ENERGY AND WATER IMPACTS OF DATA CENTER COOLING SYSTEMS: A TRIPLE BOTTOM LINE ASSESSMENT FOR FACILITY DESIGN

by

MARIANNA VALLEJO

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[Kids] don't remember what you try to teach them. They remember what you are.

-Jim Henson, *It’s Not Easy Being Green: And Other Things to Consider*

First and fore most I would like to acknowledge my family for never telling me that I couldn’t do something. I would not be who I am today without your love and support. I would like to thank my grandmother Katheryn for teaching me to see the world in color and to think outside of the box. I would like to thank my grandfather Thomas Sr. for teaching me always to ask why. Most importantly, I would like to thank my mother Suzanna for being by my side every day and for always putting my education first.

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ABSTRACT

ENERGY AND WATER IMPACTS OF DATA CENTER COOLING SYSTEMS: A TRIPLE BOTTOM LINE ASSESSMENT FOR FACILITY DESIGN

Marianna Louisa Vallejo, Ph.D.

The University of Texas at Arlington, 2015

Supervising Professor: Dereje Agonafer

In order for a data center to function properly, the environment must be tightly controlled to ensure maximum reliability of the electronic hardware components. Any method of controlling the environment has a cost, either in capital or in resources, and therefore becomes an issue for the sustainability of a building or complex. Over the past decade, data center facilities have been designed around the energy consumption with a particular focus on the cooling infrastructure. As a means of reducing power consumption, air- and water-side economizers have been widely adopted within the cooling loop. Current research and a few facility owners have focused on eliminating the chiller plant altogether by implementing dedicated chiller-less cooling solutions. Although much work has been published regarding the energy savings benefits of shifting away from the traditional chiller plant facility, very little work has considered the water cost of these solutions.

This work seeks to address this gap through a triple bottom line assessment of facility cooling systems. An experimental study is used to generate representative inputs to a thermodynamic model of the facility. This model considers cooling power, water consumption, and calculated annual operational costs of the various cooling technologies available. This constitutes the economic bottom line of the facility. Analysis across geographic regions is considered for climatic differences that affect the cooling power and water consumption. This allows for more robust life cycle impact assessment of the facility water consumption, which establishes the environmental bottom line. The water stress indicator utilized during impact assessment allows water scarcity to be considered for the societal impact of the facility. Today the focus of most corporate sustainability policies and the majority of legislative policies consider carbon footprint and/or energy efficiency of a facility. However, the discussion is beginning to include the issue of water scarcity and the associated risks. Here the impact of corporate and legislative water resource policies on facility planning and commissioning is considered.

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CHAPTER 1

INTRODUCTION

A data center facility is one that has the primary functions of housing the electronic hardware used for data processing, storage, and transmission and maintaining constant, instant access to said data [1]. This access has become entrenched in the functioning of society at large and in particular the daily operations of all sectors of the economy. In order for a data center to function properly, the environment must be tightly controlled to ensure maximum reliability of the electronic hardware components. Any method of controlling the environment has a cost, either in capital or in resources, and therefore becomes an issue for the sustainability of a building or complex. Up to the mid-1990s, the total cost of ownership (TCO) of a data center was driven by the capital expenses associated with construction of new facilities and the cost of new IT hardware purchases. The operating expense and the environmental impact of providing constant access were only marginally considered. This business model has since shifted, largely due to awareness of the energy demands of the hardware, a significant increase in the operating cost of support infrastructure, escalating energy prices, and a growing concern for resource availability. Resulting trends and best practices have focused on micro-level improvements in energy efficiency within the facility rather than holistic approaches to addressing the concerns. In recent years, the coupled nature of energy production and water consumption has become a focal point for legislation and national research priorities [2]. As drought conditions continue to spread across the country and energy demand continues to rise, water availability and access rights are coming to the forefront of societal concerns. The aim of this research is to develop a holistic approach to account for the true sustainability of a facility's cooling system.

1.1 Background

During the late 1990s to early 2000s, the Federal Administration took notice of the ever-growing demand placed on available power infrastructure and natural resources. The 2001 National Energy Policy Report [3] estimated that if growing population and economic trends continued, the demand for power would necessitate an additional 393 GW of generating capacity by 2020. This would require adding 1,300 – 1,900 new plants to the existing power grid. In 2006 in response to the rapid growth of data center facilities and the disproportionate power consumption attributed to these buildings, Congress enacted Public Law 109-431 which called for the Environmental Protection Agency (EPA) to produce an analysis of said trends, the potential cost savings of adopting energy efficient hardware, the potential impacts of implementing such hardware, and recommendations for best practices in the operation of data centers [1]. In 2012, the Department of Energy (DOE) identified the Water-Energy Nexus as a key area of research and development that will be critical to the future of power production and national security [4].

1.1.1 Building Energy Demands

In 2010, the United States (US) consumed 97.8 quads of energy or 19% of global consumption. US energy consumption is categorized by sector as buildings, industry, and transportation. The buildings sector is further split into residential and commercial based on primary end-use and represents the largest consumer of the three [5]. This sector represented approximately 41% of consumption (7% globally) in 2010 [5] and an estimated 37% in 2014 [6]. This is double the consumption reported for 1980. As such, research in efficiency improvements has long been a priority for the DOE. Heating, cooling, and lighting represent over half of commercial building consumption, as seen in Figure 1.1 [5].

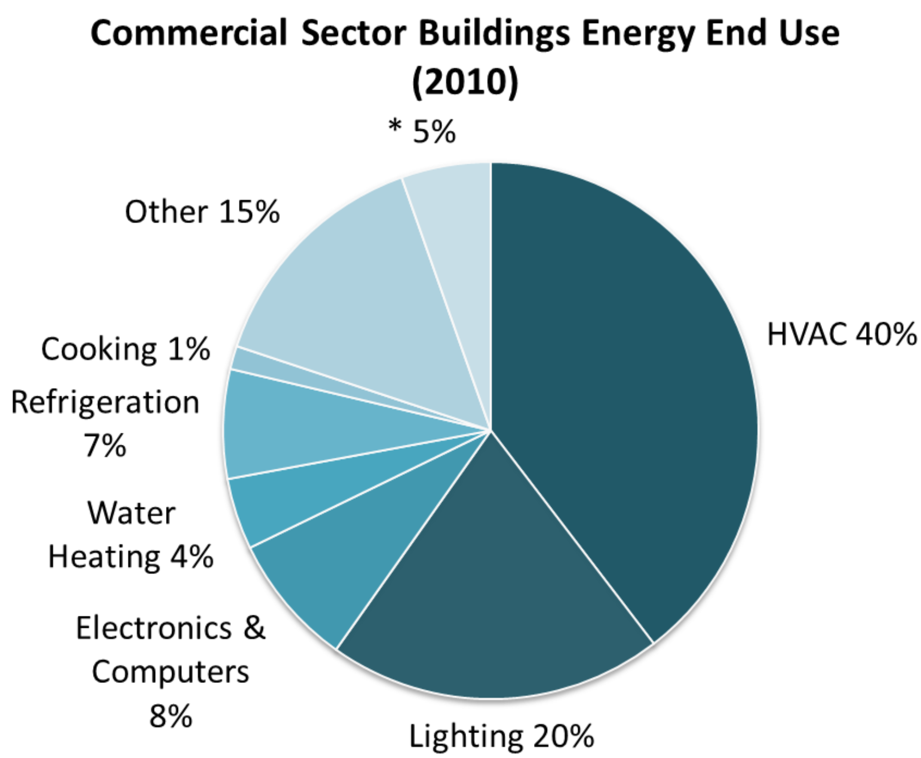


Figure 1.1 Energy end use in commercial sector.

1.2 Data Center Trends in Energy Use

Data center facilities are classified as commercial buildings. However, the energy consumption is disproportionately high (up to 40x [7]) as compared to office buildings or retail space. The magnitude of consumption and distribution within the facility is much closer to that of an industrial complex [1]. Data centers consumed an estimated 61 billion kilowatt-hours in 2006 and if then-current efficiency trends continued, this would double every five years. In response to PL 109-431, EPA called for a 25% reduction in the power consumption by 2017 through the adoption of recommendations laid out in their report to Congress [1].

In 2007, the Green Grid (an open consortium formed to improve IT and data center resource efficiency) introduced the first of their utilization effectiveness metrics. Commonly referred to as PUE, the power usage effectiveness is a metric that quantifies the energy efficiency of a data center facility as the useful work produced by the facility (IT compute power) with respect to the total energy consumed by the facility as given by (1-1) [8].

$$\text{PUE} = \frac{Total\ Facility\ Power}{Compute\ Power} \tag{1-1}$$

Over the past ten years, this metric has developed into an unofficial standard by which facilities compare against each other. EPA and DOE consider the average PUE to be 1.8. An annual survey reported by Digital Realty shows that this assumption is incorrect, with the average PUE being 2.9. Figure 1.2 shows the distribution of reported PUE values by survey responders in 2013.

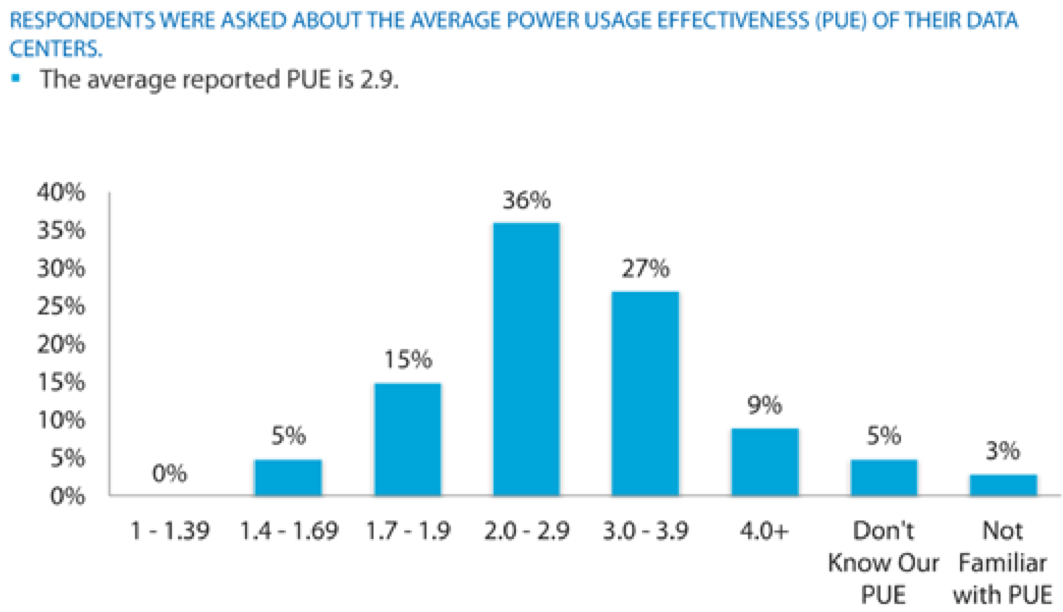


Figure 1.2: Average PUE distribution as reported by Digital Realty [9].

1.2.1 Facility Infrastructure

Data centers are traditionally housed within purpose-built buildings with an expected lifespan of 25 years. The infrastructure is initially provisioned based upon the maximum expected IT load capacity at the time of commissioning. However, the typical lifespan of the IT hardware is 10 years or less. As compute power per unit area has increased in recent years, facilities either cannot utilize the full space capacity due to insufficient power provisioning or cannot utilize the full compute power due to insufficient cooling capacity [10]. Some owners and equipment manufacturers have responded by moving to a modularized solution for building out IT capacity, much like the just-in-time (JIT) philosophy associated with lean manufacturing processes [11].

The concept was introduced as a deployment of ISO standard shipping containers that could be dropped anywhere that had sufficient power and water hook-ups. The market has shifted to deploying utility-specific “pods” within a traditional warehouse frame. A recent market report predicts that the modular market will grow at a compound annual rate of 37% [12]. However, IDC’s annual report predicts that enterprise businesses will take advantage of colocation and cloud services in order to lighten their capital burden and that over half of the market will shift to third-party infrastructure providers by 2017 [10].

The infrastructure of a facility, from the IT equipment deployed up to the building structure itself, is generally dictated by the role of the data center within the larger business model of a company. It is widely recognized that enterprise facilities (IT is not the primary focus of the business but directly influences day-to-day operations) represent the bulk of the industry footprint. Volume servers, systems designed to be utilized for any and all applications, prioritize reliability over efficiency. Koomey [13] showed that volume servers represented 90% of IT deployments in the US and about the same electricity used in 2005. In 2014, the National Resources Defense Council (NRDC) [14] published a report that related facility purpose to energy efficiencies. The distribution of data center types can be seen in Figure 1.3.

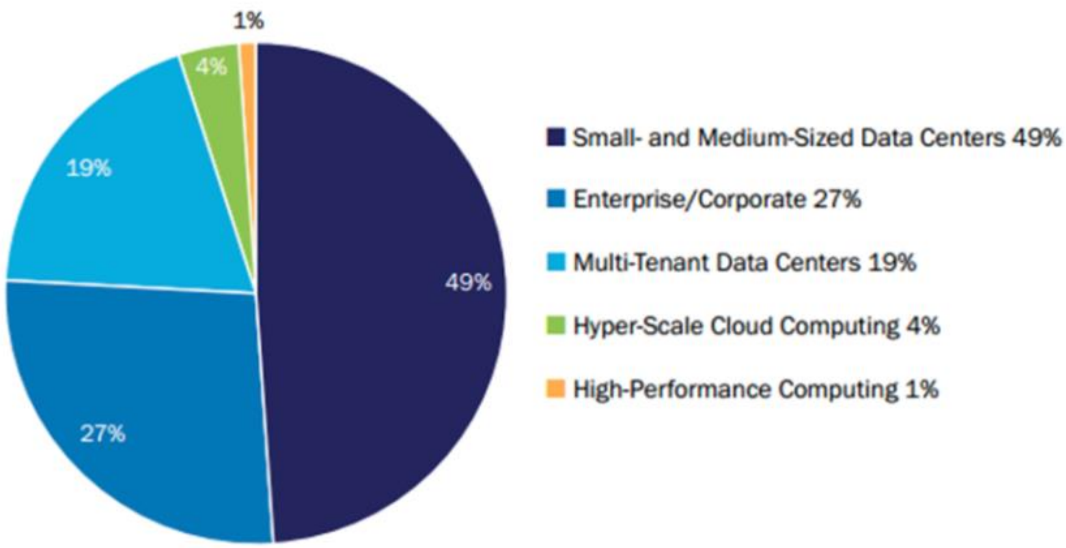


Figure 1.3: Facility electricity consumption by purpose.

1.2.2 Facility Cooling

Traditionally, facility cooling has relied on a centralized chilled water plant, computer room air handlers (CRAH), and cooling towers to provide sufficient environmental management to the IT equipment. Operators rely on the temperature set-point of the CRAH to control the room air conditions, which was previously maintained around 50°F supply air temperature to ensure hardware reliability. Although facility managers have been reluctant to change this criteria, most facilities currently maintain between 65-68°F supply air temperature [15]. As will be discussed in section 1.2.2.1, there is still significant room for this temperature to increase. As can be seen in Figure 1.4, the energy consumption is dominated by the compressor work required by the chiller plant.

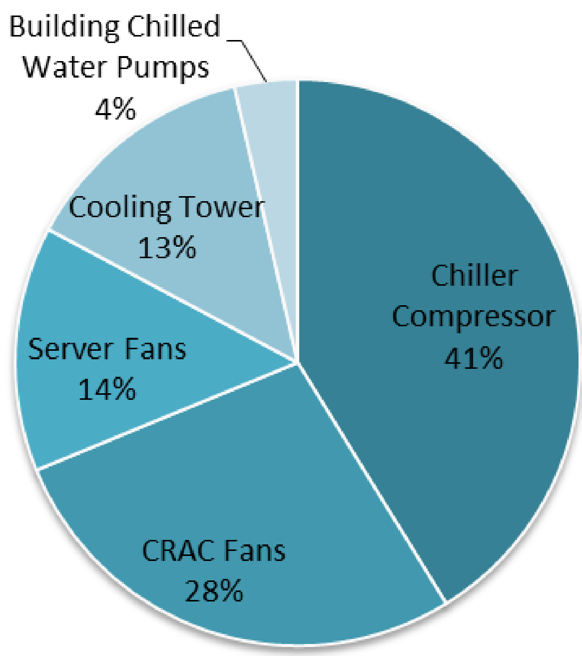


Figure 1.4: Breakdown of chilled water cooling system energy consumption.

