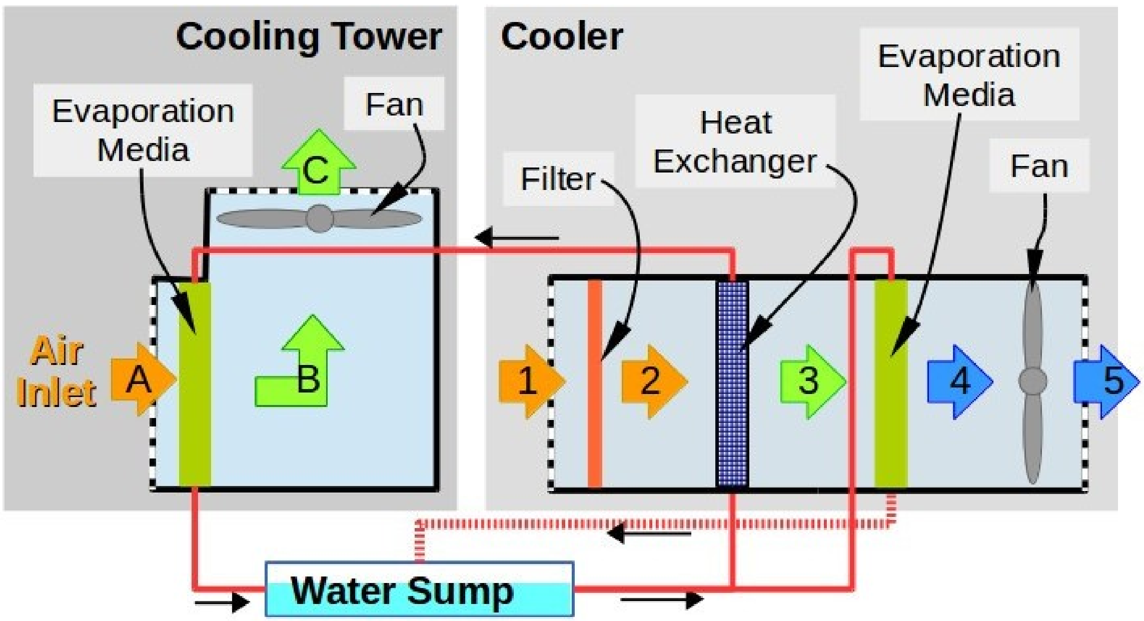

Figure 5-1. Baseline Indirect/Direct Evaporative Cooler

Figure 5-1 schematically illustrates an indirect/direct evaporative cooler. This type of cooler consists of two major sections, the cooling tower and the evaporative cooler. The cooling tower cools a water supply by an in-flow of ambient air which passes through a flooded evaporative media. In this media some of the water is evaporated and the latent heat of evaporation cools both the air and excess water. The cooler, moist air is exhausted from the cooling tower with a fan. The excess water from the media drains and collects in a water sump. A pump supplies the water in the sump to the cooler section.

The cooler section major components include filter, heat exchanger, evaporative media, and fan. A filter removes particulates from the ambient air. The heat exchanger allows thermal exchange between the filtered air and water from the sump. Cooled water from the sump passes through the heat exchanger, lowering the temperature of the filtered air without adding humidity to the air. Water warmed in the heat exchanger returns to the cooling tower to be cooled again. Air exiting the heat exchanger passes next through another evaporative media where it is further cooled by the latent heat of evaporation, increasing the humidity. This two-step cooling process can result in an air supply which is cooler than the ambient wet bulb temperature. Finally the air passes through a fan and thence to the data center to cool the electronic equipment.

5.2 Integration of Thermal Energy Storage

As a ground rule, concepts utilizing thermal energy storage should minimize interference with normal operation of the baseline cooler. The concepts interface with the data center cooler such that much of the time the cooler operates normally. Interruption of the thermal energy storage operation will not halt the baseline cooler operation. Some off-peak operation allows charging the thermal energy storage system by cooling it. During high temperature operation the interface design augments the data center cooler resulting in a lower supply air temperature or reduces the consumption of water by evaporation.

Thermal storage concepts fall into two major categories. The first major category, Single Interface, is identified by concepts which are "charged" (store cooling by solidifying the phase change material) and "discharged" (absorb heat/release cooling by melting the phase change material) with a single interface to the indirect/direct evaporative cooler. The thermal energy storage interface uses a single heat exchanger in these concepts.

A water sump interface design inserts a heat exchanger which receives water leaving the sump before the cooler-side heat exchanger, Figure 5-2. The thermal energy storage system provides the other input to the heat exchanger. During off-peak operation, the pump circulates the slurry and charges the thermal energy storage until all of the phase change material is solidified and the pump stops. When the control system determines the stored capacity should be used, the pump reverses flow direction and circulates the slurry. When the capacity is depleted or is no longer required, the control system stops the pump.

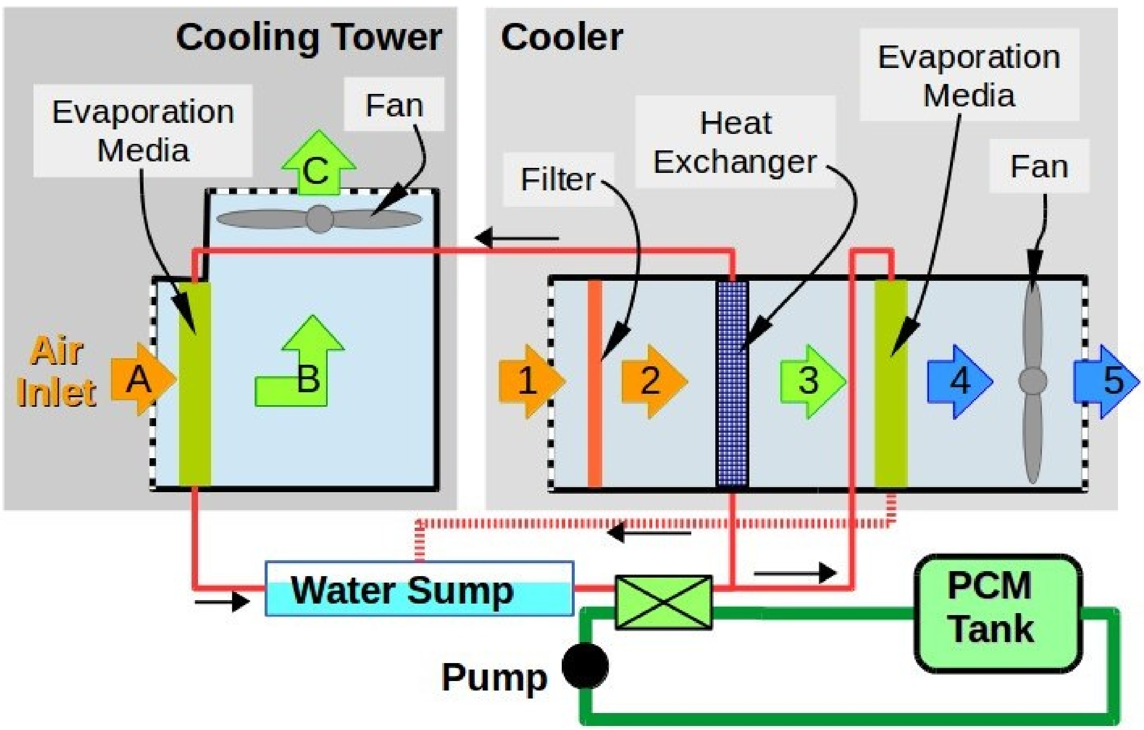

Figure 5-2. Water Sump Interface Concept

Concepts with the interface heat exchanger on the cooler side display much commonality. A post-filter interface design inserts a heat exchanger in the cooler-side air stream after the filter and before the cooler-side heat exchanger, Figure 5-3. As with the water sump interface concept, the thermal energy storage system provides one input to the heat exchanger. Concepts with the interface upstream of the filter offer no additional benefit and are not considered. A post-heat exchanger interface design inserts the thermal energy storage heat exchanger in the cooler air stream after the cooler-side heat exchanger and before the evaporative media, Figure 5-4. A <u>post-media</u> interface design inserts the thermal energy storage heat exchanger in the cooler air stream after the cooler-side evaporative media and before the fan, Figure 5-5. A <u>post-fan</u> interface design inserts the thermal energy storage heat exchanger in the cooler air stream after the cooler-side fan before distribution to the data center, Figure 5-6. Controls are similar to the water sump interface concept, with the pump circulating the slurry during off-peak operation to charge the thermal energy storage. When the control system determines the stored capacity should be used, the pump reverses flow direction and circulates the slurry. When the capacity is depleted or is no longer required, the control system stops the pump.