In the decade since EPA's report to Congress, many technological improvements and best practices have been researched and adopted in order to improve the overall energy efficiency of the facility. Figure 1.5 [16] shows 13 common cooling system configurations currently utilized by data center facilities. The proposed study focuses on three of these: the traditional chiller plant, the direct outside air evaporative cooler, and the indirect air evaporative cooler. For facilities which employee an air-cooled computer room air conditioner (CRAC), an optional economizer mode may be added on to allow for direct outside air cooling under the right ambient conditions. Due in large part to the

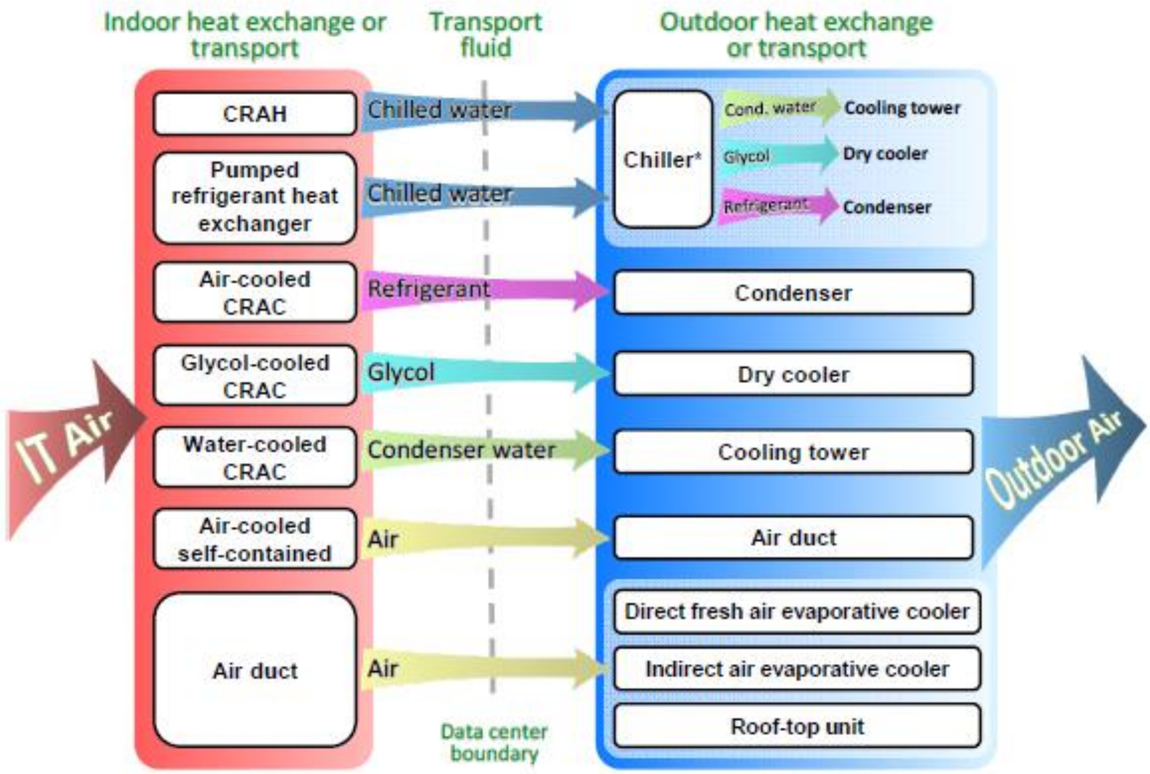


Figure 1.5: Cooling system configurations employed by data centers.

potential for energy savings, ASHRAE Standard 90.1 (2013) (this standard covers the energy efficiency of commercial building HVAC systems) added the prescriptive requirement for all new buildings, including data centers, to utilize air-side economizers. In the last five years, facilities have also begun to employee evaporative cooling systems in place of vapor-compression chillers, although this has not been widely implemented [15].

Significant work has been done to improve server efficiency in areas such as board layout design, efficient airflow management, efficient server fans, separation of compute and storage hardware, and liquid cooled solutions. Other improvements that have been implemented include the modular deployment, in-row cooling solutions, use of computational fluid dynamic (cfd) modeling software for improved airflow management within the data hall and infrastructure provisioning, and efficiency improvements to the cooling infrastructure components. One of the most critical improvements has been the separation of supply and exhaust air streams; first by adopting hot-aisle cold-aisle layouts in the data hall and then through containment of one air stream or the other. This work will focus on two recommendations to understand the potential bottom-line impact of adoption: the expanded operational envelope for data hall supply air and the use of chiller-less cooling systems across the geographic climate zones.

1.2.2.1 ASHRAE thermal guidelines and climate zones

The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) formed Technical Committee 9.9 Mission Critical Facilities, Technology Spaces and Electronic Equipment (TC 9.9) in order to address a need for better communication and understanding between equipment manufacturers and facility operators [17]. In 2004, the committee published its first recommended Thermal Guidelines for Data Processing Environments in which an envelope of operational environment conditions for IT equipment was called out. This envelope represents the range conditions of the air supplied to the inlet of the equipment that would not impact reliable operations. The dry-bulb temperature (this is the common temperature used in everyday life) range is 65° - 80°F. In 2008, the committee updated the guidelines by adding three classes of allowable conditions for those operators who sought to lower cooling costs by increasing the CRAH set-points. The committee expanded all of the envelopes and added a fourth allowable class in the 2011 update, which are shown in

Figure 1.6 as overlays within a standard psychometric chart. An update was published in the late part of 2014. However, this work will consider the A4 class operating envelope as represented in the 2011 update.

The expansion of the allowable classes has been in response to system designers and operators who are pushing the thermal limits for increased efficiency as well as the growing percentage of operators who would like to or are being required to expand the number of days that the air-side economizer can be utilized. The ASHRAE geographic climate zones, shown in Figure 1.7, consider both the outside temperature and relative humidity conditions and are utilized when determining appropriate cooling equipment for all applications.

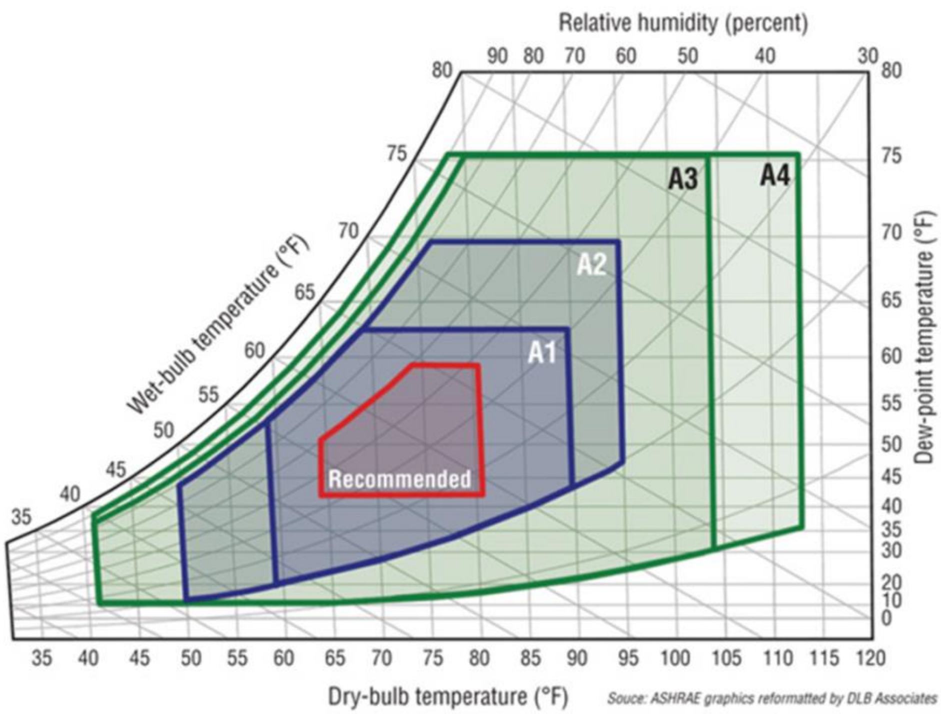


Figure 1.6: 2011 Thermal Guidelines as published by ASHRAE TC 9.9

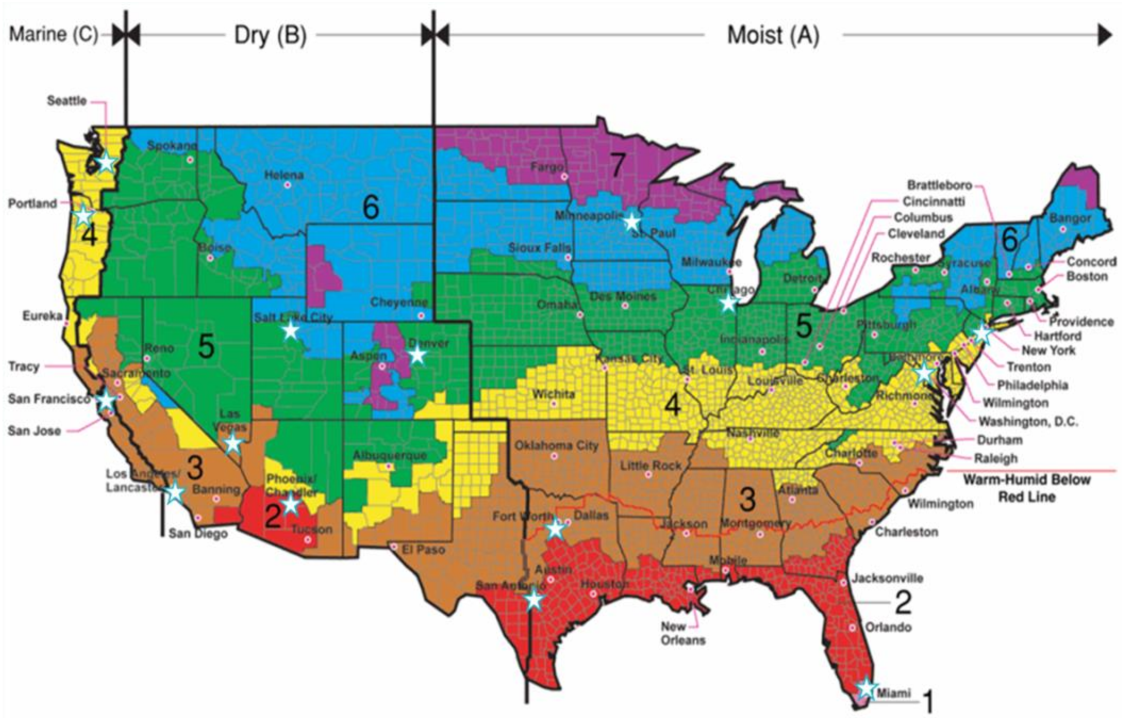


Figure 1.7: ASHRAE geographic climate zones.

1.3 The Water-Energy Nexus

Energy production is intrinsically dependent upon a reliable, abundant, and predictable water source. Water processing and distribution rely heavily upon energy sources. This work concentrates on electricity production and specifically thermoelectric power production, which represents 87% (3,547 BkWh) of electricity generated in 2014, as shown in Figure 1.8 [18]. Thermoelectric power represented 47% (161,000 Mg/d) of the water withdrawals in the US in 2010, which equates to 19 gal/kWh consumed water during the generation process [19]. Thermoelectric power withdrawals have surpassed agricultural withdrawals for the first time in the US. In response to these trends and growing awareness of potential negative impacts, DOE coined the term Water-Energy Nexus and highlighted key areas of R&D. One of the 'strategic pillars' identified is the optimization of freshwater efficiency of energy production, electricity generation, and end use systems [4].

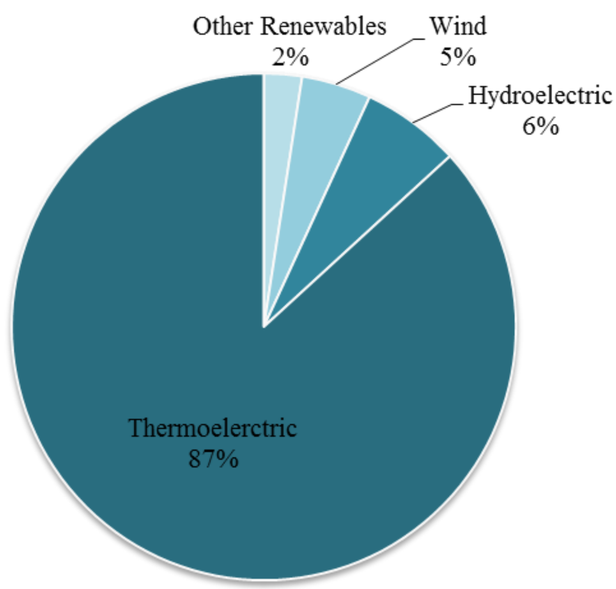


Figure 1.8: 2014 Electricity net generation by source.

In addition to the coupled nature of power production and water consumption, drought conditions and population growth have led to significant shortages in the available water supply in much of the US. This has led to a 41% increase in municipal water rates since 2010 across 30 major US cities [20]. Currently the cost of water remains a fraction of the energy cost for the end user. However, this is expected to change as water resources continue to become scarcer in key areas such as California, Texas, and New York/New Jersey and the available distribution infrastructure has to be replaced. Corporate policy shifts and legislation have led businesses to include water cost and availability in the list of considerations for decision-making. However, there is a lack in metrics or standards by which to measure and report sustainability impacts of water utilization.

1.4 Thesis Layout

This thesis is composed of nine chapters. Chapters 2 and 3 provide a background of work that previously done with regard to thermodynamic cost modeling as well as life cycle analysis and water scarcity indicators, respectively. The methodology for each is given separately in chapters 4 and 5. The results of the regional thermodynamic cost modeling are given in chapter 6. The operational life cycle impact assessment is given in chapter 7. The regional water scarcity impact assessment is given in chapter 8. Chapter 9 provides a discussion of the individual results as well as how they can be combined to give a holistic overview of a facility's cooling infrastructure sustainability.

CHAPTER 2

LITERATURE REVIEW OF THERMODYNAMIC MODELING OF FACILITY COOLING

Reliable, continuous access to IT resources often forces the building infrastructure to be over provisioned by as much as twice the actual operating loads [21] even at full IT utilization. This leads to inefficient operation of equipment, especially when considering the average IT processor utilization is considered only 20-30% of the total capacity. Although it is understood that this inefficiency has a significant impact on the bottom line, little work has been published which attempts to quantify it. Rambo-Joshi presented a thorough survey of data center modeling in 2007 [22]. Such a comprehensive review has not been completed since. However, significant work has been contributed in the gap, especially in the areas of air- and water-side economizers. It is understood that thermodynamic performance models are utilized internally by equipment manufacturers and facility designers. However, the thermodynamic performance of cooling loops that utilize evaporative cooling in place of vapor-compression refrigeration cycles has not been well documented within the body of literature at the time of this report.

2.1 Refrigeration-Based Modeling

Of particular interest for this work are the models presented by Iyengar-Schmidt and Breen et al. Iyengar-Schmidt [23] presented a design point model that considered both the flow work and the thermodynamic work of the facility cooling loop by accounting for both the air recirculation within the data hall and the parasitic heat load added by each sub-loop. The model was validated against a six MW data center and was utilized for parametric studies to determine the impact of raising the chilled water set point. The results showed that for the baseline case, the chiller consumed 41% of the cooling infrastructure total energy and that for every 10°C decrease in water temperature, a savings of 8% total plant energy was realized. Breen et al. [24] presented a simplified thermodynamic model based on both Iyengar-Schmidt and a model presented by Patel et al [25]. This model considered the impact of increasing rack inlet air temperature from chip to cooling tower. It was validated against published data given by Patel et al. and assumed a server inlet temperature as the design point. Subsequent studies [26] [27] considered the impact of server fan control algorithms and the effect of chip leakage power on the overall facility cooling efficiency. Collectively the work found that increasing rack inlet air temperature could improve the overall cooling efficiency if the server fan power and refrigeration work were balanced and if the server components had a low leakage gradient.

Each of these models, and subsequent variations, draw on component and sub-system modeling previously published. Braun [28] presented a detailed chiller plant model broken out by evaporator, condenser, and cooling tower performance. These models were validated against measurements taken from the central cooling plant at the Dallas/Fort Worth airport. Gordon-Ng [29] considered the thermodynamic performance of various water-cooled chillers and presented a universal thermodynamic model. Stout [30] presented a numerical model for characterizing cooling tower performance while evaluating the impact of variable-speed fan control. Typical meteorological year (TMY) data was utilized for five locations to simulate the ambient conditions of the cooling tower and results showed that the savings potential of implementing variable-speed fans was not directly related to the approach temperature.

2.2 Economizer Modeling

The savings potential represented by air- and water-side economizers is well-documented [7], [31], [32], [33] for optimal ambient conditions. To better understand the impact of ambient conditions on overall cooling system efficiencies, many studies have utilized a psychometric-based modeling approach coupled with TMY data for various climate zones to quantify the annual economizer hours possible and, subsequently, the energy savings realized. Sorell [34] was one of the first to present such a study in 2007. Considering four representative cities (San Francisco, Dallas, New York and London), it was shown that air-side economizers could be utilized for at least half of the year even in hot-humid climate zones.

Hellmer [35] presented a design-point model that utilized TMY data to give a comprehensive comparison of four cooling systems: refrigeration baseline, refrigeration with dry cooler, water-side economizer, and air-side economizer. He also considered three types of humidification-assist systems. A single operating point (22°C rack inlet temperature, 40% relative humidity) was considered for all cases and it was shown that air-side economizer with evaporative humidifier was the most efficient system. Water consumption was also taken into consideration and it was shown that air-side economizers were the most water efficient systems. Metzger et al. [36] utilized a bin analysis to compare potential energy savings of hybrid cooling systems across the ASHRAE climate zones. A savings of up to 80% was shown. Iyengar-Schmidt [37] was the first to present a comparative analysis of a direct evaporative outside air system. Utilizing a bin analysis method for Phoenix, AZ, a 30% energy savings was shown for the cooling system. Empirical studies [38], [39], [40] have presented the energy savings achieved by air-side economization.

CHAPTER 3

LITERATURE REVIEW OF LIFE CYCLE ANALYSIS AND WATER CONSUMPTION OF DATA CENTERS

Numerous studies have been published which focus on the operational energy consumption and utilization of data centers and especially by the facility cooling systems as shown in section 1.2. While the environmental impacts of buildings and services have long been considered in other industries [41], it is only recently that the data center industry has become concerned with the sustainability of facilities [42] [43] [44] [45]. Life cycle assessment (LCA), as defined by ISO 14000 series, is accepted as a standard methodology for evaluating environmental impacts [46] [47] of a defined product system. A full assessment will compile an inventory of all relevant inputs and outputs (or process streams) of the system over its full life (referred to as cradle-to-grave) and account for the emissions and resource utilization of each component to quantify its impacts. Recently, a methodology [48] for calculating the water footprint was added to the series. Several methodologies for assessing the impacts beyond the inventory footprint have been offered in the literature. Berger [49] presented a comprehensive review of these methodologies and categorized them by assessment type.

LCA for data center applications have either been product-based and assessed the cradle-to-grave impacts of IT hardware within the facility in terms of carbon emissions or process-based and assessed the energy consumption during the operational life cycle of the IT hardware, power distribution infrastructure, or cooling infrastructure. To date, the water footprint of the data center, and specifically the cooling infrastructure, has not been presented in published LCA as an impact indicator.

3.1 Water Scarcity Concerns

As William Sarni offers in the first chapter of his book *Corporate Water Strategies*, "[w]ater is *the* global environmental and social sustainability issue both for businesses and society [50]." Water scarcity is at the heart of public discourse around access rights, economic development, and regulatory measures. In response to growing public awareness, many companies have adopted corporate policies that aim to measure and report water utilization within different aspects of their businesses. The Carbon Disclosure Project (CDP) in 2010 added water use reporting to its list of priorities [51]. The aim of this campaign was to promote corporate water stewardship to the level of carbon footprint awareness. CDP [52] presented the results of its annual water risk survey of over 2,000 companies and reported that 68% of responding companies consider water to be a substantive risk to productivity. One-third of respondents have incorporated key performance indicators into their water management processes and over half consider the impact of legislative water regulation in risk assessment.

Lambooy [53] presented an analysis of 20 Dutch multinational companies to determine the state of corporate water strategies and practices. His analysis revealed that companies were largely expected to bear responsibility for direct impact on water resources, especially when public access to resources was impeded. Chapagain-Tickner [54] presented a review of water footprint development as a concept and as a tool for risk assessment. They concluded that water footprint assessments had historically been effective for generating overall awareness of the global water scarcity problem and motivating corporate investors to consider water-related risk assessments for potential business investments. However, it had not been very effective at translating risk into tangible improvements in water resource management. As a result, best practices were recommended for using water footprint tools in the broader context of public policy development and management of shared risks.

CHAPTER 4

METHODOLOGY FOR THERMODYNAMIC MODELING

The capital expenditures (cap-ex) traditionally represent only 15-25% of the total cost of a data center facility. Over the lifetime of the facility, the operational expenditures (op-ex) represent the bulk of the cost. As such, the simplified cost model presented here allows the user to isolate the impacts of cooling infrastructure design choices upon the economic bottom line in terms of op-ex only. The model is simply an aggregation of the hourly utility costs (electricity and water) resultant of cooling the total heat load generated within the facility over a period of one year. Traditionally, a cost model would consider other variables such as power loss in transmission, lighting, labor, or annuitized equipment cost.

A thermodynamic model was developed for each of the three cooling system types. The derivation of each model was based upon First Law Thermodynamic analysis and a single methodology was employed throughout. Each sub-loop of a given infrastructure was treated as a simplified black-box model, considered only the energy and mass flow into and out, and utilized published norms for component efficiencies with the exception of the origin heat load and flow rate of the cooling fluid through the first sub-loop. Empirically derived performance data were utilized as model inputs.

A simplified thermodynamic cost model has been developed based upon the model presented by Patel-Shah [55]. Both the energy and water consumption are calculated on an annual basis resultant from the total heat load that is processed by the facility cooling infrastructure. A separate model is considered for each cooling system and detailed in the following sections. The boundary conditions are considered the same for energy calculations across models and quasi-steady state conditions are assumed to exist at each operating point. Empirically derived values for the IT heat load and air flow rate across the server at fixed server power utilizations are taken as the model input variables. TMY-3 [56] data is utilized for outside ambient conditions considered for 14 cities across the ASHRAE climate zones. Municipal utility rates for power and water are utilized as published by the providers for the same 14 cities. It is recognized that utility rates are generally negotiated on a case-by-case basis for new facility construction. The details of these agreements are not generally available publically so commercial or volume discount rates are utilized as available.

4.1 Thermodynamic Model Sub-Loops

4.1.1 Fan Models

Throughout the modeling of the cooling systems, the flow rates of the fans and pumps are calculated assuming that continuity applies such that the working fluid is provided at exactly the flow rate required to extract the heat which has been added to the control volume (air recirculation is not considered). With the exception of the server fans, where flow rate data has been empirically collected, flow rates are calculated through the mass-energy balance given as

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \qquad (4\text{-}1)$$

The corresponding fan power consumptions are calculated using reference conditions taken from manufacturer data and application of fan affinity laws.

4.1.2 Room Model

The room model is taken as the same for each of the three cooling systems. Variations in modeling sub-loops occur from the point of exhaust from the room on.

4.1.2.1 Server modules

Each server module is comprised of a heat generation source, $\dot{Q}_{serv}$ and a fluid flow rate, $\dot{m}_{sf}$, which are given as the system inputs. $\dot{Q}_{serv}$ is derived from the following equation

$$\dot{Q}_{serv} = P_{serv} - P_{sf} \qquad (4\text{-}2)$$

where P_{serv} is the measured power draw of the server and P_{sf} is the measured power draw of all enclosed server fans. The power supply fan power is included within this term. Empirically measured power draws were utilized for this thesis. This methodology is recommended to most accurately model the cooling infrastructure performance as values vary widely from the manufacturer's nameplate values depending upon the operating environment of the data hall.

4.1.2.2 Rack modules

Each rack module is comprised of N server modules. The heat load and fluid flow rate are given respectively as:

$$\dot{Q}_{rack} = N_{serv} \times \dot{Q}_{serv} \tag{4-3}$$

$$\dot{m}_{rack} = N_{serv} \times \dot{m}_{serv} \tag{4-4}$$

Assuming uniformity of fluid properties and rack inlet temperature across the face of the rack, the rack outlet temperature can be calculated by inserting $\dot{Q}_{rack} = N_{serv} \times \dot{Q}_{serv}$ (4-3) and $\dot{m}_{rack} = N_{serv} \times \dot{m}_{serv}$ (4-4) into (4-1) giving

$$T_e = \frac{\dot{Q}_{rack}}{\dot{m}_{rack} c_p} + T_i \tag{4-5}$$

The rack cooling power requirement is given as

$$P_{rack} = N_{serv} \times P_{sf} \tag{4-6a}$$

Hardware uniformity is assumed for this thesis. However, it is recognized that non-uniformity of server hardware within a rack is common. In that case, the above becomes

$$P_{rack} = \sum_{i=1}^{n} (N_{serv} \times P_{sf})_i \tag{4-6b}$$

The room is comprised of *n* racks x *m* rows. The total heat dissipated by the IT hardware is given as

$$\dot{Q}_{IT} = \dot{Q}_{rack} \times (n \times m) \tag{4-7}$$

This is also taken as the total useful work produced by the data center for efficiency calculations.

Due to the assumption of continuity and uniformity of air across the room, the total fluid flow rate required is calculated as

$$\dot{m}_{room} = \dot{m}_{rack} \times (n \times m) \tag{4-8}$$

Assuming uniformity of racks within the room, the total cooling power requirement is given as

$$P_{room} = P_{rack} \times (n \times m) \tag{4-9}$$

An additional heat load is also considered from the losses (inefficiencies) of the fans. Fan efficiencies vary widely not only between models but also within the operating range and conditions of the room. In this model, a single efficiency is assumed for each design point (inlet air temperature). The total heat load dissipated by the room is then given as

$$\dot{Q}_{room} = \dot{Q}_{IT} + P_{room}(1 - \eta_{sf}) \tag{4-10}$$

This thesis models a facility comprised of a single data hall and thus $\dot{Q}_{room}$ and P_{room} are the common point for each cooling system model. A facility in actuality may be comprised of multiple data halls with varying loads and cooling power requirements. Due to these variations, this model requires that the heat loads and power requirements be summed at this point before considering the addition of heat losses from additional cooling infrastructure components.

4.1.3 Chiller Plant Model

This model, shown in Figure 4.1, is based upon that presented by Breen et al. [24] and is considered as a baseline for comparison to the other cooling systems because the thermodynamic performance has been well documented and understood. This model is also included due to the prevalence of this cooling system in legacy data centers and the continued use by enterprise facilities. The chiller plant is represented by an air-liquid heat exchanger (the computer room air conditioner/handler or CRAC/H), and two liquid-liquid heat exchangers (the evaporator and the condenser) coupled by a vapor-compression system.

4.1.3.1 CRAC/H modules

CRACs are typically placed around the perimeter of the data hall or in the common corridors shared by multiple data halls, depending on the type of facility. The heat load processed by the CRAC is comprised of the total heat dissipated by the room plus the heat losses of the blower motors. The total heat load is given as

$$\dot{Q}_{CRAC} = \dot{Q}_{room} + \dot{Q}_{blow} \tag{4-11}$$

where

Figure 4.1 Flow model of chiller-plant cooling infrastructure for data center facility.

$$\dot{Q}_{blow} = N_{CRAC} \times P_{blow}(1 - \eta_{blow}) \tag{4-12}$$

The blower efficiency can be calculated from manufacturer's data assuming 100% shaft efficiency or using the measured power draw of the motor and calculating the fan-motor system efficiency. Either way, the power of the blower can be calculated by applying fan affinity laws and utilizing manufacturer's reference data as

$$P_{blow} = P_{ref} \times (\frac{\dot{\forall}_{room}}{\dot{\forall}_{ref}})^3 \tag{4-13}$$

This model assumes the heat exchanger to be 100% efficient such that

$$\dot{Q}_{BCW} = \dot{Q}_{CRAC} \tag{4-14}$$

4.1.3.2 Chiller modules

The heat load transferred to the evaporator is given as

$$\dot{Q}_{evap} = \dot{Q}_{BCW} + P_{BCW}(1 - \eta_{BCW\,pump}) \tag{4-15}$$

where the pump power is calculated as

$$P_{BCW} = \frac{\dot{m}_{BCW} \times \Delta p_{BCW}}{\eta_{BCW\,pump}} \tag{4-16}$$

Δp_{BCW} is the effective pressure drop across the building chilled water line assuming the evaporator is plumbed in-line. The pump efficiency can be calculated in a similar manner as the blower above.

The chiller plant model is typically dominated by the work required by the chiller compressor. For this model, the work input is given as

$$\dot{W}_{chiller} = \frac{\dot{Q}_{evap}}{COP_{chiller}} \tag{4-17}$$

The COP of a chiller may be given by the manufacturer for a specific working condition in terms of the condenser and evaporator exit temperatures and/or in terms of kW/ton for multiple load percentages. This model utilizes the latter and the COP for a given work load percentage is interpolated between the given points. The heat load transferred from the chiller to the condenser then is given as

$$\dot{Q}_{cond,i} = \dot{Q}_{evap} + \dot{W}_{chiller} \tag{4-18}$$

Finally, the heat transferred from the condenser is given as

$$\dot{Q}_{cond,e} = \dot{Q}_{cond,i} + P_{CTW}(1 - \eta_{cond\,pump}) \tag{4-19}$$

where, just as with the building chilled water line, the pump power required for the condenser water is given as

$$P_{CTW} = \frac{\dot{m}_{CTW} \times \Delta p_{CTW}}{\eta_{cond\ pump}} \tag{4-20}$$

4.1.4 Evaporative Cooling Models

Two thermodynamic cooling models are developed to consider the removal of the vapor-compression refrigeration loop within the facility cooling system. The first model considers an indirect evaporative cooling system that utilizes a cross-flow shell tube cooling coil, a blower, and an open circuit cooling tower to cool the air supplied to the data hall. This model assumes a flooded supply-contained return closed-loop air distribution method, shown in Figure 4.2 [57]. Alternative supply methods that could be considered are overhead flooded supply (also referred to as a penthouse design) and raised-floor directed supply. Key considerations for this model are the effectiveness of the cooling coil as dictated by the available water temperature and the outside ambient conditions that impact the cooling tower performance.