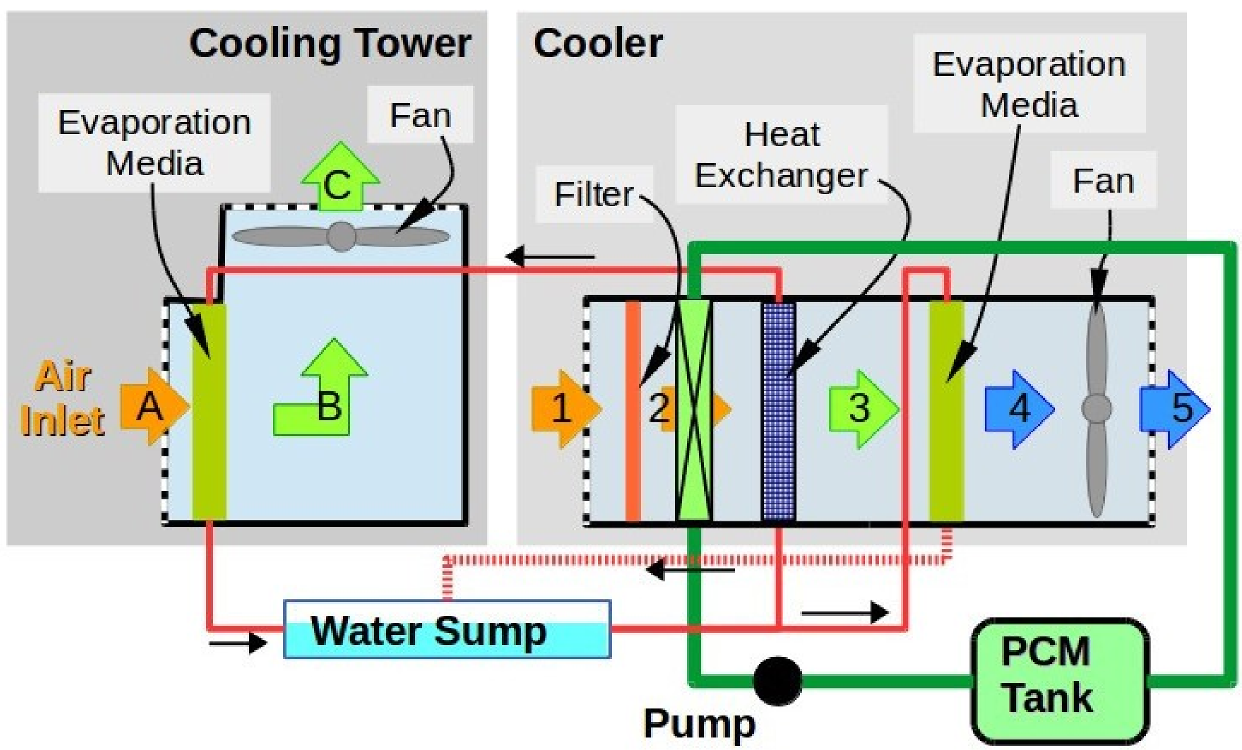

Figure 5-3. Post-Filter Interface Concept

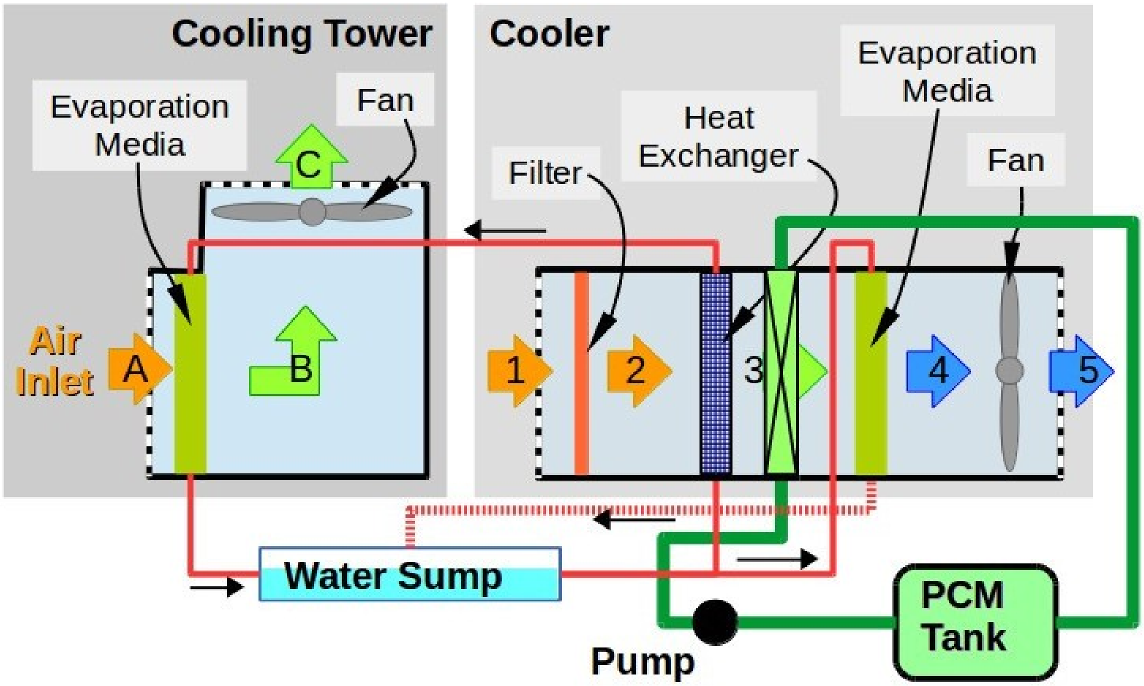

Figure 5-4. Post-Heat Exchanger Interface Concept

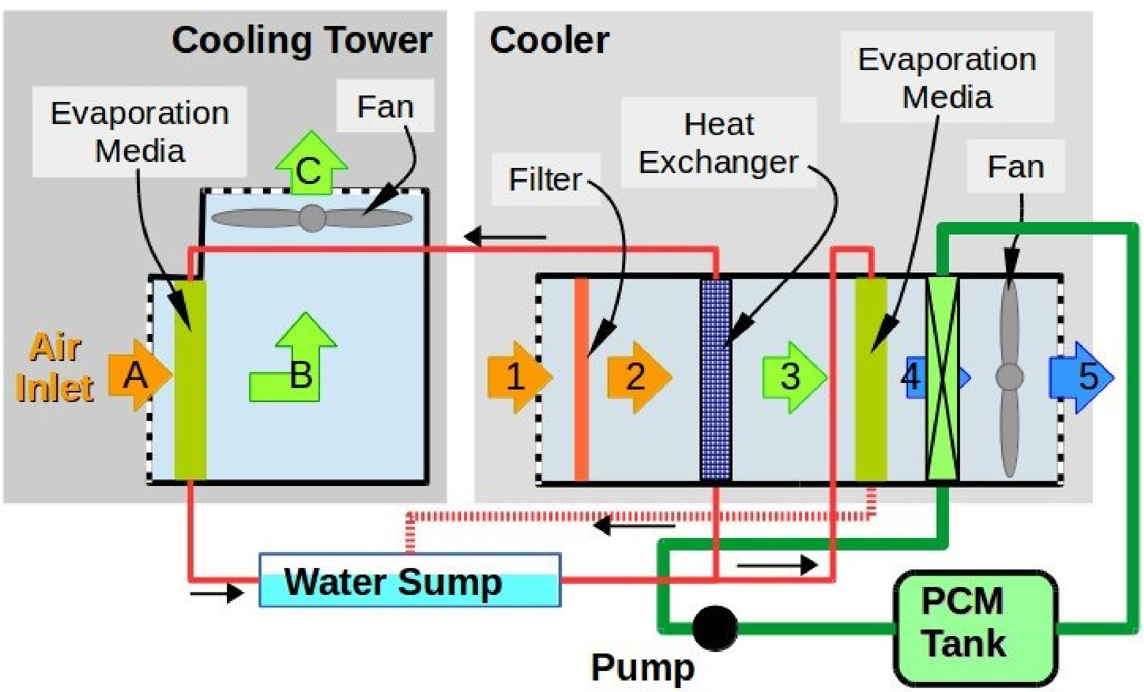

Figure 5-5. Post-Media Interface Concept

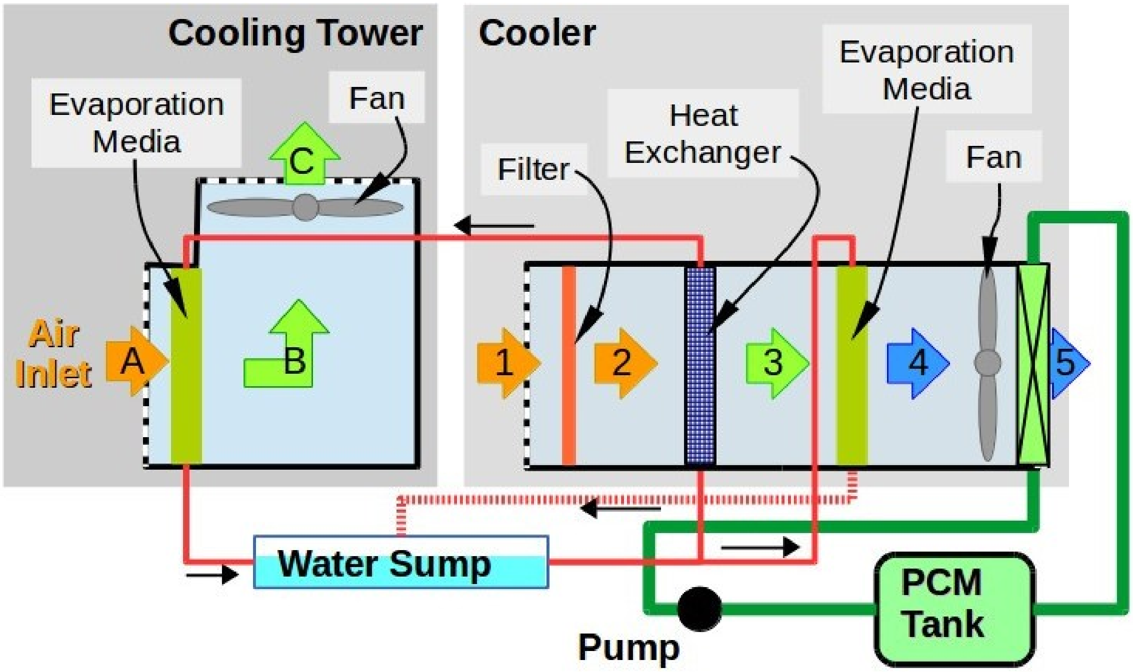

Figure 5-6. Post-Fan Interface Concept

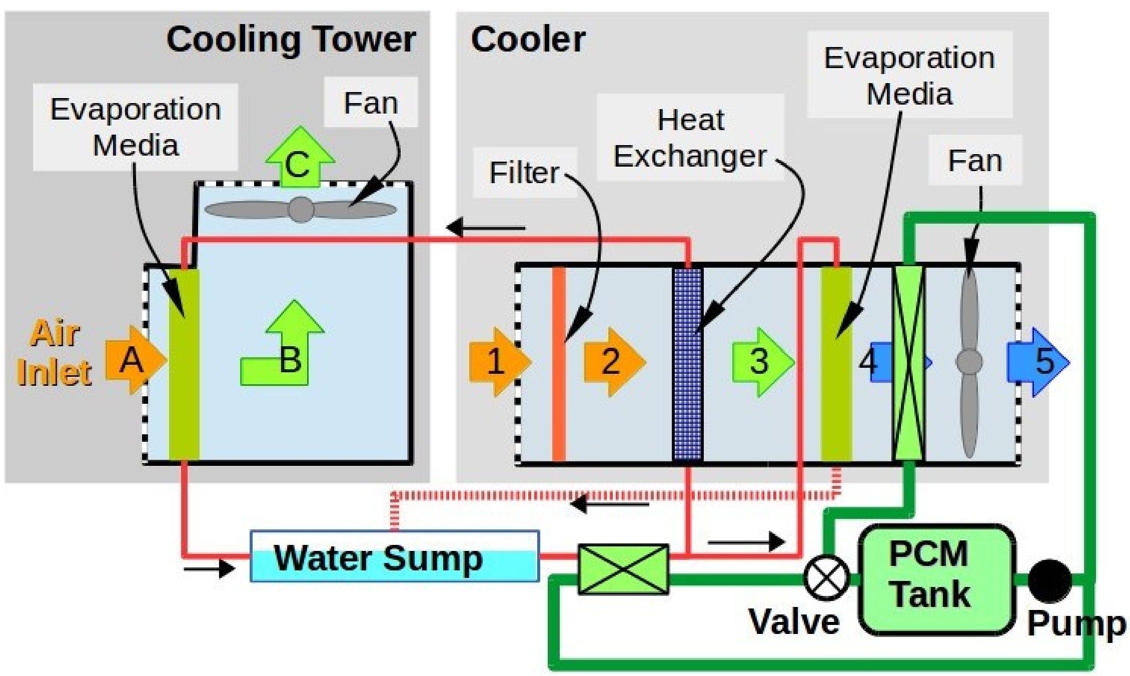

Figure 5-7. Example Dual Interface Concept

The second major category of integrated thermal energy storage, dual interface, is identified by concepts that "charge" via interface to one location in the cooler and "discharge" via a second location in the cooler. Figure 5-7 illustrates one concept where the thermal energy storage interfaces with the supply air to “charge” the thermal energy storage and uses a second interface with the water sump to “discharge” the thermal energy storage. Multiple arrangements are possible.

CHAPTER 6

COMPARATIVE EVALUATION OF CANDIDATE CONCEPTS

The comparative evaluation consists of two parts. The first part is a subjective assessment integration of thermal energy storage. The second part is an initial quantitative assessment of performance.

6.1 Subjective Design Evaluations

Subjective design evaluation considers the relative complexities for integrating thermal energy storage with the existing indirect/direct evaporative cooler hardware. Thermal energy storage adds major components internal to the baseline cooler, primarily one or more heat exchangers. Added sub-components include such items as sensors, valves, and switches to monitor and control the modified system. The baseline system may require other hardware changes or additions such as re-orienting existing components or upgrading the supply air fan to accommodate the thermal energy storage. The baseline system controls are modified to include the thermal energy storage.

The combined impacts of these aspects of thermal energy storage integration receive an integration complexity factor (ICF) for each concept. This complexity factor accounts for non-recurring costs of redesign and retooling, increased recurring and installation costs, increased service and maintenance costs. The factor for non-recurring expenses (NRE) ranges from 0 (zero) to 1 (one). These expenses receive less importance because they are amortized over a production run. The factor for recurring and installation expenses (RE) ranges from 0 (zero) to 5 (five). These charges directly affect the capital expense. The factor for service and maintenance expenses (MAINT) ranges from 0 (zero) to 4 (four). These charges directly affect the operating expense. Adding the three factors together gives the integration complexity factor which is 0 (zero) for the baseline indirect/direct evaporative cooler and ranges up to a maximum of 10 (ten).

Not surprisingly, non-recurring expenses vary among the concepts. A multiple interface design has the highest complexity with two new heat exchangers and extra valves for control so it received the highest value of 1.0. The interface designs which add a heat exchanger to the cooling air flow have the next highest complexity due to the size of the heat exchanger and a potential need for air bypass provisions to reduce system pressure drop for and idle thermal energy storage system. These receive an NRE value of 0.5. The water sump interface design adds the lowest complexity with a comparatively small liquid-to-liquid heat exchanger. This receives an NRE value of 0.25.

Variation of recurring expenses result from the required tank size with the associated volume of phase change material slurry and the cost of heat exchangers. Some results from the performance evaluations in the follow section provide estimates of the tank sizes for the concepts. Two heat exchangers plus valves and an unknown tank volume support assigning a factor of 4 to the multiple interface designs. Also assigned a factor of 4, the post-filter concept uses a single large heat exchanger and requires the largest tank volume. The water sump concept uses a smaller heat exchanger and comparably large tank, so it receives a factor of 3. The post-heat exchanger concepts requires the larger air-stream heat exchange but operates with a smaller tank, so it receives a factor of 3. Both the post-media and post-fan require the larger heat exchanger however these two require the smallest tank volume and therefore receive an RE factor of 2.

The last basic factor estimates cover service and maintenance expenses. Again, due to the complexity, the multiple interface concept receives the highest factor of 3. The four concepts which integrate the new heat exchanger have the easiest service and maintenance, worthy of a factor of 1. The water sump concept receives a factor of 2 due to slightly more complex servicing with two liquid interfaces.

The following table summarizes these subjective factors and lists the Integration Complexity Factor (ICF).