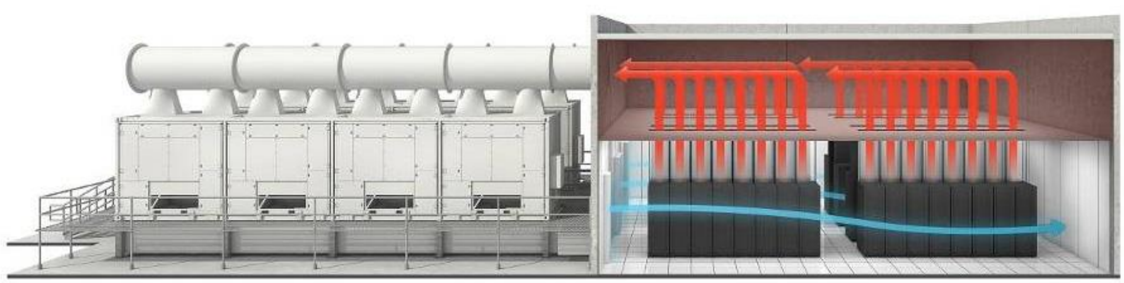


Figure 4.2: Flooded supply – contained return air distribution to data hall with cooling unit outside.

The second model considers a direct evaporative cooling system that utilizes a wet-pad cooler. This model also assumes a flooded supply-contained return method. However, it is considered as an open loop for the majority of the year.

Both cooling systems consist of an air handler unit that utilizes a supply air blower to both pull air through the heat exchanger and supply cool air to the data hall. Both systems also utilize a pump to circulate the cooling fluid through the heat exchanger. An energy balance methodology is utilized to calculate the required pumping powers and heat loads added to the system for both the supply air blower and the pump and is given as follows.

4.1.4.1 Supply air

The total heat load exiting the data hall is given as

$$\dot{Q}_{DC} = \dot{Q}_{room} + \dot{Q}_{blow} \tag{4-21}$$

where the additional heat load due to the blower motor inefficiency is given as

$$\dot{Q}_{blow} = N_{blow} \times P_{blow}(1 - \eta_{blow}). \tag{4-22}$$

The required power to supply the required volume of air to the data hall is given as

$$P_{blow} = P_{ref} \times (\frac{\dot{\forall}_{room}}{\dot{\forall}_{ref}})^3. \tag{4-23}$$

4.1.4.2 Water pump

The additional heat load due to the pump inefficiency is given as

$$\dot{Q}_{WP} = N_{WP} \times P_{WP}(1 - \eta_{WP}). \tag{4-24}$$

The required pump power is given as

$$P_{WP} = P_{ref} \times (\frac{\dot{\forall}_w}{\dot{\forall}_{ref}})^3. \qquad (4\text{-}25)$$

4.1.4.3 Direct cooler module

The direct evaporative cooling system consists of the wet-pad evaporative cooler, the water circulation pump and the supply air blower. It is modeled as an adiabatic saturation cooling process with the water flow rate fixed for all inlet-air conditions. As an open-loop model, the supply air is drawn from the outside ambient, passed through the data hall, and exhausted directly back to the outside ambient. This work assumes that the inlet and exhaust air streams do not mix. The inlet air temperature and relative humidity are given by the TMY3 data and the required exit air temperature is given by the design rack-inlet temperature. This can be taken either as a single design set-point or as an operating envelope of temperature and humidity conditions. A simple check is performed to ensure that the required conditions are met. The minimum exit air temperature that can be achieved at a given time is calculated as

$$T_{min,a} = T_{a,i} - SE_{media}(T_{a,i} - T_{wb}) \qquad (4\text{-}26)$$

where T_{wb} is the wet bulb temperature of the outside ambient air. The minimum exit air temperature must be within the set-point range (or at the temperature if a single operating point is preferred). The exit air temperature from the cooler is then taken as the supply air temperature for the data hall.

$$T_{a,e} = T_{a,i\,DC} \tag{4-27}$$

A mass balance is utilized to calculate the amount of water consumed due to evaporation. Assuming the ambient air is comprised of two ideal gasses, dry air and water vapor, the rate of evaporation is given as

$$\dot{m}_{evap} = \dot{m}_{da}(\omega_{a,e} - \omega_{a,i}) \tag{4-28}$$

where the mass flow rate of the dry air is given by

$$\dot{m}_{da} = \frac{\dot{\forall}_a}{v_{da}} \tag{4-29}$$

and the humidity ratio of the air exiting the cooler is given as

$$\omega_{a,e} = \frac{h_{da,i} - h_{da,e} + \omega_{a,i}(h_{g,i} - h_f)}{h_{g,e} - h_f}. \tag{4-30}$$

The humidity ratio of the inlet air is taken from the TMY3 data and the enthalpy values can be taken from the thermodynamic property tables for steam and air.

4.1.4.4 Indirect cooler module

The indirect module is comprised of cooling coil air-to-liquid heat exchanger, the water pump and the supply air blower. The cooling coil is modeled as two fluid streams as described in the following section.

4.1.4.4.1 Cooling coil air-side

The inlet air temperature is taken as the exhaust air temperature from the data hall,

$$T_{a,i} = T_{e,room} \tag{4-31}$$

and the mass flow rate is assumed to be the same as flow rate through the data hall,

$$\dot{m}_a = \dot{m}_{room}. \tag{4-32}$$

The exhaust air temperature is given by

$$T_{e,room} = \frac{\dot{Q}_{room}}{\dot{m}_{room} c_p} + T_{i,DC}. \tag{4-33}$$

The exit air temperature is given as

$$T_{a,e} = \frac{\dot{Q}_{DC}}{\dot{m}_{room} c_p} + T_{a,i} \tag{4-34}$$

where $\dot{Q}_{DC}$ is given by

$$\dot{Q}_{DC} = \dot{Q}_{room} + \dot{Q}_{blow}$$ (4-21

(4-35)

4.1.4.4.2 Cooling coil water-side

The inlet water temperature is calculated using the heat exchanger efficiency definition and is given as

$$T_{w,i} = T_{a,i} - \frac{T_{a,i} - T_{a,e}}{\varepsilon_{CC}}. \tag{4-36}$$

The exit water temperature is calculated from the energy balance equation and given as

$$T_{w,e} = \frac{\dot{Q}_{CT}}{\dot{m}_w c_p} + T_{w,i} \tag{4-37}$$

where the heat load processed by the cooling tower is defined as

$$\dot{Q}_{CT} = \dot{Q}_{DC} + P_{WP}(1 - \eta_{WP}). \tag{4-38}$$

4.1.5 Cooling Tower Model

The cooling tower model presented here is for an induced draft, cross-flow wet cooling tower. This model is utilized for both the chiller-based and the indirect evaporative cooling systems. The water flow rate through the system is dictated by the condenser water pump and the air flow rate is dictated by the cooling tower fan. In reality, the water distribution nozzles, the fill media, and the pressure differential within the air column all add resistances to the respective fluid flow rates. However, these resistances are not considered for this model and the mass flow rates are assumed to be uniform for the respective inlet and exit points. Assuming the following equality

$$\dot{Q}_{CT} = \dot{Q}_{cond,e} \tag{4-39}$$

and utilizing the energy balance given in $\dot{Q} = \dot{m}C_p(T_e - T_i)$ (4-1), the entering water temperature (same as condenser exit temperature) can be calculated as

$$T_{cdw,e} = \frac{\dot{Q}_{cond,e}}{(\dot{m}_{CTW} \times C_{p,water})} + T_{cdw,i} \tag{4-40}$$

4.1.5.1 Heat exchanger

The heat and mass transfer from the cooling tower water to the air can be modeled as a sensible heat exchanger following the effectiveness method given by [28].

$$\dot{Q}_{CT} = \dot{m}_{CT,air} \times \Delta h_{CT} \tag{4-41}$$

$$h_{a,e} = \varepsilon_{CT,air}\left(h_{sa,i} - h_{a,i}\right) + h_{a,i} \tag{4-42}$$

$$\dot{m}_{CT,air} = \frac{\dot{m}_w(h_{f,wi} - h_{f,we})}{h_{a,e} - h_{a,i} + h_{f,we}(\omega_{a,i} - \omega_{a,e})} \tag{4-43}$$

4.2 Cooling Power

The facility cooling power is given as a simple summation of the required pumping power calculated for each sub-loop. This is given as

$$P_{Cool} = \sum P_B + \sum P_P + P_C \tag{4-44}$$

where Pb is the blower (fan) power, Pp is the pump power, and Pc is the chiller work input into the system.

4.3 Water Consumption Due To Evaporation

4.3.1 Cooling Tower Evaporation

The mass of water evaporated from the cooling tower is calculated following the methodology given by [28], which draws on the principles of psychometrics and utilizes a basic mass balance. The total rate of evaporation is given as

$$\dot{m}_{evap} = \dot{m}_{air}(\omega_{a,e} - \omega_{a,i}) \tag{4-45}$$

defined by the humidity ratio of the exit and inlet air, respectively. The humidity ratio of the inlet air is calculated from the given TMY3 data. The exit air humidity ratio is given by

$$\omega_{a,e} = \omega_{SE} + e^{-NTU}(\omega_{a,i} - \omega_{SE}) \tag{4-46}$$

which considers the humidity ratio of air at the saturated condition. This is related through the enthalpy of air at the saturated condition given as

$$\omega_{SE} = \omega_{@h_{SE}} \tag{4-47}$$

$$h_{SE} = h_{a,i} + \frac{(h_{a,e} - h_{a,i})}{1 - e^{-NTU}} \tag{4-48}$$

The enthalpy of the inlet air is taken from the property table for ideal air. The enthalpy of the exit air is given by $h_{a,e} = \varepsilon_{CT,air}\left(h_{sa,i} - h_{a,i}\right) + h_{a,i}$ (4-42).

$$NTU = \frac{ln(1-SE)+ln(m \times SE)}{-2(1-m)} \tag{4-49}$$

$$m = \frac{\dot{m}_{air}}{\dot{m}_w \frac{C_{p,w}}{C_s}} \tag{4-50}$$

$$C_s = \frac{h_{sa,i}-h_{sa,e}}{T_{w,i}-T_{w,e}} \tag{4-51}$$

4.3.2 Adiabatic Saturation Evaporation

$$T_{min,a} = T_{a,i} - SE_{media}(T_{a,i} - T_{wb}) \tag{4-52}$$

$$T_{a,e} = T_{a,i\,DC} \tag{4-53}$$

$$\dot{m}_{evap} = \dot{m}_{da}(\omega_{a,e} - \omega_{a,i}) \tag{4-54}$$

$$\dot{m}_{da} = \frac{\dot{\forall}_a}{v_{da}} \tag{4-55}$$

$$\omega_{a,e} = \frac{h_{da,i}-h_{da,e}+\omega_{a,i}(h_{g,i}-h_f)}{h_{g,e}-h_f} \tag{4-56}$$

4.4 Annual Operational Cost

The power requirements of the sub-loops up to the direct evaporative heat exchange (either the wet-pad media or the cooling tower) as well as the water pump power are assumed to be constant for a given year. The total kW-hr consumption then is simply the cooling power multiplied by 8,736 hours per year. The fan power consumed by the cooling tower fan is calculated on an hourly basis and summed. The water consumption is calculated on an hourly basis as described in the previous section and summed for the total annual water consumption. The total operational cost associated with the cooling infrastructure is then calculated by applying the appropriate utility rate for each geographic location to the respective utility consumption (electricity and water).

Utility rates for this work were taken directly from the municipalities represented as the commercial (or volume discounted where available) rate at the time of publication. Although common practice is to utilize electricity rates published by DOE for a given state, two drawbacks were identified for this work. Similar water rates are not currently available making it necessary to utilize the individual rates published. Additionally, utility rates can vary widely within a given state and change more frequently than DOE updates reference material.

CHAPTER 5

METHODOLOGY FOR LIFE CYCLE IMPACT ANALYSIS OF COOLING INFRASTRUCTURE

To date, there is no methodology that accounts for both the water consumption of a process stream (water inventory) and the water scarcity of a region (impact assessment) as granular as a state. This is especially true within the US as there is no single regulatory body that accounts for the total annual freshwater withdrawal for human use or the available renewable water supply for each state. This is largely due to the variable nature of water rights development and adoption across the US as well as the interstate movement of the key freshwater sources (i.e. the Colorado River Basin touches seven states and is the primary water source of six of those as well as parts of Northern Mexico [58]). The USGS does attempt to provide some of this data through the National Water Information System [59]. However, this database is a conglomeration of site-specific data as reported and there is not guaranteed consistency in the parameters that are reported or monitored. Existing indices typically utilized for water impact assessment only report national scarcity and do not account for power generation very well.

The methodology presented for this work is an operational life cycle impact assessment (OLCIA) utilizing a simple water inventory of the cooling system cycle (an annual peak-efficiency inventory is already embedded within the thermodynamic cost model) in conjunction with the water stress indicator (WSI) developed by Smakhtin et al. [60] and given by (5-1)

$$WSI_i = \frac{WU_i}{WR_i - EWR_i} \tag{5-1}$$

as a characterization factor.

5.1 Life Cycle Water Inventory

The lifetime water inventory also considers the embedded water consumption of the electricity required by the cooling process, which includes the power consumed by all pumps and fans within the system as well as the power consumed by the IT hardware. The embedded water is calculated using the results presented by Torcellini et al. [61]. Utilizing the values presented by Fthenakis-Kim [62], the embedded water calculation can be further refined by fuel type. The operational life of the cooling system is taken to be that of the building (25 years) while the operational life of the individual components varies significantly. It is also recognized that performance of those components degrades with increased use. As such, annual power and water consumption values are adjusted to account for these variations. The final inventory is presented in normalized terms of gallons of water per kWh consumed to account for differences in annual operating hours.

5.1.1 Embedded Power Generation Water

In many markets, the exact source of the grid electricity cannot be identified as the power plants collectively sell into a marketplace and various providers then buy as demand requires. In order to account for this, a power mix profile is calculated for each state utilizing the net electricity generation by source data published in [63]. Using the weighting method from [61] along with the averaged water consumption by source from [62], a weighted average embedded water consumption rate is calculated for each location. This is later applied to the calculated lifetime power consumption of the facility to calculate the total embedded water consumption.

5.2 Water Stress Indicator

The WSI is a variation of the use-to-availability ratio commonly applied as an indicator for water footprinting. The numerator is the total regional water consumption associated with a process or product. The denominator is the difference between available renewable water reserves and the environmental water requirement of the defined region. By accounting for both available reserves and the environmental water requirement, WSI quantifies the water scarcity impact of the process for a given region. The 14 cities previously identified are considered the regions and the USGS database is utilized to calculate the WSI for each.

5.2.1 Water Availability

The total water availability for a specific geographic location was calculated utilizing watershed maps and site data from the National Water Information System in conjunction with data published by each municipality stating the watershed sources for their respective water supply. Water reports for water year 2013 were selected for each reporting site feeding a given watershed. The average annual water runoff was taken from each report and summed to determine the available water supply for the given water region associated with each geographic location.

5.2.2 Environmental Water Requirement

The EWR was determined utilizing the MAR methodology presented in [60].

Table 5-0-1 Environmental water requirement (EWR) as a percentage of the mean annual runoff (MAR) for the major water basins in the US.

Basin	EWR (% of MAR)
Arkansas	41
California	23
Colorado	27
Colombia	33
Great Lakes	49
Mississippi	42
Missouri	40
Ohio	45
Pacific North America	30
Rio Grande	28
Southeast US	35
US Northeast	38

5.2.3 Water Utilization

The total water utilization is calculated utilizing the same data sources named in 5.2.1. The total withdrawals for each line (public supply, agriculture, power generation, etc.) are taken and summed to find the total water utilization for each geographic location.

5.3 Water Scarcity Impact

The water scarcity of each location is determined utilizing the values from the previous section.

CHAPTER 6

THERMODYNAMIC COST MODELING CASE STUDY

6.1 Overview

This study utilized a representative facility with a 1-MW IT capacity to compare the thermal performance of each cooling infrastructure. The facility comprised of a single data hall with hot-isle containment. The IT heat loads and room airflow rates utilized were collected experimentally and reported in a separate study. The experimental set-up and complete results were given in detail by [64]. Each sub-loop was modeled assuming an N+1 redundancy configuration. This study did not consider the capital expenditures associated with any of the cooling infrastructures presented. As such, the author had the luxury of selecting highly efficient components. However, similar efficiencies were assumed for components across the three infrastructures to allow for comparison.

6.2 Model Assumptions

The chiller plant model followed a legacy facility infrastructure with down flow CRAHs located inside the data hall with a raised-floor supply configuration. This study assumed that the underfloor plenum was sufficiently pressurized and that back flow recirculation did not occur. The CRAHs were chilled-water cooled from the centralized chiller plant. It should be noted that the CRAH selected for this study was the unit utilized in the experimental set-up detailed in [64]. This model is considered older and inefficient by current standards. Although the unit is not equipped with variable frequency drives in actuality, this functionality was assumed for the purposes of this study. Both the evaporator and condenser pumps were placed in line and appropriate pressure drops were calculated for the pipe runs. All other system pressure drops were taken from manufacturer's data for the respective component.

This study assumed a single facility infrastructure for both of the evaporative cooling models. A series of modular air handler units (AHU) fed the data hall through a flooded-supply, contained-return air distribution configuration shown in Figure 4.2. The supply air blower drove the air distribution. A cooling tower and cooling coil were coupled with each AHU for the indirect model. A water sump and circulation pump were coupled with each AHU for the direct model. A 100% return air loop was assumed for the indirect model while a 100% outside air loop was assumed for the direct model.

The results presented here assumed a specific rack inlet air temperature design set-point criteria for each model in order to understand the impacts of the A4 operating envelope when considering standard operating procedures typically employed today. No model included an economizer mode of operation in order to understand the impacts of geographic region when considering chiller-less versus chiller-based cooling infrastructures. The modeling methodology presented in Chapter 4 could allow for the inclusion of economizer operation as well as operating with a defined envelope rather than a specific set-point.

6.3 Model Results

6.3.1 Operating Through the A4 Envelope

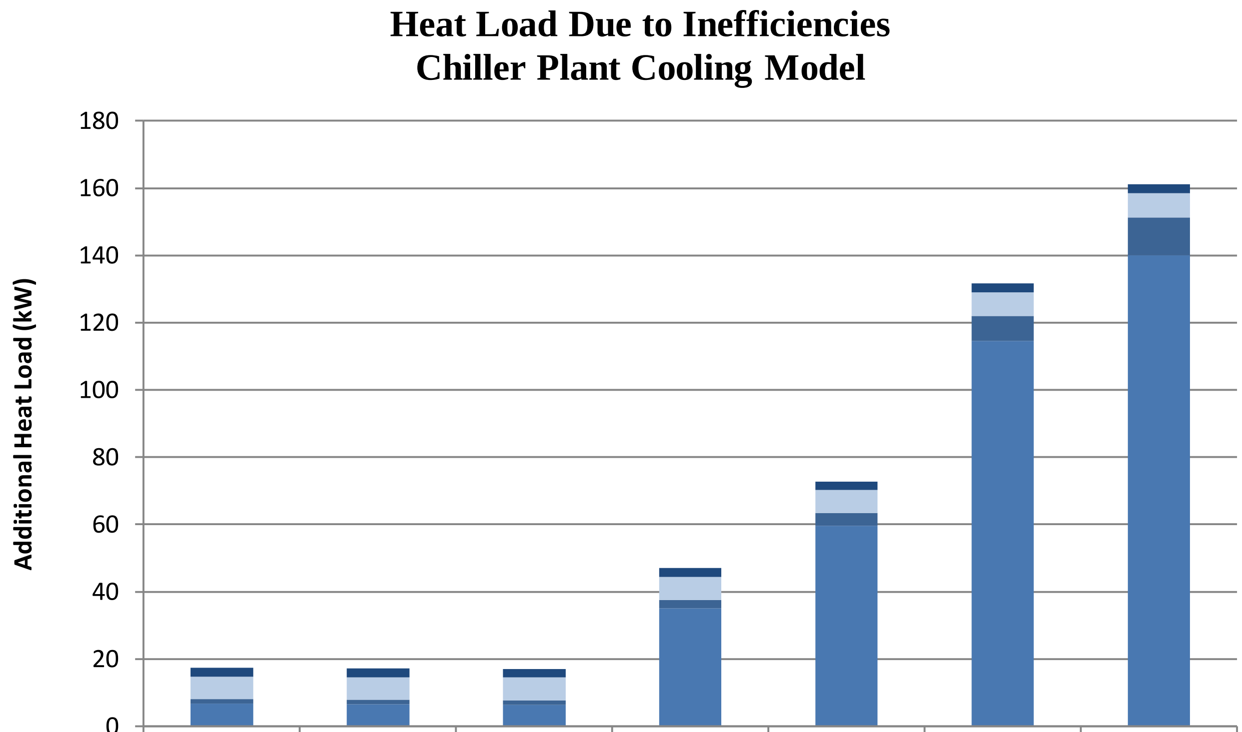
Figure 6.1through Figure 6.3 show the additional heat loads contributed to the cooling loop by each of the major system components. Each of the models shows similar trends with the heat loads relatively consistent up to 25°C and then increasing linearly. This is consistent with the results presented in [64]. Each of the models also demonstrates similar trends with the supply air blowers dominating from 30°C on.

Figure 6.4 through Figure 6.6 show the power draw of each of the major system components. Again, each of the models shows similar trends with the required power consistent up to 25°C and then increasing linearly. Although the compressor power required is the significant draw back for the chiller plant infrastructure, it should be noted that the power requirements of the other system components are approximately the same or less than the dedicated chiller-less systems. This demonstrates the importance of component efficiencies in the overall operation of the cooling infrastructure.

6.3.2 Operating Across Geographic Regions

Figure 6.7 through Figure 6.9 give the total annual operating cost inclusive of on-site water consumption, which is presented in detail in the following chapter. Based on operational cost, Minneapolis represents the best climate zone for any of the cooling infrastructures presented. The cost per kilowatt-hour is consistent across the A4 operating envelope for each of the systems modeled in this study. The exception is that dedicated direct evaporative cooling cannot be achieved above 40°C.

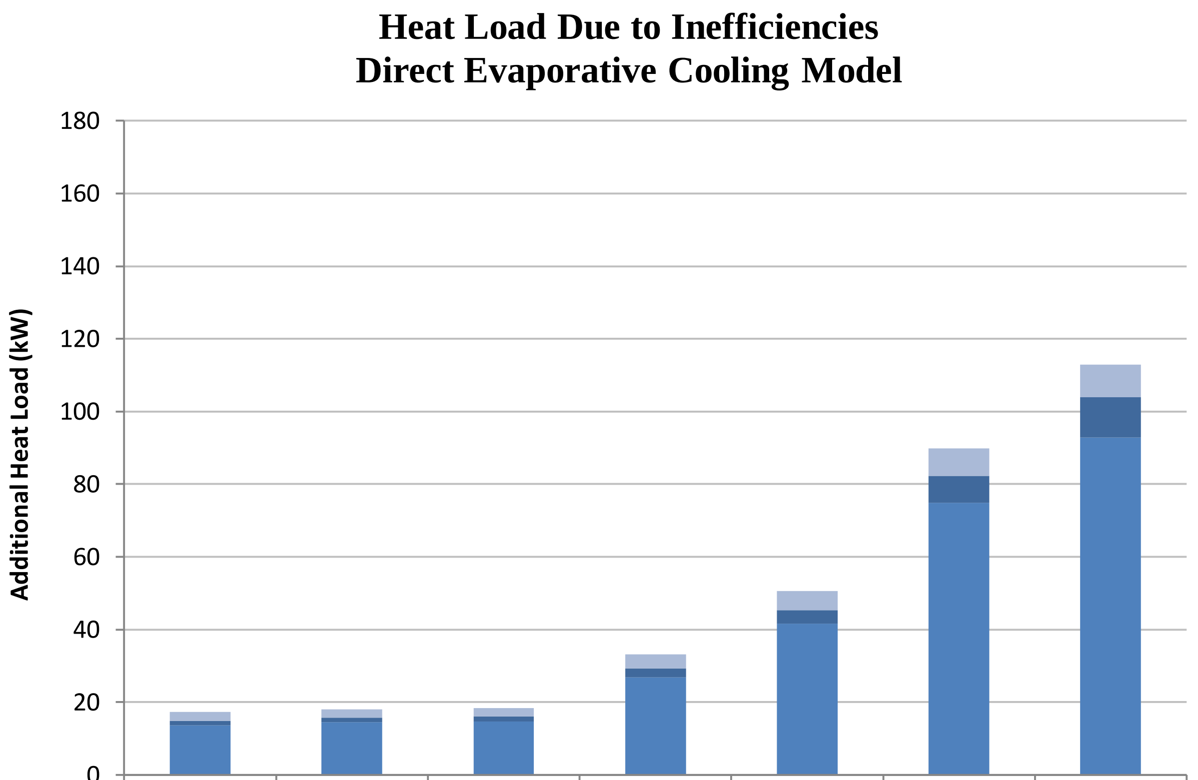
As expected, the geographic region does not significantly influence the operating cost of a chiller plant facility. Figure 6.7 shows that the cost trend through the operating envelope is similar across geographic regions. The variation in cost is largely due to utility price differences. A breakdown of utility prices can be found in Appendix A. Figure 6.8 and Figure 6.9 show the impact of climate on the operation of dedicated evaporative cooling systems. Of particular interest is the cost differential between the direct evaporative model and the other two models. In most cases, the cost is an order of magnitude higher for this model even though the power draw is very similar. It is also important to note that while the chiller plant model and the direct evaporative model show a general linearly decreasing trend in cost across the operating envelope, the indirect evaporative model shows an increasing cost up to 35°C and then dropping. This trend is seen for most of the geographic regions with minor variations in where the peak cost is.



	15	20	25	30	35	40	45
CW Q (kW)	2.56	2.59	2.60	2.62	2.54	2.64	2.64
Pump Q (kw)	6.80	6.81	6.78	6.81	6.75	6.99	7.09
Serv Fan Q (kW)	1.22	1.24	1.26	2.51	3.76	7.50	11.25
Blower Q (kW)	6.76	6.55	6.43	35.06	59.69	114.48	140.09

Rack Inlet Temperature (°C)

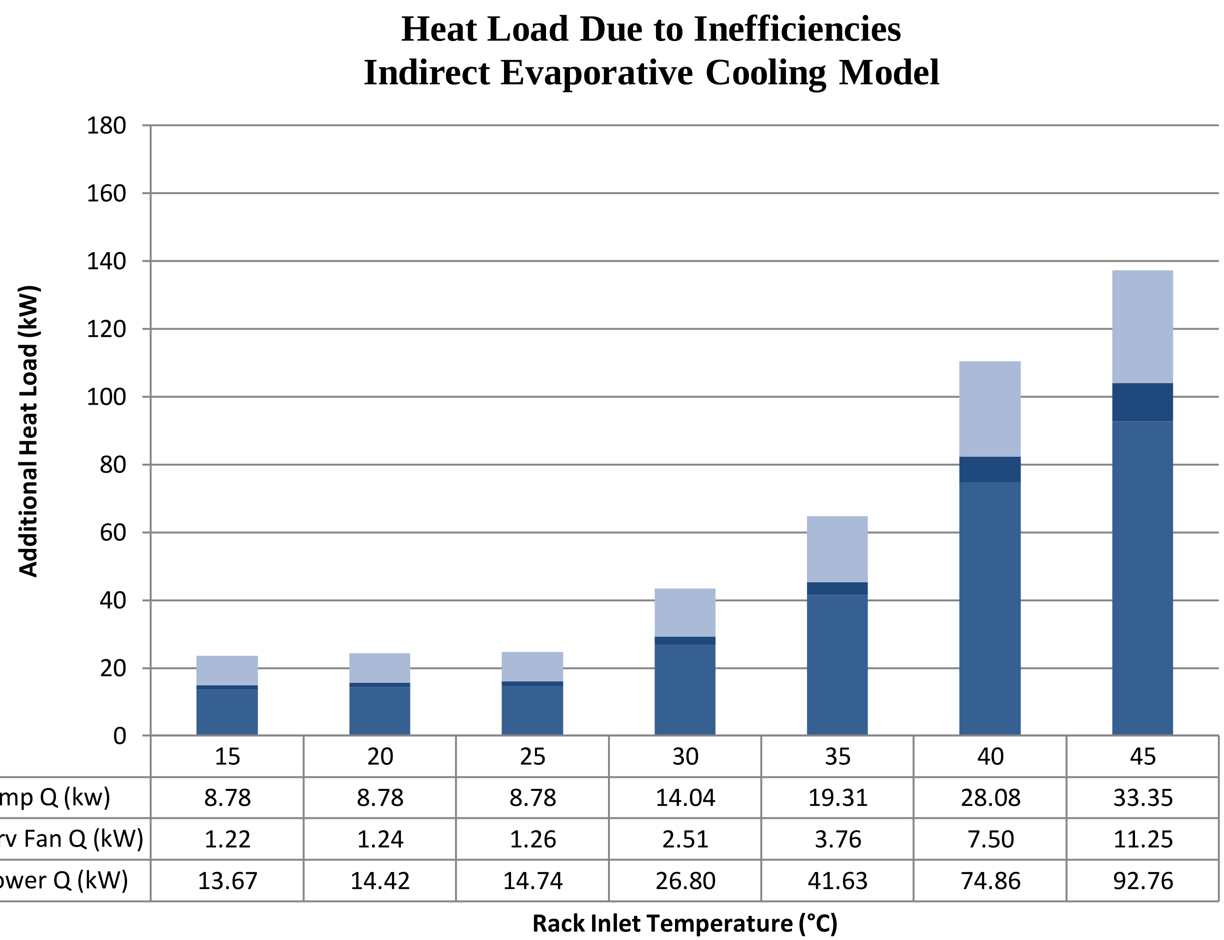
Figure 6.1 Additional heat load due to component inefficiencies across the A4 operating envelope for chiller plant cooling infrastructure.



	15	20	25	30	35	40	45
Pump Q (kw)	2.36	2.36	2.36	3.77	5.19	7.55	8.96
Serv Fan Q (kW)	1.22	1.24	1.26	2.51	3.76	7.50	11.25
Blower Q (kW)	13.67	14.42	14.74	26.80	41.63	74.86	92.76

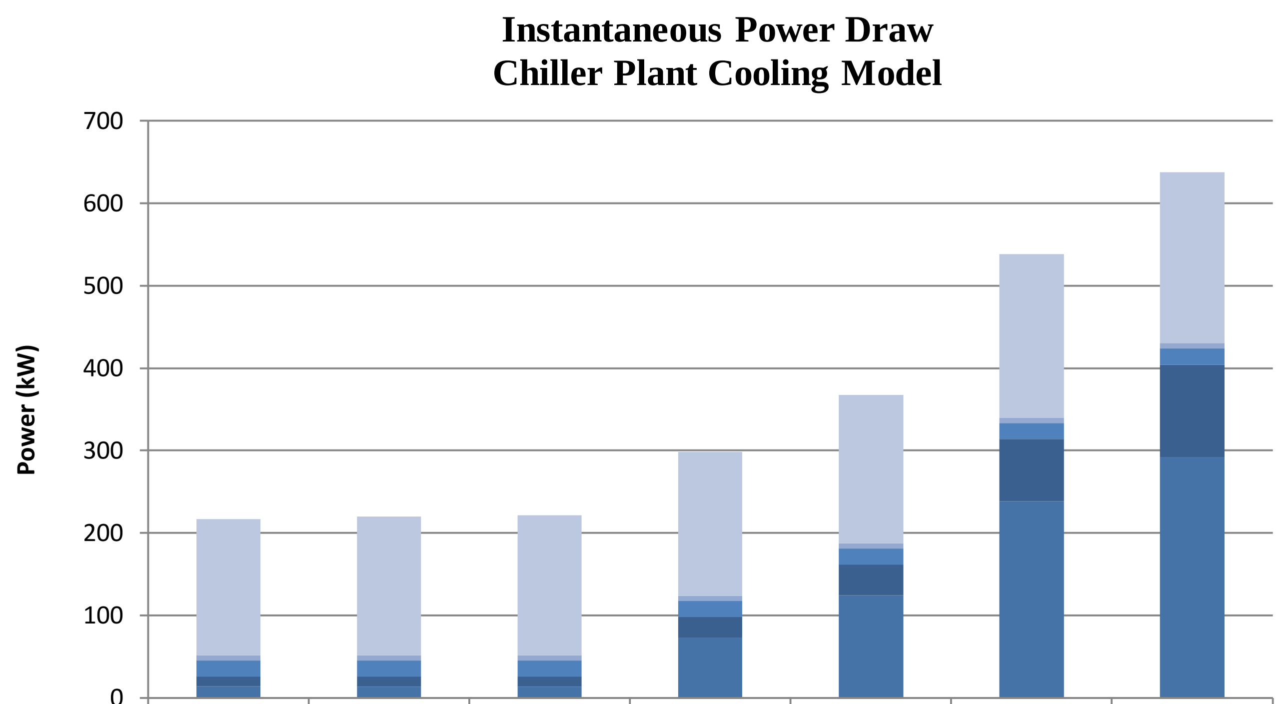
Rack Inlet Temperature (°C)

Figure 6.2 Additional heat load due to component inefficiencies across the A4 operating envelope for direct evaporative cooling infrastructure.