SYSTEM CONCEPT	NRE	RE	MAINT	ICF
Baseline	0	0	0	0
Water Sump	0.25	3	2	5.25
Post-Filter	0.5	4	1	5.5
Post-Heat Exchanger	0.5	3	1	4.5
Post-Media	0.5	2	1	3.5
Post-Fan	0.5	2	1	3.5
Multiple	1.0	4	3	8.0

Table 6-1. Subjective Evaluation Factors

6.2 General Performance Evaluations

The quantitative assessment defines thermodynamic characteristics of each concept in comparison to the baseline system. These quantitative assessments utilize the simulation methods mentioned in chapter 4 as applied to the concepts presented in chapter 5.

The first thermodynamic characteristic is the cooling energy storage potential which represents the energy, or cooling, charge that can be stored. Cooling energy storage potential is calculated from baseline indirect/direct evaporative cooler performance during a diurnal cycle when the cooler has excess cooling capacity. The storage potential is a function of the phase change temperature. Thermal energy storage only charges when the charge location temperature at the interface is below the phase change temperature. The representative quantity is the time integral of the difference between the temperature at the charge location, T_{CL}, and the phase change temperature, T_{PC}, when the value T_{PC} - T_{CL} is positive. Considering phase change temperatures, T_{PC}, over the diurnal range of temperatures at the charging source location, the cooling energy storage potential is a function of the phase change temperature.

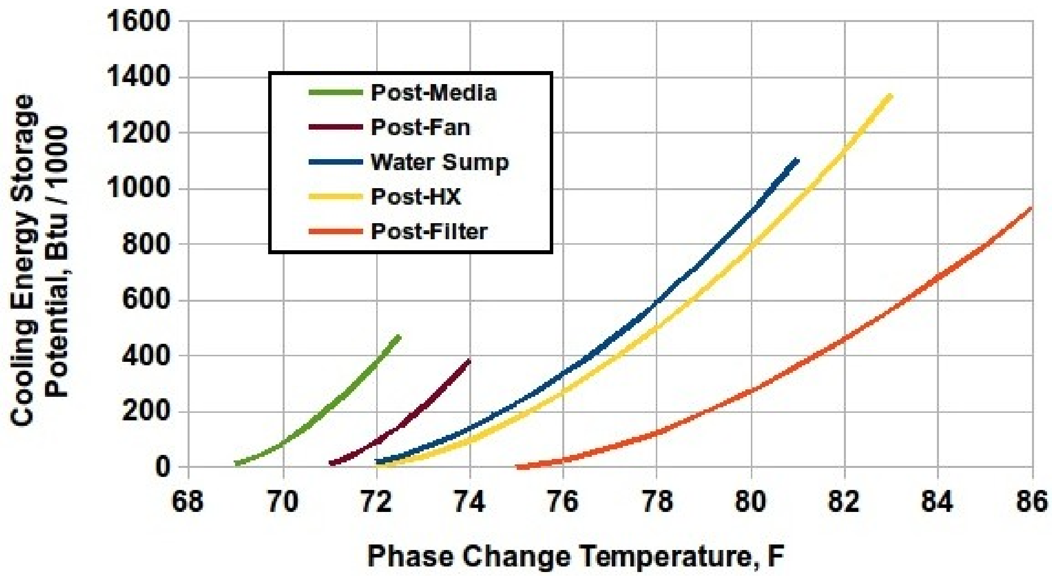

Figure 6-1. Cooling Energy Storage Potential for 5 Configurations

Results of the evaluation are shown in Figure 6-1. Here it is clear the Post-Heat Exchanger concept has the potential to store over 1.2 million Btu's (1.266 MJ) of cooling while the Post-Fan concept has a maximum potential of 337 thousand Btu's (356 MJ). However this does not mean that the Post-Heat Exchanger concept is the better design. These results suggest the storage tank sizes used in section 6.1 to rank recurring expenses and the useful phase change temperatures.

The second thermodynamic characteristic is the supply air temperature depression potential. Ideally, stored cooling lowers the temperature <u>at the cooler interface location</u> of an indirect/direct evaporative cooler to the phase change temperature. Supply air temperature depression potential is the difference between the maximum allowable supply air temperature ($T_{SA,max}$) and the supply air temperature if the interface location is reduced to the phase change temperature. Considering phase change temperatures (T_{PC}) over the range of temperatures at the cooling source location identified above, the supply air temperature depression potential will be defined as a function of the phase change temperature.

Results of this evaluation in Figure 6-2 show trends opposite to the cooling potential. Each configuration has the greatest effect on the cooling air temperature at the lowest usable phase change temperature. The lowest usable phase change temperature also corresponds to the lowest cooling potential.

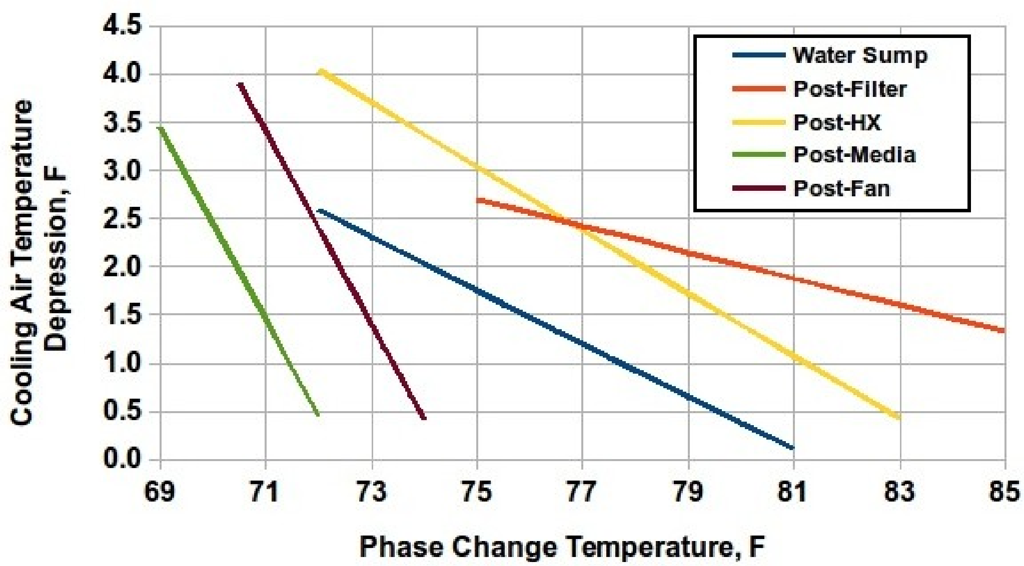

Figure 6-2. Cooling Air Temperature Depression for 5 Configurations

The two characteristics are analyzed together. Phase change temperatures at the low end of the range have the greatest impact on supply air temperature depression but have the least cooling energy storage potential. Conversely phase change temperatures at the high end of the range will have a smaller effect on supply air temperature depression but have the largest cooling energy storage potential. The next part of the performance evaluation identifies an optimum phase change temperature for each concept and the corresponding thermal energy storage. Knowing the thermal energy storage capacity allows calculation of the phase change material mass and storage tank size.

6.3 Specific Scenarios Performance Evaluations

Thermodynamic simulations of each concept result in three figures of merit. Each figure of merit derives from simulation of a single twenty-four hour diurnal cycle. A synthetic diurnal cycle provides a standard for comparison. The diurnal peak temperature and humidity condition are close to the ASHRAE A2 envelope limit from "Thermal Guidelines for data Processing Environments" [32], 95°F (35°C) dry bulb temperature and 75°F (24°C) wet bulb temperature. The synthetic diurnal assumes the absolute humidity remains constant and the diurnal minimum dry bulb temperature is 75°F (24°C). Dry bulb and wet bulb temperatures are constrained to follow sinusoidal cycles. As a result, dry bulb temperature is a sinusoidal cycle between 75°F (24°C) and 95°F (35°C) and wet bulb temperature is a sinusoidal cycle between 69.5°F (20.8°C) and 75°F (24°C).

The first scenario finds the cooling supply air temperature reduction relative to the baseline cooler on a hot, humid day. Three parameters of the thermal energy storage system are adjusted for the lowest possible supply air temperature for the hot portion of a diurnal cycle using cooling capacity stored during the cooler portion of the diurnal cycle. Phase change temperature is the first parameter. This establishes how much thermal energy may be stored or released. The second parameter is the temperature below which the thermal energy storage system is charging, solidifying the material, using excess cooling capacity of the cooler. Charging too early introduces a higher interface temperature which raises the supply air temperature. The third parameter is the temperature above which the thermal energy storage system is discharging, melting the material, providing cooling to the cooler. This parameter balances the application of cooling to straddle the peak temperature equally. The difference in temperatures between the baseline system supply air and each concepts' time-integrated supply air at the peak diurnal temperature condition constitutes one figure of merit, temperature improvement of the supply air (TISA). The baseline indirect/direct evaporative cooler supply air temperature improvement is 0 (zero).

The second scenario evaluates the performance of each concept while increasing peak diurnal dry bulb temperatures and maintaining the same absolute humidity until the time-integrated supply air temperature equals the baseline system supply air temperature at the baseline temperature and humidity diurnal cycle. Again, parameters of the thermal energy storage system are adjusted to achieve best performance. The difference between the peak diurnal temperature and the baseline temperature constitutes a second figure of merit, temperature improvement of the environment (TIE). The baseline indirect/direct evaporative cooler environment temperature improvement is 0 (zero).

The third scenario evaluates the performance of each concept for water consumption reduction with a diurnal cycle that varies from a low temperature of 65°F (18°C) to a high temperature of 85°F (29°C) with a fixed humidity of 0.010 lb_{water} / $lb_{dry\ air}$ (0.010 kg_{water} / $kg_{dry\ air}$). This cycle allows nocturnal free cooling and otherwise requires evaporative cooling. The reduction in water consumption constitutes a third figure of merit, reduction of water consumption (RW).

Selection of the most promising concept(s) incorporates results from the preceding evaluations: hardware integration complexity factor, supply air temperature improvement, environment temperature improvement, and reduction of water consumption. Two ratings for each concept are calculated per the formulae below. The hot performance rating (HPR) formula rates environment temperature improvement significantly more important than supply air temperature improvement and reduces the combination by the inverse of the integration complexity factor. The water reduction rating (WRR) adjusts the reduction of water consumption (RW) by the inverse of the complexity factor. The concepts with the highest HPR and WRR rating will be selected for further evaluation.

HPR = (3*TISA + TIE) * (1/ ICF)

WRR = RW * (1/ICF)

6.4 Specific Scenarios Performance Results

Results of analyses for the first scenario find the thermal energy storage parameters which produce the best temperature improvement of the cooling air supply, TISA. Analysis of the water sump interface concept considers phase change temperatures from 75°F (24°C) to 77.5°F (24°C). Phase change temperatures outside this range provide lower performance. The highest TISA value is 1.875°F (1.04°C) at a phase change temperature of 76.5°F (24.7°C) with a thermal energy storage capacity of 709,620 Btu (0.7486 MJ). Analysis of the post-filter interface concept considers phase change temperatures from 85°F (29.4°C) to 87°F (30.6°C). Phase change temperatures outside this range provide lower performance. The highest TISA value is 1.866°F (1.04°C) at a phase change temperature of 85°F (29.4°C) with a thermal energy storage capacity of 1,570,260 Btu (1.657 MJ). Analysis of the post-heat exchanger interface concept considers phase change temperatures from 77°F (25°C) to 79°F (26.1°C). Phase change temperatures outside this range provide lower performance. The highest TISA value is 1.877°F (1.04°C) at a phase change temperature of 78°F (25.6°C) with a thermal energy storage capacity of 819,660 Btu (0.8647 MJ). Analysis of the post-media interface concept considers phase change temperatures from 70°F (21.1°C) to 71°F (21.7°C). Phase change temperatures outside this range provide lower performance. The highest TISA value is 1.881°F (1.05°C) at a phase change temperature of 70.5°F (21.4°C) with a thermal energy storage capacity of 282,000 Btu (0.2975 MJ). Analysis of the post-fan interface concept considers phase change temperatures from 72°F (22.2°C) to 73°F (22.8°C). Phase change temperatures outside this range provide lower performance. The highest TISA value is 1.902°F (1.06°C) at a phase change temperature of 72.5°F (22.5°C) with a thermal energy storage capacity of 222,000 Btu (0.2342 MJ). Table 6-2 summarizes these results with the indicated phase change temperatures (T_{PC}).