	15	20	25	30	35	40	45
Pump Q (kw)	8.78	8.78	8.78	14.04	19.31	28.08	33.35
Serv Fan Q (kW)	1.22	1.24	1.26	2.51	3.76	7.50	11.25
Blower Q (kW)	13.67	14.42	14.74	26.80	41.63	74.86	92.76

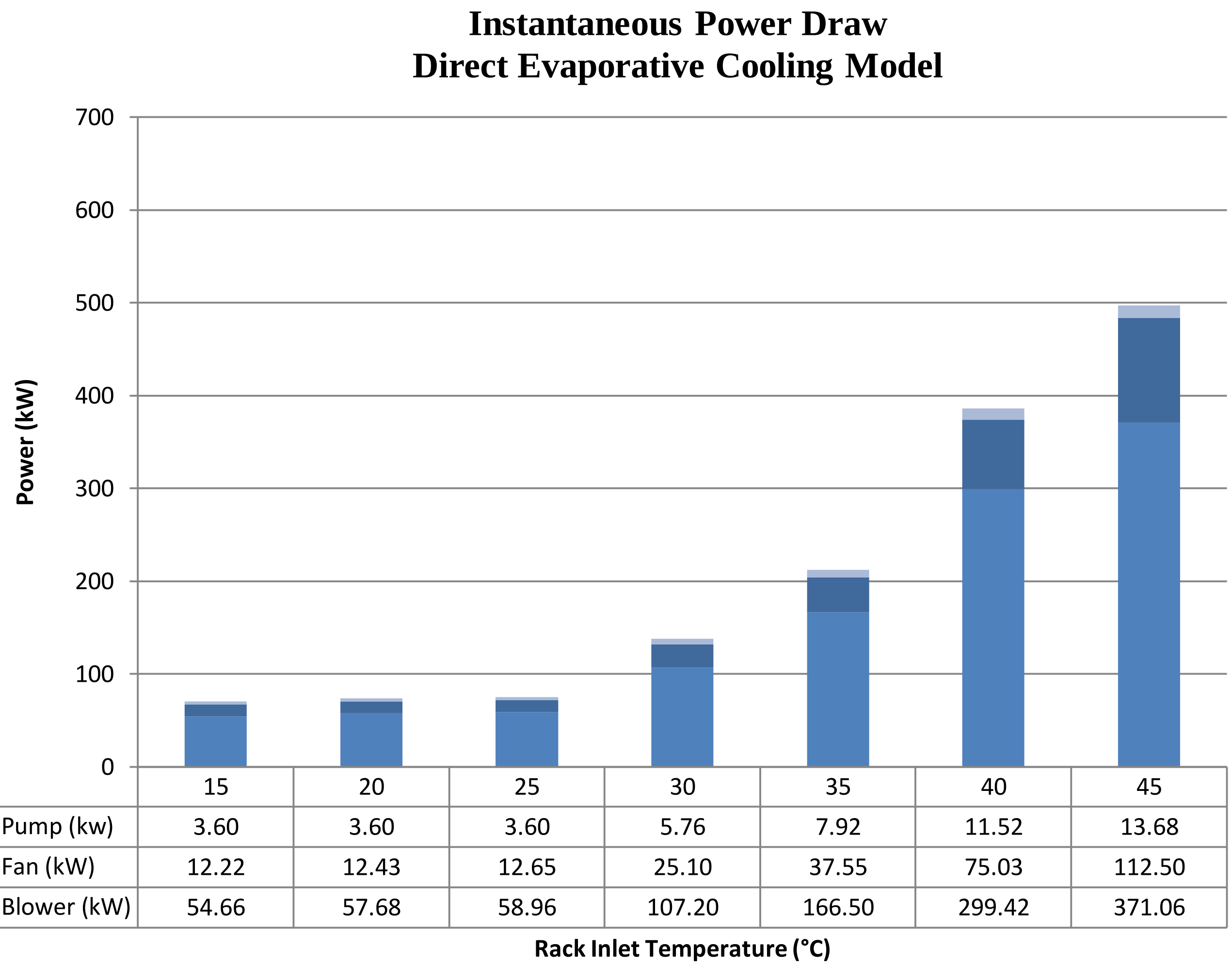
Figure 6.3 Additional heat load due to component inefficiencies across the A4 operating envelope for indirect evaporative cooling infrastructure.



	15	20	25	30	35	40	45
W Chiller (kW)	165	169	171	175	181	199	207
P CW (kW)	6.23	6.30	6.33	6.38	6.25	6.48	6.50
P Pump (kw)	19.05	19.07	18.99	19.06	18.92	19.59	19.87
P Fan (kW)	12.22	12.43	12.65	25.10	37.55	75.03	112.50
P Blower (kW)	13.80	13.37	13.12	73.05	124.35	238.51	291.86

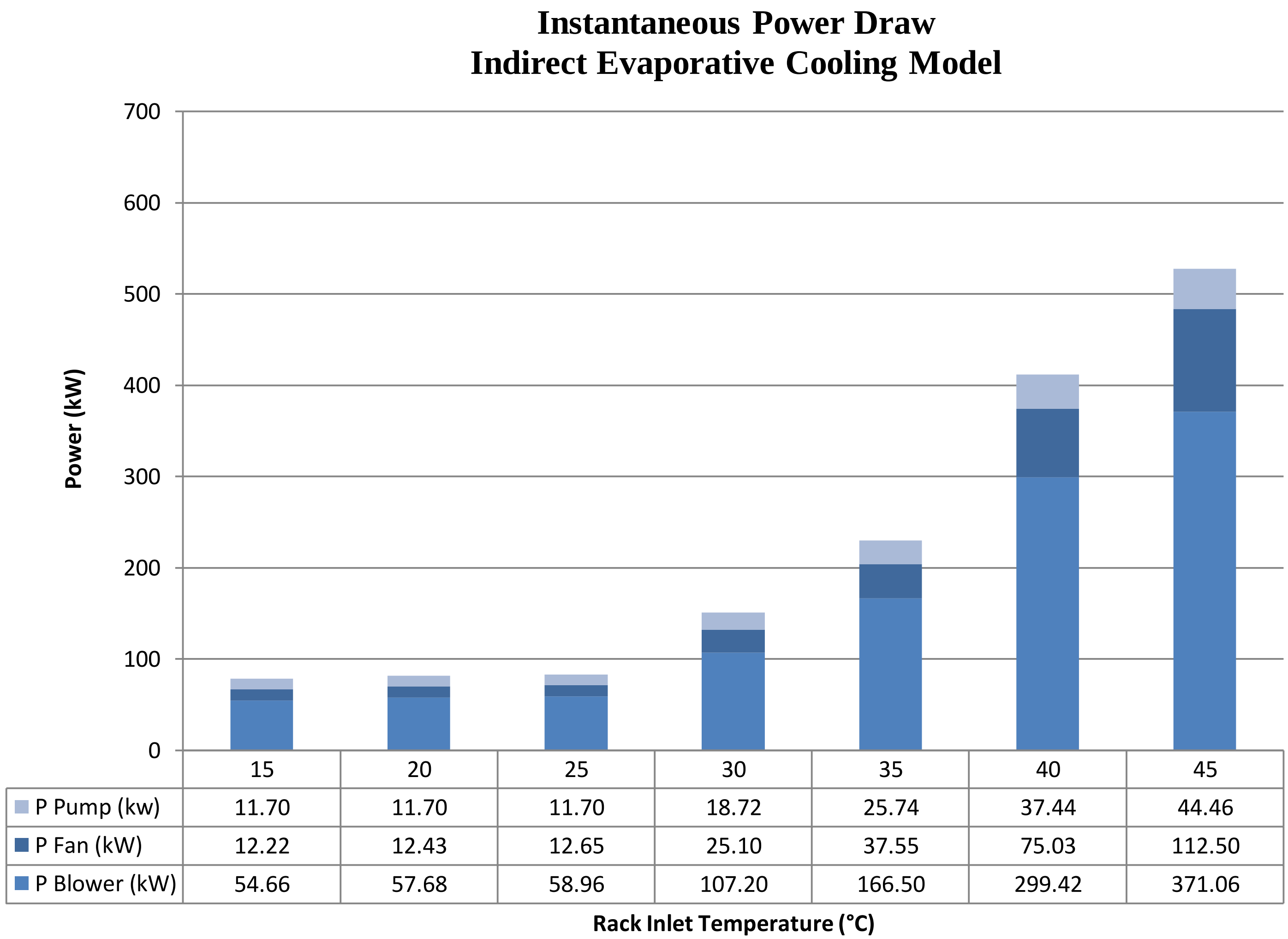
Rack Inlet Temperature (°C)

Figure 6.4 Cooling power across the A4 operating envelope for chiller plant cooling infrastructure.



	15	20	25	30	35	40	45
P Pump (kw)	3.60	3.60	3.60	5.76	7.92	11.52	13.68
P Fan (kW)	12.22	12.43	12.65	25.10	37.55	75.03	112.50
P Blower (kW)	54.66	57.68	58.96	107.20	166.50	299.42	371.06

Figure 6.5 Cooling power across the A4 operating envelope for direct evaporative cooling infrastructure.



	15	20	25	30	35	40	45
P Pump (kw)	11.70	11.70	11.70	18.72	25.74	37.44	44.46
P Fan (kW)	12.22	12.43	12.65	25.10	37.55	75.03	112.50
P Blower (kW)	54.66	57.68	58.96	107.20	166.50	299.42	371.06

Figure 6.6 Cooling power across the A4 operating envelope for indirect evaporative cooling infrastructure.

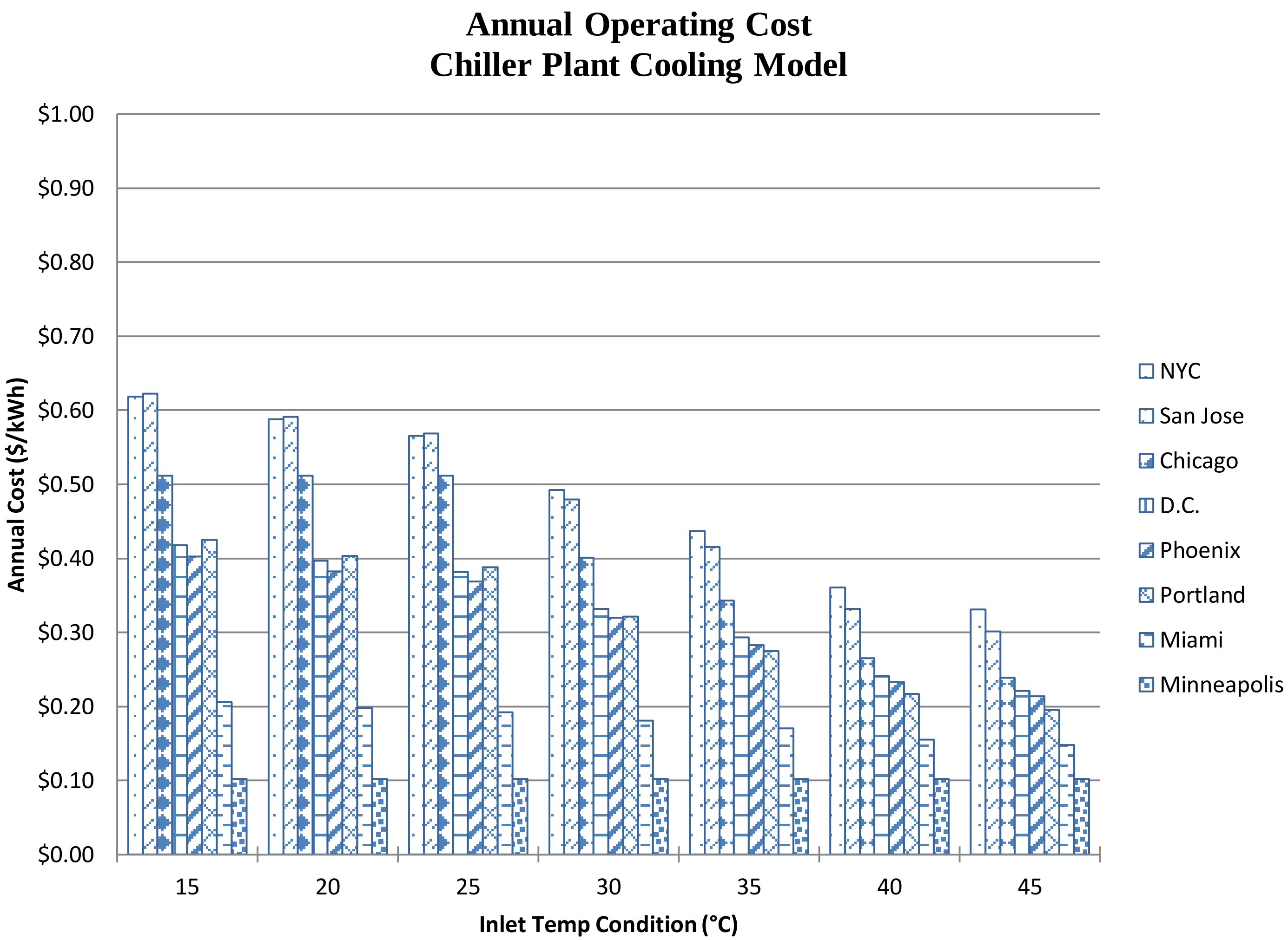


Figure 6.7 Annual operating cost for chiller model at various rack-inlet temperatures and geographic regions.