SYSTEM CONCEPT	T_{PC}	TISA	CAPACITY
Baseline	NA	0	0
Water Sump	76.5°F / 24.7°C	1.875°F / 1.04°C	709,620 Btu 748.6 MJ
Post-Filter	85°F / 29.4°C	1.866°F / 1.04°C	1,570,260 Btu 1656.6 MJ
Post-Heat Exchanger	78°F / 25.6°C	1.877°F / 1.04°C	819,660 Btu 864.7 MJ
Post-Media	70.5°F / 21.4°C	1.881°F / 1.05°C	282,000 Btu 297.5MJ
Post-Fan	72.5°F / 22.5°C	1.902°F / 1.06°C	222,000 Btu 234.2 MJ

Table 6-2. TISA Specific Performance Results

The five concepts produce comparable temperature improvement of the supply. The required thermal energy storage varies almost by an order of magnitude. It is noteworthy that the post-media and post-fan concepts accomplish the improvement with substantially smaller thermal energy storage capacity.

Results of analyses for the second scenario indicate the capability of each concept to expand the operating envelope to higher temperatures. The water sump, post-filter, and post-heat exchanger interface concepts have the capability to provide the same cooling supply air temperature when the diurnal temperatures increase more than 9°F (5°C). The post-media and post-fan interface concepts have the capability to provide the same cooling supply air temperature when the diurnal temperatures increase slightly less than 8°F (4.4°C). Table 6-3 summarizes these results.

SYSTEM CONCEPT	T_{PC}	TIE	CAPACITY
Baseline	NA	0	0
Water Sump	81°F / 27.2°C	9.37°F / 5.20°C	369,048 Btu 389.4 MJ
Post-Filter	94°F / 34.4°C	9.10°F / 5.05°C	774,679 Btu 817.3 MJ
Post-Heat Exchanger	83°F / 28.3°C	9.34°F / 5.19°C	444,279 Btu 468.7 MJ
Post-Media	72°F / 22.2°C	7.68°F / 4.26°C	168,418 Btu 177.7 MJ
Post-Fan	74°F / 23.3°C	7.95°F / 4.42°C	178,014 Btu 187.8 MJ

Table 6-3. TIE Specific Performance Results

Here again the post-media and post-fan interface concepts energy storage capacities are smallest. The post-filter interface concept requires more than four times larger capacity.

Results of analyses for the third scenario indicate the capability of each concept to reduce water consumption. The water sump interface concept has the lowest water use reduction. Post-heat exchanger, post-media, and post-fan interface concepts have the highest water use reduction potential. All three require comparable size thermal energy storage capacity of 521,829 Btu (550.5 MJ). The post-fan concept is limited to a narrow range of phase change temperatures. Figure 6-3 presents plots of these results.

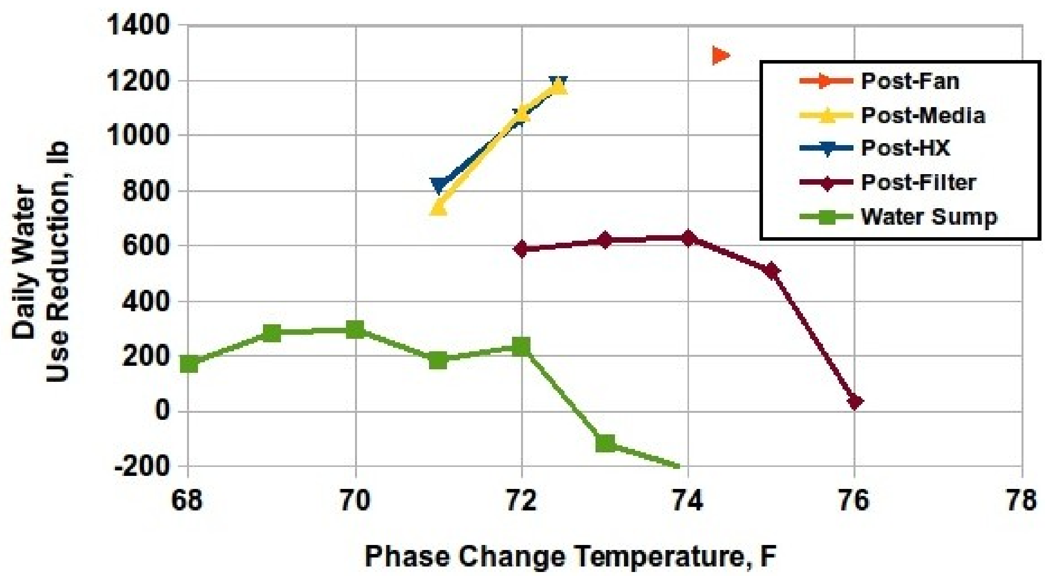

Figure 6-3. Water Reduction Specific Performance Results

The two figures of merit, HPR and WRR are calculated from the results above. Note that the phase change temperatures differ for the TISA and TIE results and the RW factor is a function of phase change temperature. If hot environment performance and water use reduction have equal value, the factors must use the same phase change temperature. Only the post-media and post-fan allow use of the same phase change temperature, hence a single material. Table 6-4 lists figures of merit for the five concepts along with the required thermal energy storage capacity. The post-media and post-fan interface concepts have the two highest ranks for both hot environment operation, HPR, and water use reduction, WRR. These two concepts require the smallest thermal energy storage capacity for hot environment operation. Water use reduction storage capacity of these two concepts fits in the middle of the range for all concepts. The post-media and post-fan interface concepts can use the same phase change temperature for both hot environment operation and water use reduction.

SYSTEM CONCEPT	HPR	Capacity for HPR	WRR	Capacity for WRR
Baseline	0	NA	0	NA
Water Sump	2.855	709,620 Btu 748.6 MJ	56.9	235,474 Btu 248.4 MJ
Post-Filter	2.672	1,570,260 Btu 1656.6 MJ	114.5	692,654 Btu 730.8 MJ
Post-Heat Exchanger	3.328	819,660 Btu 864.7 MJ	263	520,186 Btu 548.8 MJ
Post-Media	3.817	282,000 Btu 297.5MJ	338.4	521,362 Btu 550.0 MJ
Post-Fan	3.902	222,000 Btu 234.2 MJ	369.4	521,829 Btu 550.5 MJ

Table 6-4. Figures of Merit

The post-media and post-fan interface concepts rank closely. Phase change temperature becomes the deciding factor for the detail evaluation. The post-media concept could operate well with a phase change temperature of 72°F (22.2°C) which is close to some commercially available materials and producers indicate the temperature is tailorable. The post-media concept progresses to the detail analyses with this phase change temperature.

CHAPTER 7

PERFORMANCE EVALUATION OF POST-MEDIA INTERFACE CONCEPT

Thermal energy storage is intended to augment the performance of an indirect/direct evaporative cooler.

7.1 Sinusoidal Diurnal Limits

One goal for using phase change materials in an indirect/direct evaporative cooler is expansion of the operating range to higher temperature and humidity limits. Achievement of this goal will expand the market for evaporative coolers for the data center cooling application. The first step in characterizing the performance advantages of the candidate thermal energy storage concepts is determining the operating hot/humid operating limits. The analysis of the temperature improvement of the environment (TIE) found that the post-media interface concept has the capability to extend operation beyond the baseline with a peak temperature of 95°F (35°C) to a diurnal that is 7.68°F (4.26°C) hotter while maintaining a constant absolute humidity of approximately 100 grains of water per pound of dry air (14.3 grams of water per kilogram of dry air) or about 40% relative humidity. The hot-humid environment limits are further defined by evaluating the post-media interface concept at higher absolute humidity. The dry bulb temperature diurnal cycle remains constant from a low of 75°F (24°C) to a peak of 95°F (35°C) while the wet bulb temperature cycle increases. The thermal energy storage system maintains a suitable cooling air supply temperature when the humidity increased to 108 grains of water per pound of dry air (15.4 grams of water per kilogram of dry air) which is a relative humidity of 43.4%.

The results of these assessments provide an indication of the expanded capability of an indirect/direct evaporative cooler augmented with thermal energy storage. These studies are performed using a sinusoidal variation of the dry bulb temperature and constant absolute humidity. Actual diurnal behaviors differ from this assumed sinusoidal pattern, but a sinusoidal variation provides a standard for comparison without relying on any specific geographic location. The next step is evaluating the candidate concept using realistic climate data.

7.2 The Meteorological Year

Artificial diurnal cycles are useful for screening the concepts. The actual diurnals form the basis for evaluating specific performance. Actual diurnal cycles are site-specific. The National Renewable Energy Laboratory (NREL) maintains meteorological data for over 1000 locations in the United States. The Meteorological Years from the NREL are a derived data set representing one full year with hourly meteorological data for each location which includes the information for this study; dry bulb temperature, relative humidity, and barometric pressure. Data from three locations are selected to evaluate the performance of the post-media interface thermal energy storage concept. Humidity levels are intentionally different. The locations reveal the capability of the system with varying levels of humidity: hot coastal (Orlando, Florida), hot humid inland (Dallas, Texas), and hot dry (Phoenix, Arizona). Table 7-1 presents temperature extremes from The Meteorological Year datasets for each location.

The Meteorological Year data provide input to models of the indirect/direct evaporative cooler. The baseline cooler is the first version of the model and does not include thermal energy storage. This version provides the basis for comparison. The second model version includes thermal energy storage with a capacity near that indicated in Table 6-4, above: 522,000 Btu

(550.7 MJ). Specific geographic locations may benefit from a larger or smaller capacity. Therefore, a third model doubles the energy storage capacity and the fourth model halves the energy storage capacity.

LOCATION	Maximum Dry Bulb	Minimum Dry Bulb	Maximum Dew Point	Minimum Dew Point
Orlando	96°F (35.6°C)	32°F (0°C)	81°F (27.2°C)	5°F (-15°C)
Dallas	104°F (40°C)	11°F (-11.7°C)	80°F (26.7°C)	-4°F (-20°C)
Phoenix	112°F (44.4°C)	36°F (2.2°C)	74°F (23.3°C)	-8°F (-22.2°C)

Table 7-1. TMY Extreme Temperatures

7.3 Simulation Controls

Simulations for the three locations of Orlando, Dallas, and Phoenix, use the post-media concept model and run the full year datasets. The thermal energy storage starts fully charged, assuming that the system takes advantage of low winter temperatures to develop the initial charge while free cooling is prevalent. The model tracks state-of-charge for the thermal energy storage system, preventing exceedence beyond fully charged or discharged.

The cooling tower fan speed adjusts to regulate cooling. Actual systems stop the cooling tower fan during free cooling. The simulation limits the minimum fan flow to 200 cubic feet per minute (5.66 cubic meters per minute) even during free cooling for numerical stability. This does result in some small additional water use which might not occur in the actual system.

Controls of the post-media concept begin the year operating in water use reduction mode and this continues as the spring-time conditions turn warmer. The mode switches from water use reduction to hot environment improvement when the weather conditions are consistently warm during summer months. Entering autumn, when the weather conditions moderate, the control mode switches back to water use reduction. Prior to this portion of the study, models had fixed control logic. The Meteorological Year simulations include logic to automatically switch between these modes depending on conditions.

The primary difference between the two modes is control of the direct side evaporative media which operates as either OFF or ON. This operation is different from the cooling tower where the fan speed adjusts to provide proportional regulation of the total cooling. The water use reduction mode reduces use of the direct side evaporative media, absorbing excess cooling (if any) when the direct side evaporative media operates, and offsetting use of this evaporative media with stored cooling when possible. Hot environment improvement mode normally focuses on providing cooling if conditions exceed the capability of the indirect/direct evaporative cooler to maintain an acceptable cooling air supply temperature. Logic for this mode selection looks at the maximum dry bulb temperature over the preceding 48 hours. Water use reduction mode operates if that maximum is less than 88°F /31.1°C. Otherwise controls select the hot environment improvement mode. The control logic provide basic control for this mode selection. A specific site may benefit from adjustments to this logic.

January 1 results from the Orlando simulation provide an example of the water use reduction mode benefit. Figure 7-1 presents a 24-hour history of the ambient temperature (Tamb on the left vertical scale), the thermal energy storage state-of-charge (TES on the left vertical scale), water use of the baseline cooler (WU_bl on the right vertical scale), and water use of the post-media interface cooler (WU_tes on the left vertical scale). The ambient temperature holds steady around 60°F (15.6°C) during the night from hours 1 to 11. At hour 12 the ambient temperature increases to, and holds steady near, 79°F (26.1°C) until hour 17 and then begins a slow decline back toward 60°F (15.6°C). The water use of the baseline cooler increases from near-zero to more than 160 pounds of water per hour from hour 13 to hour 17, then drops back to near-zero after hour 18. This evaporative cooling episode consumes over 136 gallons (513 liters) of water. The post-media interface cooler with integrated thermal energy storage substitutes stored cooling for water consumption. The state-of-charge begins dropping when the higher ambient temperature occurs and ultimately drops to about 7% of capacity at hour 18. The thermal energy system then fully recharges by hour 23. During this 24-hour cycle the post-media interface system consumes 3.6 gallons (13.4 liters) of water, less than 3% of the water used by the baseline cooler. Note that the thermal energy storage is fully recharged and ready for another occurrence.