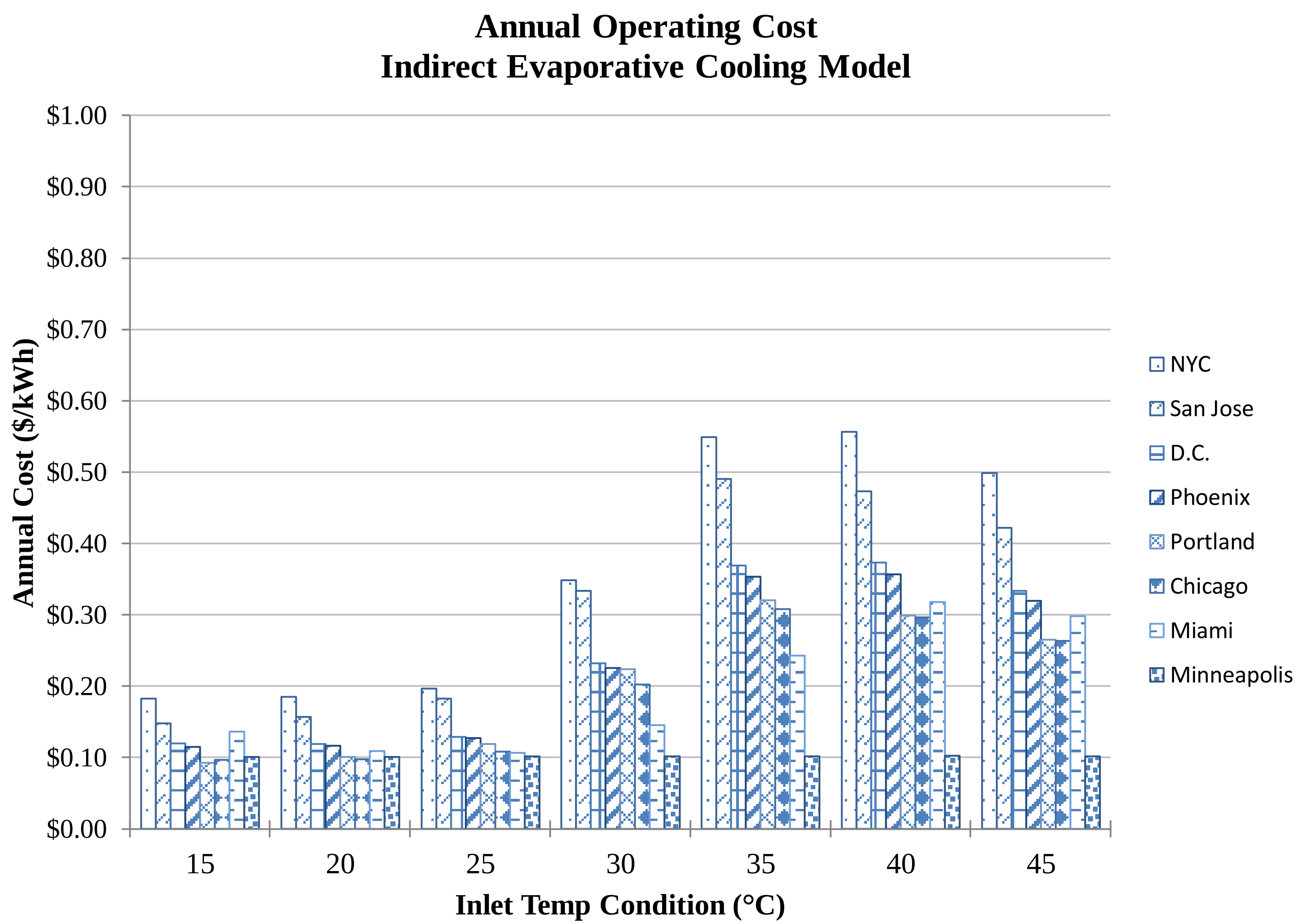


Figure 6.8 Annual operating cost for Indirect Evaporator Model at various rack-inlet temperatures and geographic regions.

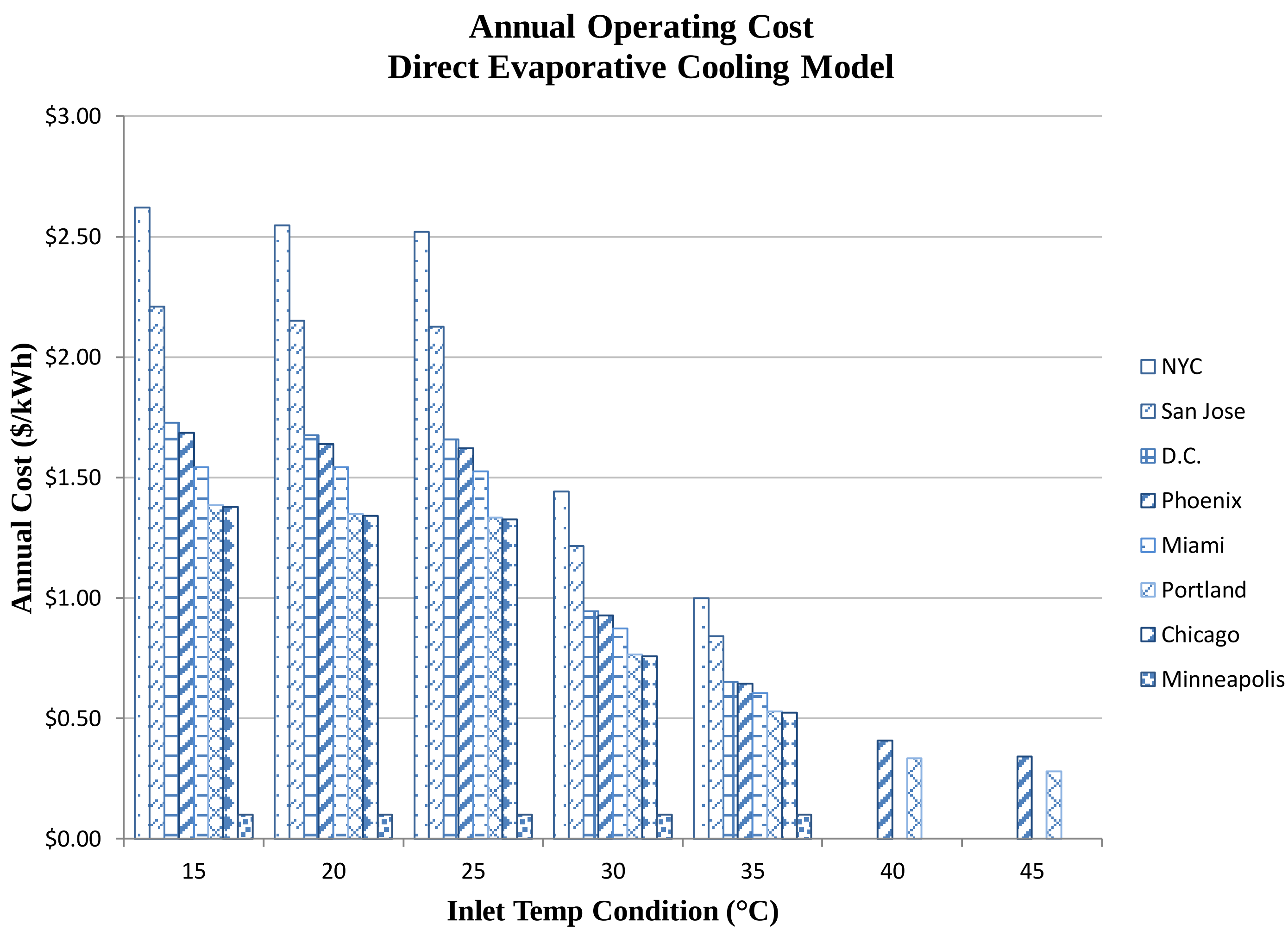


Figure 6.9 Annual operating cost for Direct Evaporator Model at various rack-inlet temperatures and geographic regions.

CHAPTER 7

LIFE CYCLE IMPACT ANALSYIS CASE STUDY

7.1 Overview

This study utilized the chiller plant facility described in Chapter 6 and characterized the water consumption of the cooling infrastructure over the operational life of the facility. An annual water inventory was embedded within each of the thermodynamic models. This water inventory included only the on-site water consumption due to evaporation during the cooling process. The results are presented for each cooling system in this chapter. The lifetime water inventory was conducted according to the methodology described in Chapter 5.

7.2 Model Results

7.2.1 Annual Water Inventory

The annual water consumption of each cooling system is presented in Figure 7.1 through Figure 7.3. Figure 7.2 shows that although the final cooling method is the same, the indirect evaporative facility consumes less water than the chiller plant facility. An unexpected result, which can be seen in Figure 7.3, is the significant water consumption for the direct evaporative facility sited in Phoenix relative to the other regions as well as the other cooling systems.

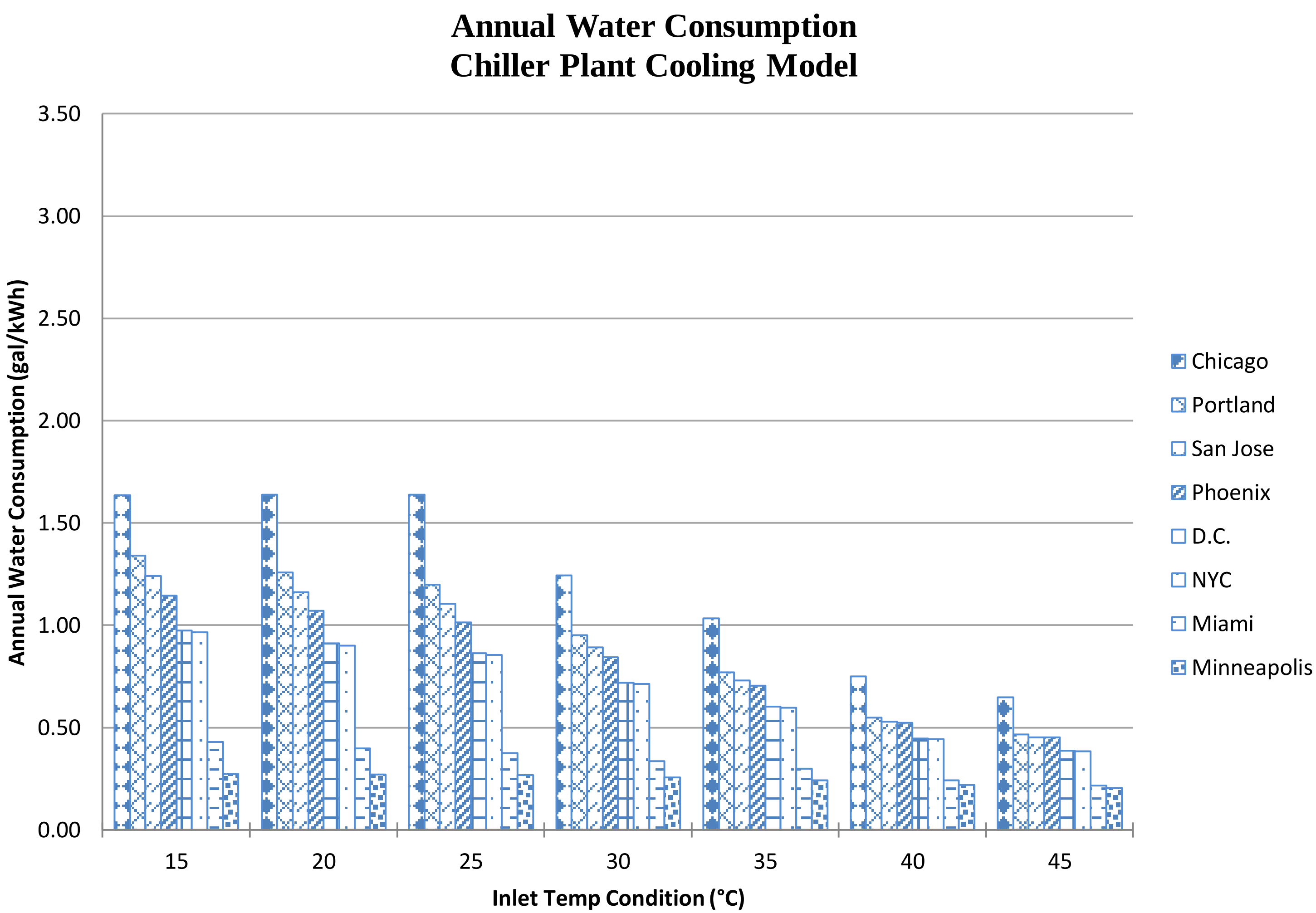


Figure 7.1 Annual water consumption for Chiller Plant Model at various rack-inlet temperatures and geographic regions.

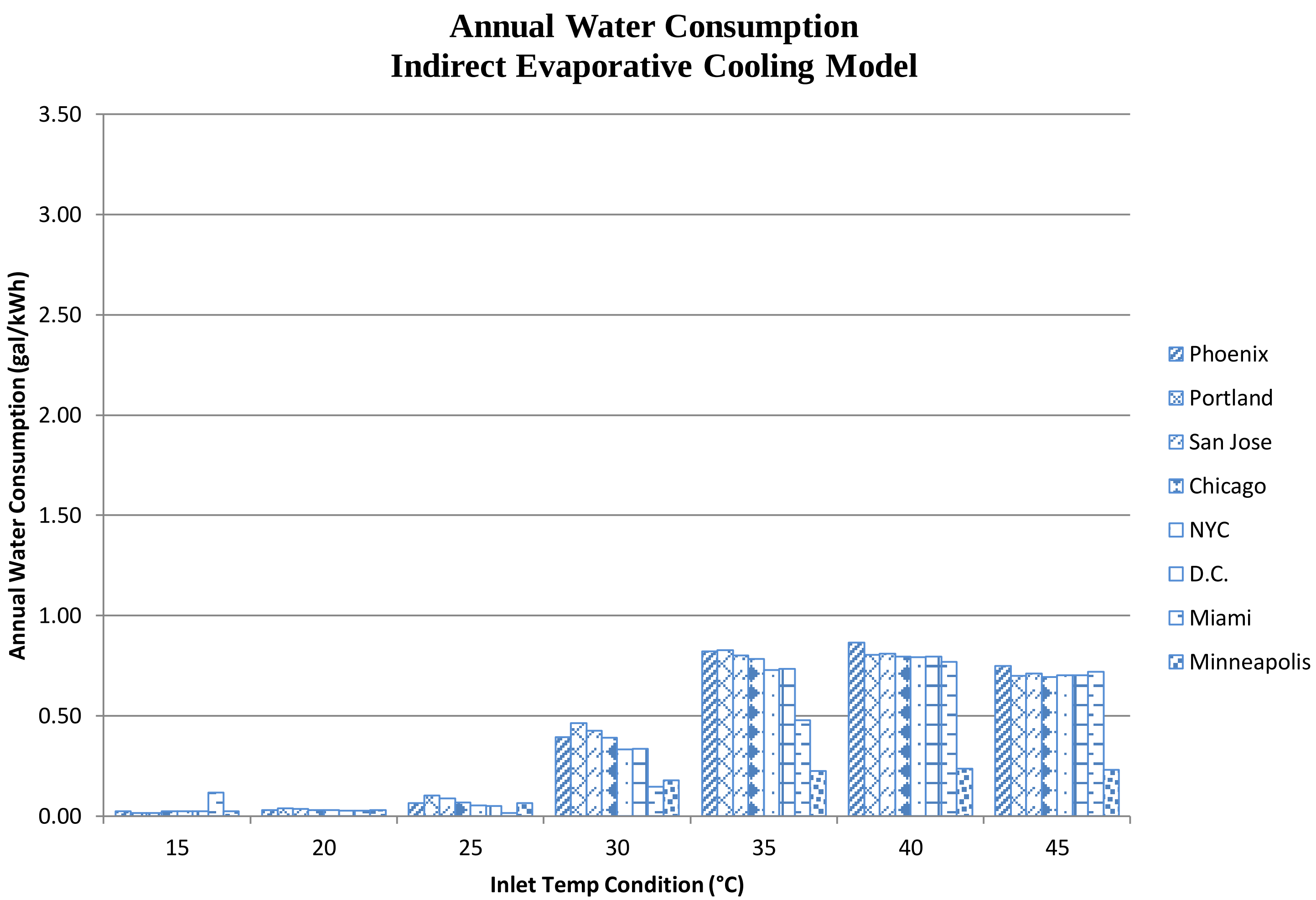


Figure 7.2 Annual water consumption for Indirect Evaporative Model at various rack-inlet temperatures and geographic regions.