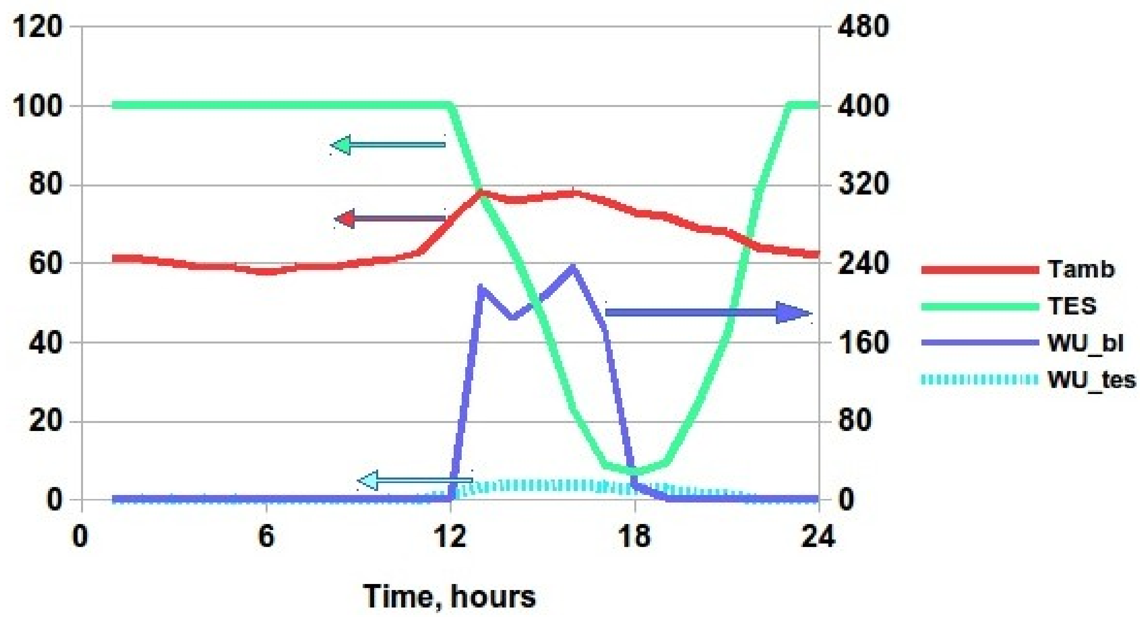

Figure 7-1. Orlando January 1 Water Use Reduction Example

July 31 - Aug 1 results from the Dallas simulation provide an example of the hot environment improvement mode. Figure 7-2 presents a 48-hour time history of the ambient temperature (Tamb on the left vertical scale), the thermal energy storage state-of-charge (TES on the left vertical scale), the baseline system cooling supply air temperature (Tsa_bl on the right vertical scale), and the thermal energy storage system cooling air supply temperature (Tsa_tes on the right vertical scale).

The ambient dry bulb temperature appears benign with a maximum near 90°F (32.2°C) near the end of these two days but the relative humidity averages 91% for the first 36 hours. The baseline system fails to maintain a supply air temperature of 75°F (23.9°C) at several time points and is consistently above that value from hour 12 (noon) on July 31 until hour 23. The thermal energy storage system responds during this time, the supply air temperature is steady and the state-of-charge decreases then recovers on August 1.

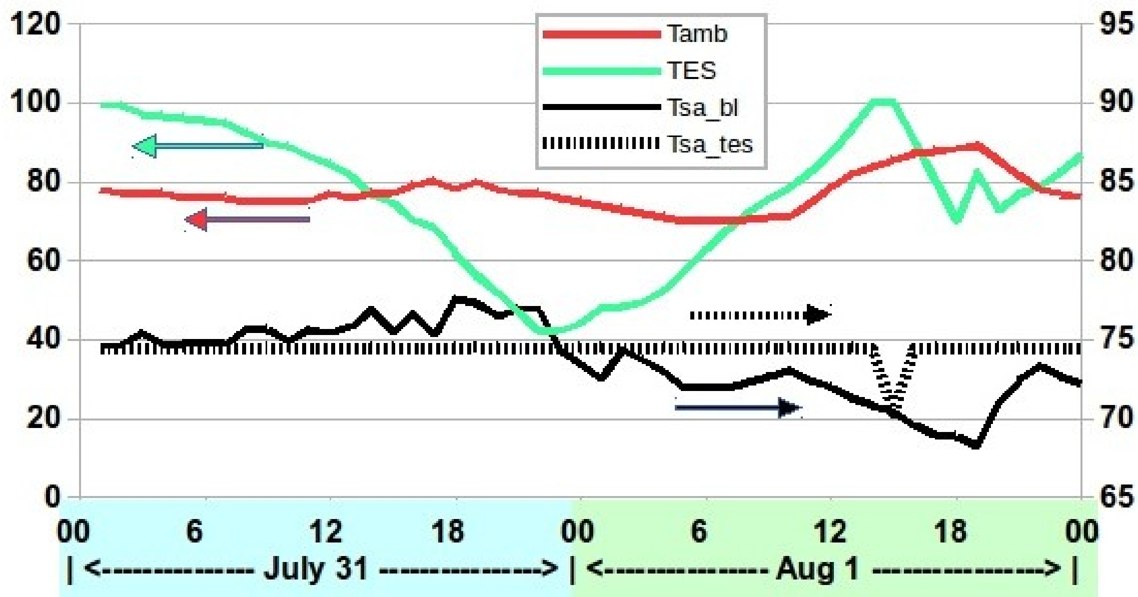

Figure 7-2. Dallas July 31-Aug 1 Hot Environment Improvement Example

A subordinate mode in the hot environment improvement mode increases utilization of the thermal energy storage. This subordinate mode engages the thermal energy storage if recharge conditions are expected to be favorable. Logic for this subordinate mode selection looks at the minimum wet bulb temperature over the preceding 48 hours and the current time-of-day. If the minimum wet bulb temperature over the preceding 48 hours does not exceed 69°F /20.6°C and the time-of-day is between 3pm and 7pm and the state-of-charge is greater than 80% , stored cooling can augment the cooler to reduce water use. In the example, the cooling capacity of the baseline evaporative system can meet the supply air temperature requirement, aided by low humidity. The thermal energy storage system is available to reduce water consumption. As with the other mode selection, this control logic should be tailored to the conditions at a specific location to obtain optimum benefit. August 10 results from the Phoenix simulation provide an example of this control feature of the hot environment improvement mode.

Figure 7-3 presents the hourly variations of the ambient temperature (Tamb on the left scale), the thermal energy storage state-of-charge (TES on the left scale), water use of the baseline cooler (WU_bl on the rightscale), and water use of the post-media interface cooler (WU_tes on the right scale). The ambient temperature holds steady around 85°F (29.4°C) during the night from hours. At hour 8 the ambient temperature increases gradually to 107°F (41.7°C) at hour 18 and then decreases. The water use of the baseline cooler increases from 300 to a peak about 850 pounds of water per hour. The thermal energy storage system substitutes stored cooling for water consumption. The state-of-charge and water use drop about hour 14. Then control logic briefly uses excess capacity to recharge the thermal storage, increasing water consumption for this period. Then the state-of-charge and water use drop again at hour 18. The thermal energy storage is fully recharged and ready for another occurrence after hour 20. This event saves 89 gallons (337 liters) of water.

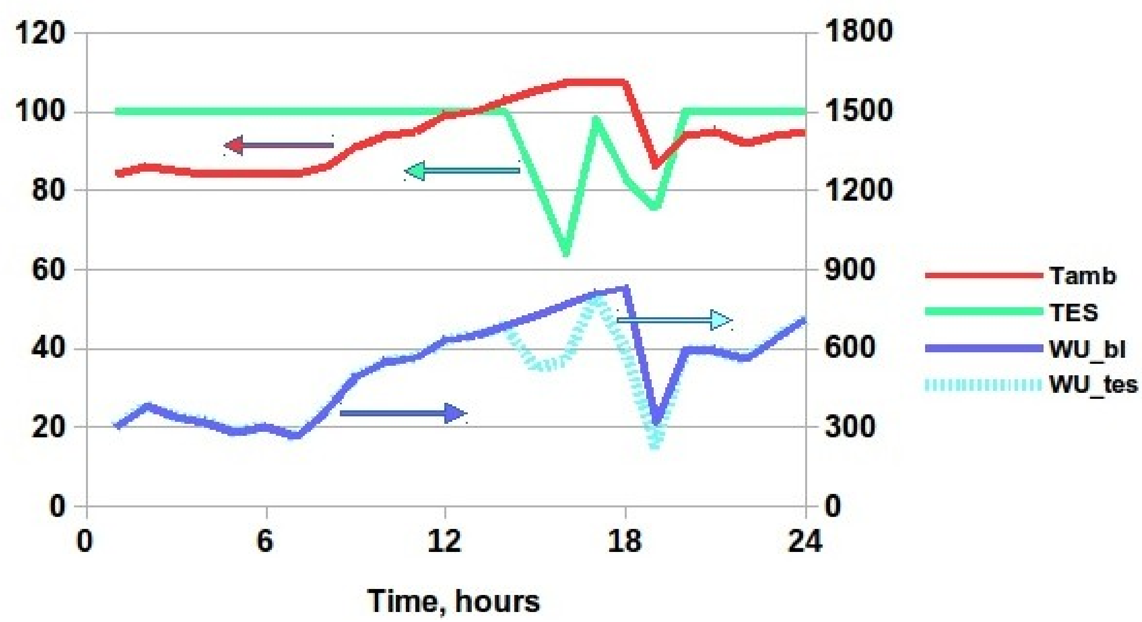

Figure 7-3. Phoenix August 10 Hot Environment Subordinate Mode Example

Actual cooler controls implement numerous other functions which are not included in the simulation model. These regulate operation for optimum performance, protection of the data center equipment, and self-protection of the cooler equipment. For example, controls may recirculate data center exhaust air to keep the minimum supply air temperature above freezing. This protects the heat exchanger filled with water. Hot environment improvement and water use reduction should not be affected by this control. However this control benefits the modified system by also keeping the thermal energy storage heat exchanger above freezing. Recirculated data center exhaust air protects both systems from freezing.

7.4 Results for Orlando

Orlando, Florida experiences a coastal, hot, humid climate where evaporative cooling would be expected to provide limited benefit. Figure 7-4 shows a plot of the hourly dry bulb ambient temperature for the Orlando dataset from The Meteorological Year, confirming the limit values in

Table 6-1. The winter temperatures are quite mild and summer temperatures are very warm. The 15-20°F (8.3-11.1°C) diurnal temperature spread during the summer indicates high relative humidity. The remainder of the year has a similar relatively small diurnal temperature spread indicative of high humidity, but day to day variation make this less obvious.

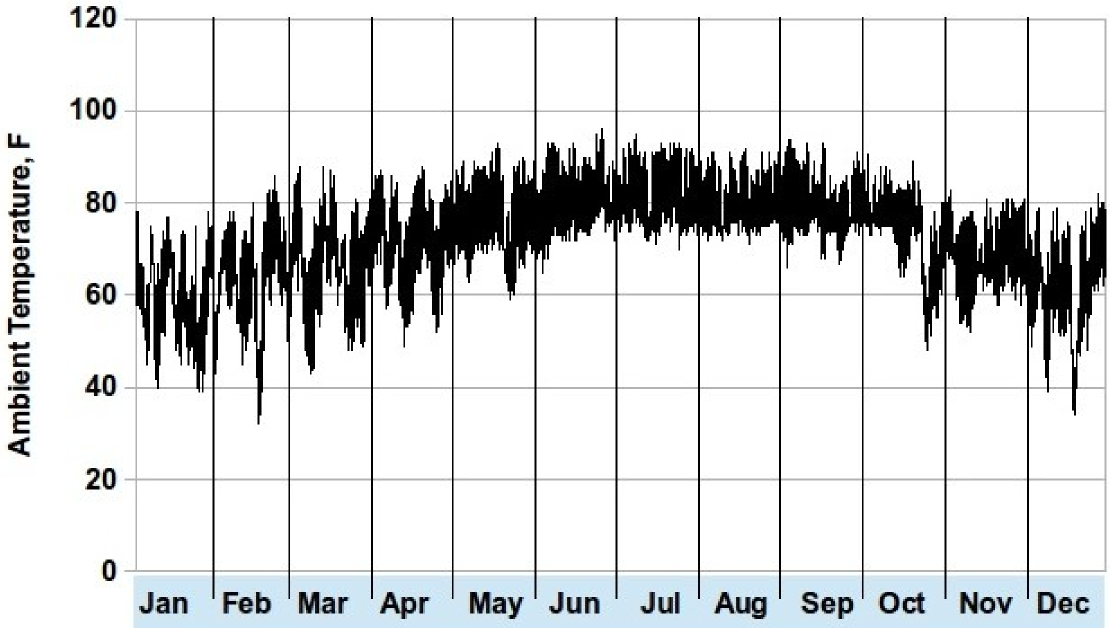

Figure 7-4. Orlando TMY Hourly Dry Bulb Temperature

The evaporative cooler alone may not be a good option for a warm humid climate. A cooling "shortfall" occurs when the cooling air supply temperature exceeds the design limit of 75°F (24°C). The cooling shortfall equals the difference between the actual and desired cooling supply air extensive enthalpies multiplied by the duration. Simulation finds this under-capacity occurs several times. The Orlando simulation results find the baseline indirect/direct evaporative cooler requires additional cooling in the amount of 37.76 million Btu (39,840 MJ) over time periods totaling 2101 hours. Water consumption of the evaporative cooler gives an indication of how much cooling the system produces. The simulation finds the baseline cooler evaporated

80,646 gallons of water. Additional water is drained to control total dissolved solids without being evaporated. Drain water adds 2420 gallons for a total water consumption of 83,066 gallons for the year.

Simulation of the indirect/direct evaporative cooler with the post-media interface concept compares favorably with the baseline. One way of observing the thermal energy storage relative performance is by tracking the state-of-charge. The thermal energy storage state-of-charge is normalized, ranging from zero (0) when fully discharged to 100 when fully charged. Figure 7-5 presents the Orlando state-of-charge overlaying the ambient dry bulb temperature.

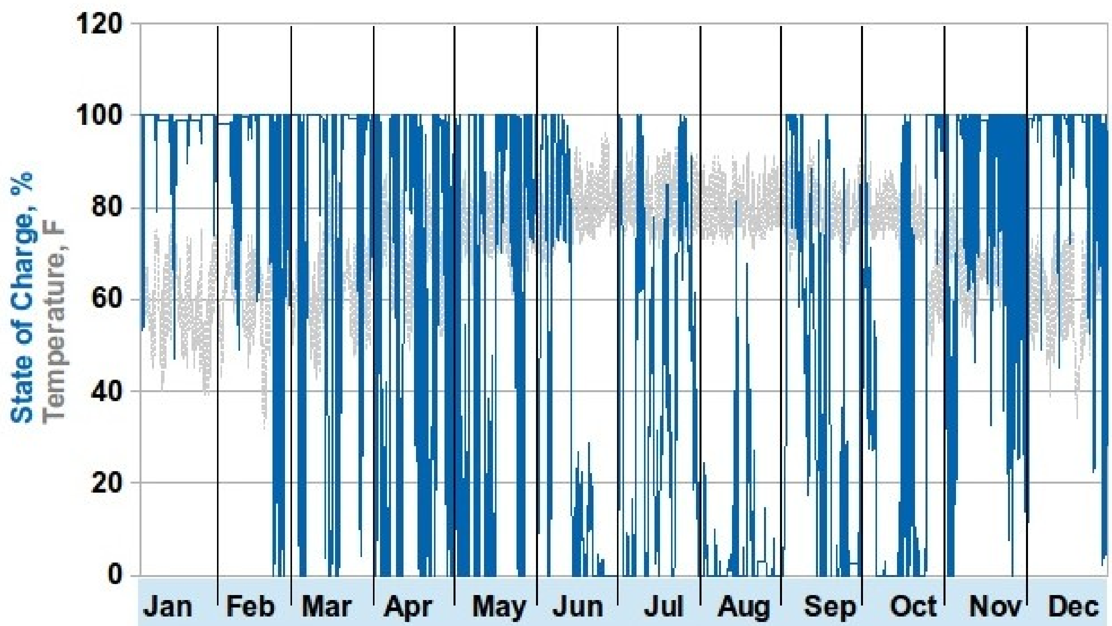

Figure 7-5. Orlando TMY State-of-Charge

During the less-warm months of January to May plus November and December, the thermal energy storage system state-of-charge cycles extensively from full charge (100) to various lower levels which occasionally reach full discharge (0).

Simulation results indicate the indirect/direct evaporative cooler with the thermal energy storage cooling experiences a shortfall of 27.07 million Btu (28,563 MJ). This represents a 28.3% improvement over the baseline cooler and a reduction of the shortfall time periods to 1288 hours. Thermal energy storage reduces water consumption to a total of 61,651 gallons which represents a 25.8% improvement. Table 7-2 compares results from simulations of the baseline model of the baseline cooler, the model with the capacity indicated above, the third version with double capacity and the fourth version with half capacity.

MODEL	Cooling Shortfall	Shortfall Time	Cooling Improvement	Water Use	Water Use Improvement
Baseline	37.76 MBtu (39,840 MJ)	2101 hrs	NA	83,066 gal. (314.4 m^3)	NA
Half Capacity	27.14 MBtu (28,632 MJ)	1401 hrs	28.1%	65,359 gal. (247.4 m^3)	21.3%
Full Capacity	27.07 MBtu (28,563 MJ)	1288 hrs	28.3%	61,651 gal. (233.4 m^3)	25.8%
Double Capacity	25.79 MBtu (27,207 MJ)	1197 hrs	31.7%	58,956 gal. (223.1 m^3)	29.0%

Table 7-2. Orlando Simulation Results Comparison

Evaporative cooling provides some benefit for data center cooling in the Orlando climate. Thermal energy storage added to the evaporative cooler reduces the cooling shortfall which must be met by other means. Thermal energy storage substantially reduces water consumption relative to the baseline indirect/direct evaporative cooler. The small differences in performance benefits between the half capacity and full capacity systems suggest a optimum capacity lies below the full capacity system.

7.5 Results for Dallas

Dallas, Texas experiences an inland hot, humid climate, less humid than a coastal climate with wider spreads of diurnal and annual temperatures. Evaporative cooling would be expected to provide more benefit than for the coastal climate. Figure 7-6 shows a plot of the hourly dry bulb ambient temperature for the Dallas dataset from The Meteorological Year. The winter temperatures are colder and summer temperatures are hot with a peak of 104°F (40°C). The diurnal temperature spread during the summer suggests the humidity may be high. The remainder of the year diurnal temperature spreads appear larger indicative of moderate humidity.

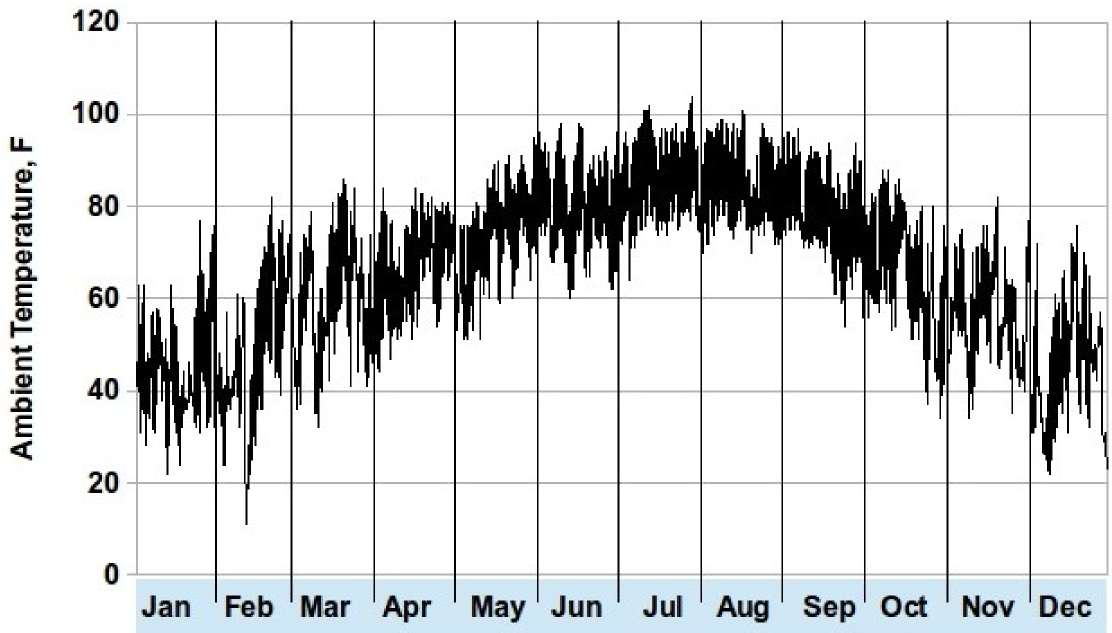

Figure 7-6. Dallas TMY Hourly Dry Bulb Temperature

Evaporative cooler alone does not provide the cooling supply air temperature needed for data center cooling. The Dallas simulation results find the baseline indirect/direct evaporative cooler requires additional cooling in the amount of 4.335 million Btu (4573 MJ) over time periods totaling 312 hours, substantially less than Orlando. Water consumption of the evaporative cooler gives an indication of how much cooling the system produces. Post-processing the simulation results finds the baseline cooler evaporated 104,635 gallons of water.

Some water is drained to control total dissolved solids and does not contribute to cooling. Drain water adds an additional 1815.5 gallons for a total water consumption of 106,451 gallons for the year.

Simulation of the indirect/direct evaporative cooler with the post-media interface concept compares with the baseline. Figure 7-7 presents the Dallas state-of-charge overlaying the ambient dry bulb temperature.

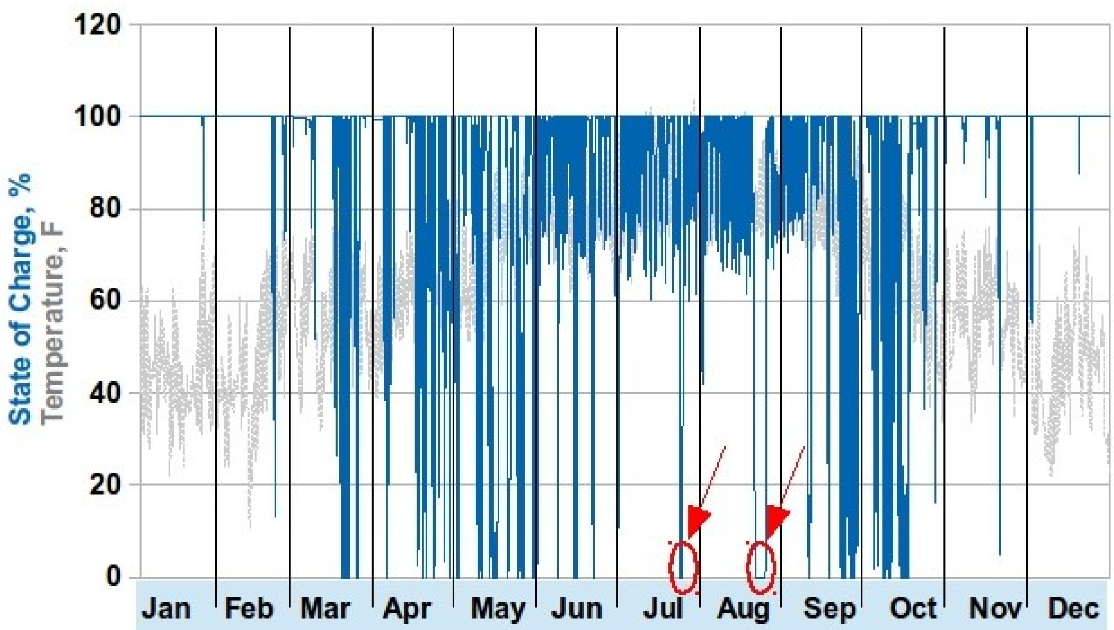

Figure 7-7. Dallas TMY State-of-Charge

During the colder months of January, November and December, the thermal energy storage system state-of-charge cycles little and normally carries a full charge (100), indicative of free-cooling. The thermal energy storage water use reduction mode cycles extensively from late February through most of June plus September and October. State-of-charge drops to zero many times. This indicates full utilization of the thermal energy storage to reduce water consumption.

The system remains in the hot environment improvement mode through July and August. When stored cooling drops to zero in this mode, the system fails to maintain the supply air temperature. The two red ovals indicate the state-of-charge going to zero when the system cannot maintain the cooling air supply temperature and the system experiences a cooling shortfall. The baseline cooler system experiences sustained shortfalls lasting 20, 31, and 63 hours at these times. The full-capacity and double-capacity thermal energy storage systems reduce but do not eliminate these shortfalls. A thermal energy storage system to eliminate these shortfalls would require much larger capacity.

Simulation results show the indirect/direct evaporative cooler with the 522,000 Btu (550.5 MJ) capacity thermal energy storage shortfall totals 2.173 million Btu (2292 MJ), representing a 49.9% improvement over the baseline cooler and a reduction of the shortfall time periods to 100 hours. Thermal energy storage reduces water consumption to a total of 88,708 gallons which represents a 16.7% improvement.

Table 7-3 compares results from simulations for the baseline cooler version, the full capacity version indicated above, and the versions with double capacity and half capacity.

Evaporative cooling provides a large benefit for data center cooling in the Dallas climate but it does not fulfill all cooling requirements. Extended periods of hot, humid weather exceeds the capability of evaporative cooling, requiring other means such as a direct expansion system to maintain the supply air temperature. This cooling shortfall being met by a direct expansion system is greatly reduced with the added thermal energy storage. Two sustained cooling shortfalls last beyond 30 hours. Comparison of results from the full and double capacity systems suggest only a very large, uneconomical thermal energy storage system might accommodate the

Dallas climate without direct expansion supplemental cooling.

MODEL	Cooling Shortfall	Shortfall Time	Cooling Improvement	Water Use	Water Use Improvement
Baseline	4.335 MBtu (4573 MJ)	312 hrs	NA	106,451 gal. (402.9 m^3)	NA
Half Capacity	2.462 MBtu (2598 MJ)	107 hrs	43.2%	92,480 gal. (350.0 m^3)	13.1%
Full Capacity	2.173 MBtu (2292MJ)	100 hrs	49.9%	88,708 gal. (335.8 m^3)	16.7%
Double Capacity	1.950 MBtu (2057 MJ)	90 hrs	55.0%	85,718 gal. (324.4 m^3)	19.5%

Table 7-3. Dallas Simulation Results Comparison

Thermal energy storage reduces total water consumption relative to the baseline indirect/direct evaporative cooler. The differences in performance benefits between the half capacity and double capacity systems suggest a optimum capacity depends on other factors such as cost of water and cost for the system to cover the cooling shortfall.

7.6 Results for Phoenix

Phoenix, Arizona experiences an high desert hot, dry climate. Figure 7-8 shows a plot of the hourly dry bulb ambient temperature for the Phoenix dataset from The Meteorological Year. The winter temperatures are mild and summer temperatures are hot with a peak of 112°F (44.4°C). Evaporative cooling would be expected to very effective, but cannot handle the highest peak temperatures.

The Phoenix simulation results show the baseline indirect/direct evaporative cooler requires additional cooling in the amount of 18,616 Btu (19.6 MJ) over widely separated time periods totaling 4 hours which is dramatically lower than Orlando and Dallas. Water consumption of the evaporative cooler gives an indication of how much cooling the system produces. The Phoenix result finds the baseline cooler evaporated 227,457 gallons of water. This represents the evaporated water. For control of total dissolved solids some water is drained without being evaporated. Drain water adds 2320 gallons for a total water consumption of 229,776 gallons for the year.

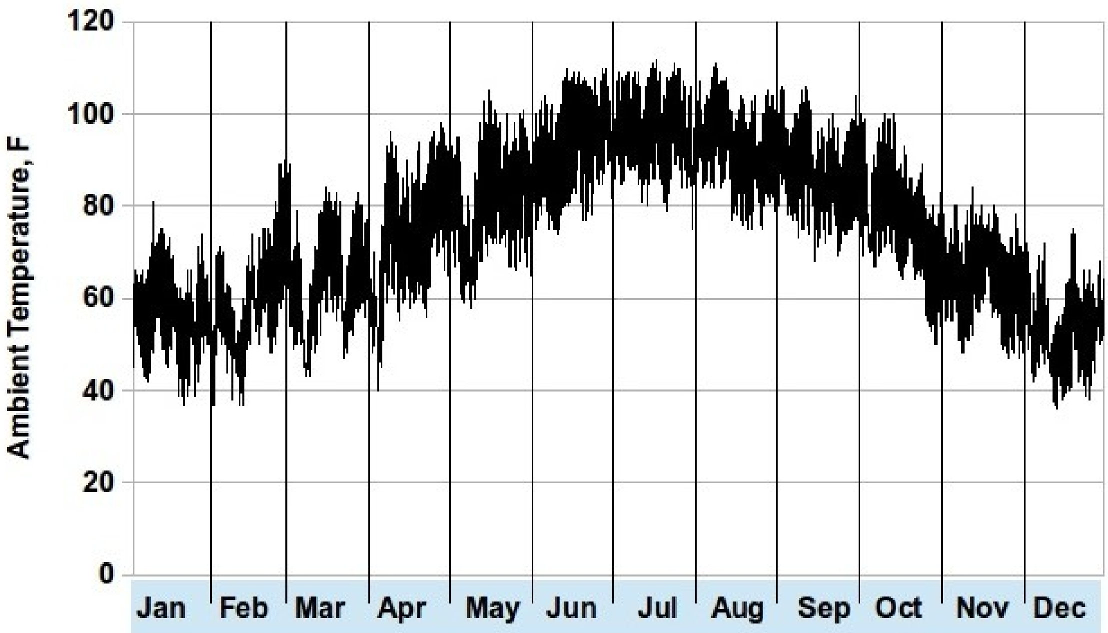

Figure 7-8. Phoenix TMY Hourly Dry Bulb Temperature

Simulation of the indirect/direct evaporative cooler with the post-media interface concept compares favorably with the baseline. Figure 7-9 presents the Phoenix state-of-charge overlaying the ambient dry bulb temperature. During the cooler months of January and December, the thermal energy storage system state-of-charge cycles little and normally carries a full charge (100). The water use reduction mode cycles extensively from late February through early May plus October and November. The system remains in the hot environment improvement mode from late May through September. The red oval indicates the state-of-charge remains above 60 during this hot environment improvement segment. The system maintains the cooling air supply temperature and there is no cooling shortfall.

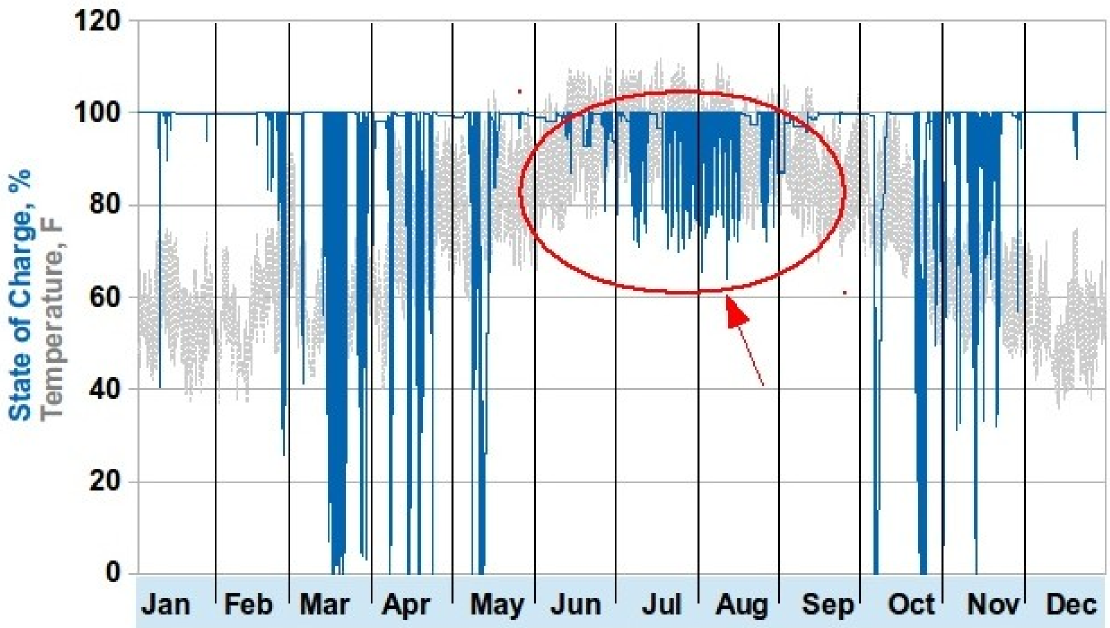

Figure 7-9. Phoenix TMY State-of-Charge

Simulation results indicate the indirect/direct evaporative cooler with the thermal energy storage cooling shortfall total drops to zero, completely eliminating the shortfall time periods. Thermal energy storage reduces water consumption 18,868 gallons which represents an 8.2% improvement. These results come from the second version of the model with the thermal energy storage system with the capacity of 522,000 Btu (550.5 MJ). Table 7-4 compares results from simulations of the first model version of the baseline cooler, the second model version with the capacity indicated above, the third version with double capacity and the fourth version with half capacity. In this case, the fourth version with half the thermal energy storage capacity of the baseline thermal energy storage system fully compensates for the small cooling shortfall of the basic indirect/direct evaporative cooler. This suggests an even smaller optimum capacity. A fifth version of the model reduced the thermal energy storage to one quarter of the baseline thermal energy storage, 130,500 Btu (137.7 MJ). This quarter-size system still eliminates the cooling shortfall and reduces the water use by 13,377 gallons (51.5 m^3) which is 5.8%.

MODEL	Cooling Shortfall	Shortfall Time	Cooling Improvement	Water Use	Water Use Improvement
Baseline	15.88 MBtu (16,753 MJ)	4 hrs	NA	229,776 gal. (884.6 m^3)	NA
Quarter Capacity	0.0 MBtu (0.0 MJ)	0 hrs	100 %	216,400 gal. (833.1 m^3)	5.8%
Half Capacity	0.0 MBtu (0.0 MJ)	0 hrs	100 %	213,669 gal. (822.6 m^3)	7.0%
Full Capacity	0.0 MBtu (0.0 MJ)	0 hrs	100 %	210,909 gal. (812.0 m^3)	8.2%
Double Capacity	0.0 MBtu (0.0 MJ)	0 hrs	100 %	209,495 gal. (806.6 m^3)	8.8%

Table 7-4. Phoenix Simulation Results Comparison

Basic evaporative cooling works very well for data center cooling in the Phoenix climate but still experiences a cooling shortfall. Thermal energy storage fulfills the requirement for extra cooling. Thermal energy storage still manages to reduce water consumption relative to the baseline indirect/direct evaporative cooler. The differences in performance benefits between the half capacity and quarter capacity systems suggest a optimum capacity falls near this range. Additional simulations find the minimum thermal storage capacity of only 96,000 Btu (101.3 MJ) eliminates the cooling shortfall.

CHAPTER 8

COST COMPARISON

Comparison of costs considers operating costs and hardware costs. The following costs are not included in this comparison. Non-recurring costs such as design effort, controls and software development are difficult to estimate at a conceptual stage and end up amortized over a production run. Installation costs depend several variables such as the design details and installation site. Costs of on-going maintenance and service depend on the design and component reliability.

8.1 Operating Costs

In this study, operating costs include water consumption and electrical power demand. Water consumption ties directly to the evaporated water and the drained water to control total dissolved solids. Electrical power demand includes two parts. When a cooling shortfall occurs where the thermal energy storage system lacks capacity to help the cooler maintain the supply air temperature, this study assumes the shortfall is overcome with a direct expansion system operating with a coefficient of performance of 3.5. The cooling demand of the shortfall and the coefficient of performance allow calculation of the electrical energy demand for a direct expansion system to handle the shortfall and maintain the supply air temperature. The second part of the electrical demand comes from the pump used to move the phase change material slurry when the thermal energy storage system is active. Flow of the slurry must be regulated to obtain the best performance. The electrical demand for the pump depends on the flow rate and pressure drop. Pressure drop is assumed to be 25 psi (172.4 kPa) at full flow and proportional to the square of the flow rate. Electrical demand calculations assume an overall electrical efficiency of 40% for the variable speed pump. The low electrical efficiency of 40% comes from the expectation that speed regulation incurs significant losses, particularly at low flows.

Utility cost data come from several sources. The Orlando Utilities Commission [33] publishes water and electrical power rates. Their rate for commercial water is $1.541 per thousand gallons. They offer a rate for electrical power at $0.06482 per kilowatt-hour. The city of Dallas publishes a water rate [34] of $3.71 per thousand gallons. Dallas enjoys a competitive market for electrical power, creating a challenge to identify a representative rate. A dissertation by Marianna Vallejo [35] includes some representative water and electrical utility rates. The dissertation lists an electrical power rate of $0.088 per kilowatt-hour for the city of Hillsboro which is located near Dallas. The dissertation also lists utility rates for Phoenix as $0.11 per kilowatt-hour of electrical power and $5.00 per thousand gallons of water.

MODEL	Water Use	Water Cost	DX Power kwh	Pump Power kwh	Electric Cost	Utility Costs W + E
Baseline	83,066 gal. (314.4 m^3)	$128	38,722	0	$2510	$2638
Half Capacity	65,359 gal. (247.4 m^3)	$101	29,829	124	$1942	$2043
Full Capacity	61,651 gal. (233.4 m^3)	$95	27,831	147	$1814	$1909
Double Capacity	58,956 gal. (223.1 m^3)	$91	26,446	162	$1725	$1816

Table 8-1. Orlando Operating Costs

Table 8-1 provides the Orlando individual costs for water and electrical power and the combined total for an indirect/direct evaporative cooler with for a data center with a 67.7

kilowatt total heat load. These costs do not include the electrical costs for the basic cooler as these costs should not change substantially with the addition of the thermal energy storage systems with varying capacities. The electrical power cost for the data center is not included. Water cost ranges from a high of $128 per year for the baseline cooler to a low of $91 per year for the double capacity thermal energy storage system. Electrical power cost ranges from a high of $2510 per year to a low of $1725 per year for the double capacity system. The double capacity system reduces utility costs by $822 annually and the half capacity system offers a annual savings of $595.

MODEL	Water Use	Water Cost	DX Power kwh	Pump Power kwh	Electric Cost	Utility Costs W + E
Baseline	106,451 gal. ($402.9 m^3$)	$395	4445	0	$391	$786
Half Capacity	92,480 gal. ($350.0 m^3$)	$343	2529	100	$231	$574
Full Capacity	88,708 gal. ($335.8 m^3$)	$329	2232	122	$207	$536
Double Capacity	85,718 gal. ($324.4 m^3$)	$318	2003	141	$189	$507

Table 8-2. Dallas Operating Costs

Table 8-2 provides the Dallas individual costs for water and electrical power as for Orlando. Water cost ranges from a high of $395 per year for the baseline cooler to $318 per year for the double capacity thermal energy storage system. Electrical power cost ranges from a high of $391 per year to a low of $189 per year for the double capacity system. The full capacity system reduces utility costs by $250 annually.

MODEL	Water Use	Water Cost	DX Power kwh	Pump Power kwh	Electric Cost	Utility Costs
Baseline	229,776 gal. ($869.9\ m^3$)	$1149	19.09	0	$2.1	$1151
Reduced Capacity	217,363 gal. ($822.9\ m^3$)	$1087	0	142	$15.7	$1102
Quarter Capacity	216,400 gal. ($819.3\ m^3$)	$1082	0	143	$15.7	$1098
Half Capacity	213,669 gal. ($808.9\ m^3$)	$1068	0	126	$13.9	$1082
Full Capacity	210,909 gal. ($798.5\ m^3$)	$1055	0	131	$14.4	$1069
Double Capacity	209,495 gal. ($793.1\ m^3$)	$1047	0	123	$13.5	$1061

Table 8-3. Phoenix Operating Costs

Table 8-3 provides the Phoenix individual costs for water and electrical power as for Orlando and Dallas. Water cost ranges from a high of $1149 per year for the baseline cooler to $1047 per year for the double capacity thermal energy storage system. Electrical power costs are very different with the baseline system having the minimum. The reduced capacity system has the peak cost and the costs decrease with increasing size of the thermal energy storage. Thermal energy storage allows trading electrical power for reduced water consumption. The quarter capacity system reduces total utility costs by $153 annually. Any of the thermal energy storage systems can eliminate the need for a direct expansion system to handle cooling shortfalls.

8.2 Hardware Costs

Design changes to the baseline cooler to accommodate thermal energy storage modifications include hardware of the thermal energy storage (phase change material slurry, storage tank, heat exchangers, plumbing, valves, actuators, wiring, sensors, switches, indicators, support structure, miscellaneous). The variations of the thermal energy storage systems extend only to the tank and amount of phase change material slurry. The other hardware costs do not change appreciably. Luttrell et.al. [30] estimated the costs for pump, heat exchanger, and miscellaneous components as shown in Table 8-4 with a total cost of $1178.

COMPONENTS	ESTIMATED COST
Pump	$435
Heat Exchanger	$440
Miscellaneous	$303
TOTAL	$1178

Table 8-4. Thermal Energy Storage System - Constant Cost

Costs of five different size versions of the thermal energy storage are shown Table 8-5. Tank costs are estimated by applying the required size to a cost versus size trend from a commercial source. Slurry cost is expected to be $300 per barrel which contains 42 gallons, about 337 pounds (153 kg).

VERSION	THERMAL STORAGE	TANK COST	SLURRY COST	TOTAL COST (inc $1178)
Reduced Size	96,000 Btu (101.3 MJ)	$370	$2760	$4308
Quarter-size	130,500 Btu (137.7 MJ)	$500	$3750	$5428
Half-size	261,000 Btu (275.4 MJ)	$685	$7500	$9363
Full-size	522,000 Btu (550.7 MJ)	$1048	$15,000	$17,226
Double-size	1,044,000 Btu (1101 MJ)	$1775	$30,000	$32,953

Table 8-5. Thermal Energy Storage Components and Total Cost

8.3 Results Normalized to One Megawatt Data Center

Results for operating costs and hardware costs are linearly scaled up for a one megawatt data center. Table 8-6 provides linearly scaled hardware costs and annual operating cost savings for a data center with a one megawatt cooling load.

LOCATION	HARDWARE COST	ANNUAL OPERATING COST SAVINGS
Orlando (full size)	$254,446	$10,768
Dallas (full size)	$254,446	$3693
Phoenix (reduced size)	$63,634	$724

Table 8-6. One Megawatt Data Center Costs

CHAPTER 9

CONCLUSIONS

This study investigates augmentation of indirect/direct evaporative coolers with thermal energy storage toward the goals of extending hot environment operation and reducing water consumption. An important part of this study involves development of a simulation of the indirect/direct evaporative cooler. The simulation model results compare favorably with performance values published in the Mestex Technical Guide for their Aztec coolers [36].

Six concepts integrated the thermal energy storage at different locations in the baseline cooler. Subjective evaluation of the six concepts' integration complexity favors two of the concepts. Qualitative evaluations of six systems' performance for the stated goals again favored the same two concepts.

The two concepts integrate a heat exchanger for the thermal energy storage system either after the evaporative media or after the cooling air supply fan. While ranking almost equal, the post-media concept which integrates the heat exchanger after the evaporative media has an optimal phase change temperature of 72°F (22.2°C) which is near the value for a commercially available product so this concept was selected for detailed evaluation.

Simulation results for hotter environments indicate the post-media interface concept extends the upper limit for ambient temperature. Since results are integrated over diurnal cycles the upper temperature limit depends on the conditions preceding. One artificial diurnal cycle sustained acceptable cooling with an increase over 7°F (3.9°C) above the dry bulb temperature limit with a constant absolute humidity of 0.0142 pounds of water per pound of dry air (0.0142 kg of water per kg of dry air). The system could extend operation to higher humidity, allowing an increase to an absolute humidity of 0.0154 pounds of water per pound of dry air (0.0154 kg of water per kg of dry air).

Use of Meteorological Year data sets provide a general indication of the capabilities of thermal energy storage. These datasets do not provide representation of more extreme conditions commonly used for the design of systems for specific locations. Consideration of a thermal energy storage system must include potential extreme conditions at the specific location.

Simulations using The Meteorological Year for Orlando, Dallas, and Phoenix show interesting results. Evaporative systems' ability to meet data center cooling loads vary significantly with ambient humidity. A baseline indirect/direct evaporative cooler in Orlando does provide useful cooling but requires additional cooling, such as from a direct expansion system, for more than 2000 hours during the year which is almost 25% of the time. Thermal energy storage reduces that time where additional cooling is needed to 1197 hours and also reduces water consumption by 29%. A baseline indirect/direct evaporative cooler in Dallas provides useful cooling but also requires additional cooling, such as a direct expansion system, but for only 312 hours during the year. Thermal energy storage can reduce that time for additional cooling to 90 hours and also reduce water consumption by 19.5%. A baseline indirect/direct evaporative cooler in Phoenix works well as a consequence of the lower humidity. It still requires additional cooling, such as a direct expansion system, but for only 4 hours during the year. Thermal energy storage completely eliminates that need for additional cooling and reduces water consumption at least 5.8 %.

Cost of the system, that is capital expense, appears high when compared to just the reduction in operating expense. A system installed at Orlando could have hardware costs of $5400 to almost $33,000 while producing operating cost savings of $595 to $822 per year. Due to cooling shortfall's, a supplemental cooling system like direct expansion is required. A system installed at Dallas has the same range of hardware costs while producing savings around $250 annually. Like Orlando, a system in Dallas experiences cooling shortfall's therefore a supplemental cooling system like direct expansion is required.

The effectiveness of evaporative cooling in Phoenix permits a relatively small thermal energy storage system to overcome the cooling shortfall conditions. It has a hardware cost around $4300 while producing operating cost savings of $150 per year. This system obviates the need for supplemental cooling from a direct expansion system at a capital cost similar to the direct expansion system plus it generates annual cost savings on utilities.

The decision to add thermal energy storage to an indirect/direct evaporative cooler depends on the value of extended hot environment operation and possible elimination of a direct expansion system. Follow-on work will consider the Total Cost of Ownership (TCO) of an evaporative cooler with thermal energy storage, possible size reduction or elimination of an auxiliary cooling system, climate extremes, and optimized controls.

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APPENDIX A

EVAPORATIVE COOLER PARAMETERS

Several component performance characteristics affect the performance of the indirect/direct evaporative cooler. These performance parameters include: evaporative media effectiveness and pressure loss, heat exchanger effectiveness and pressure loss, data center pressure loss, and fan efficiency. For this study, guidance was taken from the Mestex, Technical Guide [36] for an ASC-25 unit operating at 10,000 CFM but values may be adjusted to produce the expected air-water mixture properties indicated in the Technical Guide.

The Technical Guide indicates the direct evaporative media produces an 88% saturation efficiency and the indirect media produces a 75% saturation efficiency. The guide does not specify definitions of these efficiencies. For this study the saturation efficiency applies to the difference between the saturation partial pressure of water vapor and the inlet condition partial pressure of water vapor. The fractional saturation efficiency times the difference of those two partial pressures results in the actual change for the partial pressure of water vapor.

The direct evaporative media pressure loss is 0.14 inches (0.356 cm) of water in the technical guide. The indirect media pressure loss is not given and will vary with cooling tower air flow. The simulation uses constant values of 0.14 inches (0.356 cm) of water pressure loss for both the direct and indirect evaporative media.

The direct side heat exchanger characteristics also come from the technical guide. Pressure loss is 0.51 inches (1.3 cm) of water in the technical guide. The simulation uses 0.51 inches (1.3 cm) of water pressure loss for this heat exchanger and the thermal energy storage heat exchanger in the air flow. The concept with a thermal energy storage heat exchanger in the water sump flow stream does not account for pressure loss. Effectiveness of the direct side heat exchanger is not provided in the Mestex guide. The simulation uses a fairly high effectiveness of 90% which is necessary for good cooling performance. Effectiveness times the difference of the inlet temperatures results provides the temperature change for the flow with the minimum heat capacity rate.

The simulation allocates a total pressure drop of 2.5 inches (6.35 cm) of water from the exit of the evaporative cooler to the air outlet of the data center.

Fan efficiency affects the cooling supply air temperature because the electrical losses transfer to the air. Instead of a fan efficiency, the fan effect on the cooling air supply is calculated as the temperature rise accompanying an isentropic compression plus 1°F (0.56°C). This approaches the performance of the Mextex technical guide.

BIOGRAPHICAL INFORMATION

Jeff Luttrell received his baccalaureate (B.S.) Mechanical Engineering degree in 1977 from Texas Tech University. He received his master's (M.S.) Mechanical Engineering degree in 1982, also from Texas Tech University.

Mr. Luttrell began work in the aerospace industry in January 1979. He progressed to positions of thermodynamics or heat transfer lead engineer on major US military aircraft programs including the F-22 engineering and manufacturing development, C-130 re-engining and reliability improvement, and A-12 full scale engineering development. He retired from Lockheed Martin in 2013 after almost 30 years of service.

Mr. Luttrell began studies toward a doctorate degree in August 2007 at the University of Texas-Arlington with particular interest in cooling systems and energy conservation. In the fall of 2013 he was designated the project lead on a study looking to apply phase change materials to evaporative data center cooling systems. This topic included his to areas of particular interest and became the subject of his dissertation. His innovative ideas which integrate thermal energy storage with evaporative coolers are patent pending. Nine years after beginning, Jeff received his mechanical engineering doctorate degree in August 2016.