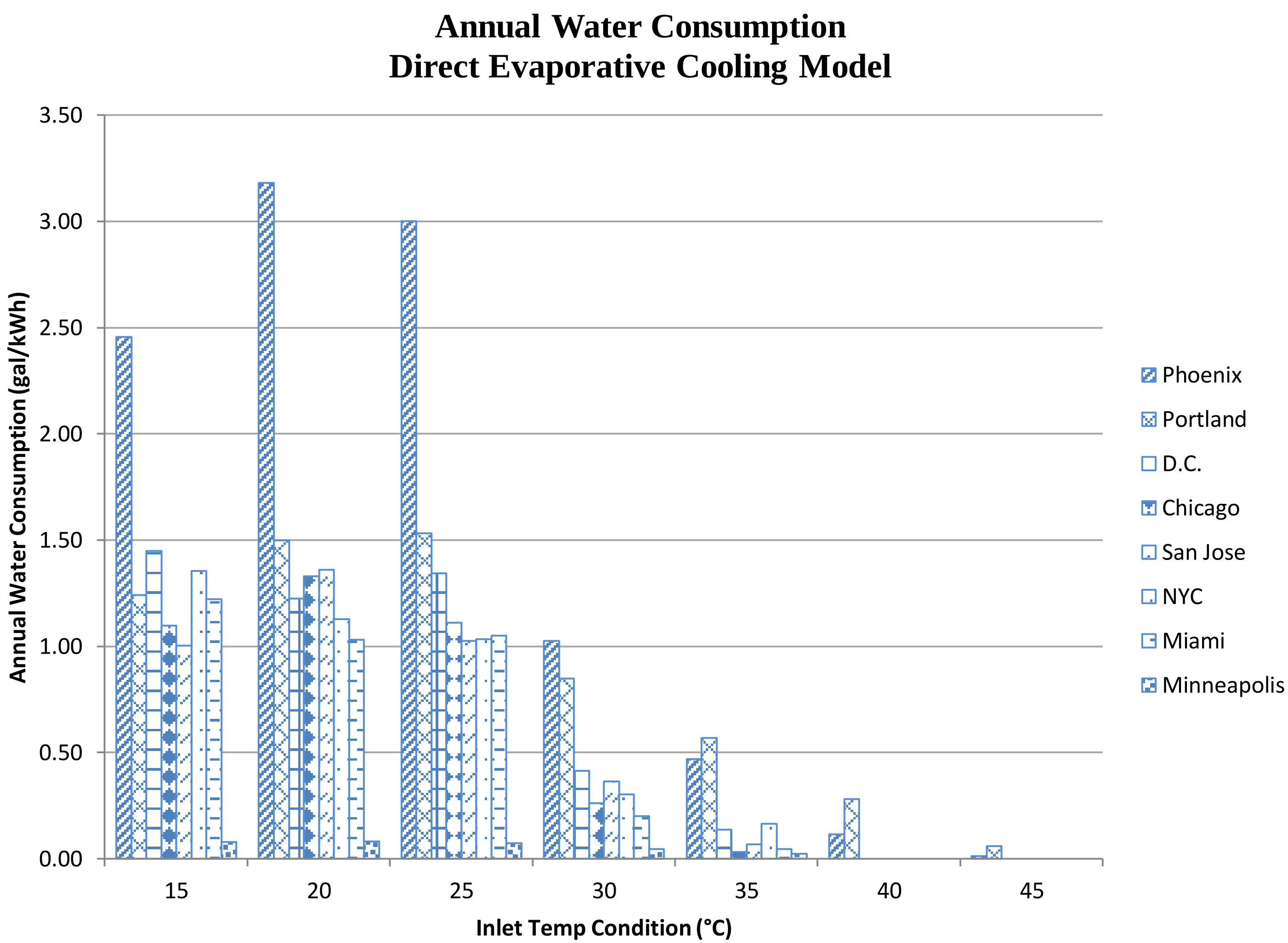


Figure 7.3 Annual water consumption for Direct Evaporative Model at various rack-inlet temperatures and geographic regions.

7.2.2 Lifetime Water Inventory

In addition to the embedded water due to power generation, the variations in component life expectancies were accounted for in the lifetime water inventory. The life expectancy of each component was taken from [65] and given in Appendix D. The operational life water inventory was calculated for each location and a summary is presented in Figure 7.4.

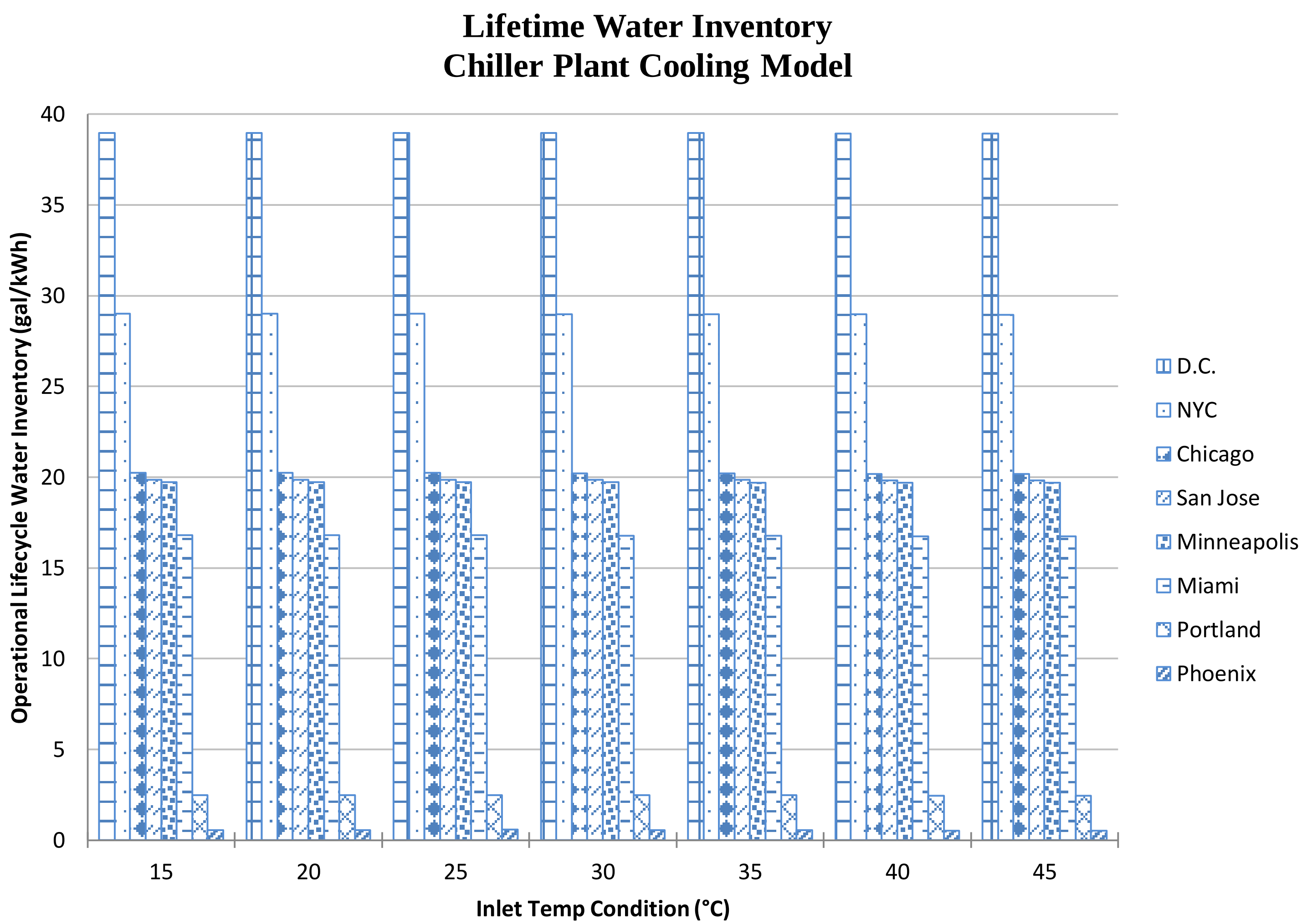


Figure 7.4 Lifetime water consumption for Chiller Plant Model at various rack-inlet temperatures and geographic regions.

CHAPTER 8

WEIGHTING THE BOTTOM LINES FOR HOLISTIC DESIGN

For true sustainable development to be achieved, each bottom line would be weighted equally for facility design and siting choices. Priorities vary from business to business. This is especially true with regard to the data center facility. No two companies are exactly the same. Factors that might influence the weighting decision are uptime, redundancy, corporate footprint, energy efficiency, and latency. Another major factor in the weighting decision is ownership of the facility. Many times the IT organization is responsible for making IT hardware decisions while the facility operations organization is responsible for paying the utility bills. Increasingly, the decision is made to turn over the operation to a third party whose priorities may be different.

There is a push today to move away from chiller-based cooling in order to reduce the energy cost and/or to improve the overall efficiency rating of the facility. The ideal climate to deploy evaporative cooling systems is a hot-dry one. Following this logic, Phoenix is the ideal candidate for siting a dedicated chiller-less system. In the following sections, Phoenix is evaluated from each of the bottom lines individually and then holistically to determine if it is the best site for evaporative cooling.

8.1 Economic Bottom Line

Referring back to Chapter 6, Figure 6.7 Annual operating cost for chiller model at various rack-inlet temperatures and geographic regions.through Figure 6.9 show that Phoenix has a moderate annual operating cost for all of the cooling systems. The annual

PUE of the direct evaporative cooling system is calculated to be 1.068, which is a very desirable PUE. This option then would be considered the best from an economic perspective.

8.2 Environmental Bottom Line

Chapter 7 shows that the annual water consumption for both the chiller plant facility and the indirect facility are reasonably close to the other regions. The water consumption, however, for the direct evaporative facility is significantly higher in Phoenix than in the other regions. This is especially true at the lower end of the operating envelope. From this perspective, the direct evaporative facility would be the best cooling infrastructure for a facility sited in Phoenix.

8.3 Societal Bottom Line

There are two ways to approach this bottom line. From the perspective of corporate responsibility, Phoenix is in a region experiencing record draught conditions. The WSI for Phoenix is calculated as 1.26, which places it in a Scarcity or overexploited ranking on the scarcity index. From the perspective of corporate risk management, placing a facility that is heavily dependent upon water for cooling operations in Phoenix is not a good choice.

8.4 Holistic Evaluation

Although Phoenix seems ideal for direct evaporative cooling from an energy perspective, this does not tell the entire story. When on-site water consumption is factored in, direct evaporative cooling becomes less attractive as a design choice. When the projected lifetime water inventory is considered along with the scarcity impact of Phoenix, direct evaporative cooling is possibly the worst of the three solutions.

CHAPTER 9

CONCLUSIONS

In this work, one holistic approach to facility cooling design was presented. This approach considered not only the operational cost of cooling the data center, the cooling PUE, or the instantaneous water consumption of a given cooling system. Rather, all of these things were considered in total. Additionally, the impacts of operating the facility across a wider range of air conditions and moving to a dedicated chiller-less system were presented. In considering the triple bottom line rather than just a single motivating priority, it was shown that there is no one best option for data center facility cooling infrastructure.

APPENDIX A

UTILITY PRICING

Table A-1 Utility cost as published by the provider by city as of October 2015.

Location	Electricity	Water
Minneapolis	\$0.10/kWh	\$0.0044/gal
New York	\$0.17/kWh	\$0.0049/gal
Chicago	\$0.09/kWh	\$0.0035/gal
Hillsboro	\$0.088/kWh	\$0.0038/gal
Phoenix	\$0.11/kWh	\$.0050/gal
Miami	\$0.098/kWh	\$0.0050/gal
San Jose	\$0.1408/kWh	\$0.0056/gal
D.C.	\$0.109/kWh	\$0.0159/gal

APPENDIX B

ANNUAL DATA COMPARISON

Table B-1 Calculated component heat load additions and instantaneous power draws in kilowatts.

			Added Heat Load				Power Draw				Work
	T inlet (°C)	Q IT (kW)	Server Fan (kW)	Blower (kW)	Building CW Pump (kW)	Cooling Tower Pump (kW)	Server Fan (kW)	Blower (kW)	Building CW Pump (kW)	Cooling Tower Pump (kW)	Chiller (kW)
	Chiller Plant										
A4 Operating Envelope	15	1033	1.22	11.19	15.79	20.68	12.22	13.80	78.95	103.41	168.34
	20	1048	1.24	13.35	16.03	20.90	12.43	13.37	80.17	104.52	172.05
	25	1057	1.26	13.09	16.17	21.03	12.65	13.12	80.85	105.14	174.10
	30	1053	2.51	28.47	16.35	21.19	25.10	73.05	81.74	105.95	177.46
	35	1056	3.76	38.30	16.55	21.37	37.55	124.35	82.74	106.87	181.64
	40	1074	7.50	73.46	17.38	22.13	75.03	238.51	86.90	110.67	199.08
	45	1078	11.25	89.89	17.73	22.45	112.50	291.86	88.64	112.25	206.33
	IDEC										
	15	1033	1.22	13.67	0.00	8.78	12.22	54.66	0.00	11.70	0
	20	1048	1.24	14.42	0.00	8.78	12.43	57.68	0.00	11.70	0
	25	1057	1.26	14.74	0.00	8.78	12.65	58.96	0.00	11.70	0
	30	1053	2.51	26.80	0.00	14.04	25.10	107.20	0.00	18.72	0
	35	1056	3.76	41.63	0.00	19.31	37.55	166.50	0.00	25.74	0
	40	1074	7.50	74.86	0.00	28.08	75.03	299.42	0.00	37.44	0
	45	1078	11.25	92.76	0.00	33.35	112.50	371.06	0.00	44.46	0
	DEC										
	15	1033	1.22	22.69	2.36	0.00	12.22	90.78	3.60	0.00	0
	20	1048	1.24	23.95	2.36	0.00	12.43	95.79	3.60	0.00	0
	25	1057	1.26	24.48	2.36	0.00	12.65	97.92	3.60	0.00	0
	30	1053	2.51	44.35	3.77	0.00	25.10	177.40	5.76	0.00	0
	35	1056	3.76	68.64	5.19	0.00	37.55	274.56	7.92	0.00	0
	40	1074	7.50	122.18	7.55	0.00	75.03	488.72	11.52	0.00	0
	45	1078	11.25	149.87	8.96	0.00	112.50	599.47	13.68	0.00	0

APPENDIX C

ANNUAL OPERATING COST

Table C-1 Annual operating cost per kilowatt-hour.

		Inlet Temp Condition (°C)						
		15	20	25	30	35	40	45
Annual Cost ($/kWh)	Zone 6A (Minneapolis)							
	Chiller	$0.102	$0.102	$0.102	$0.102	$0.102	$0.102	$0.102
	IDEC	$0.101	$0.101	$0.101	$0.102	$0.102	$0.102	$0.102
	DEC	$0.101	$0.101	$0.101	$0.101	$0.101	$ -	$ -
	Zone 5A (NYC)							
	Chiller	$0.619	$0.588	$0.566	$0.492	$0.437	$0.361	$0.331
	IDEC	$0.182	$0.185	$0.196	$0.348	$0.549	$0.557	$0.498
	DEC	$2.621	$2.546	$2.520	$1.442	$1.000	$ -	$ -
	Zone 5A (Chicago)							
	Chiller	$0.511	$0.511	$0.511	$0.401	$0.343	$0.265	$0.238
	IDEC	$0.097	$0.097	$0.109	$0.202	$0.308	$0.297	$0.263
	DEC	$1.378	$1.340	$1.325	$0.758	$0.525	$ -	$ -
	Zone 4A (Washington D.C.)							
	Chiller	$0.418	$0.397	$0.382	$0.331	$0.293	$0.241	$0.221
	IDEC	$0.120	$0.119	$0.128	$0.232	$0.369	$0.373	$0.334
	DEC	$1.726	$1.675	$1.660	$0.945	$0.653	$ -	$ -
	Zone 4C (Portland)							
	Chiller	$0.425	$0.403	$0.388	$0.322	$0.275	$0.217	$0.195
	IDEC	$0.093	$0.100	$0.119	$0.224	$0.321	$0.299	$0.265
	DEC	$1.385	$1.347	$1.334	$0.764	$0.530	$0.335	$0.280
	Zone 3C (San Jose)							
	Chiller	$0.623	$0.591	$0.569	$0.480	$0.415	$0.332	$0.301
	IDEC	$0.148	$0.157	$0.183	$0.333	$0.491	$0.473	$0.422
	DEC	$2.210	$2.150	$2.126	$1.217	$0.842	$ -	$ -
	Zone 2B (Phoenix)							
	Chiller	$0.403	$0.383	$0.368	$0.320	$0.283	$0.233	$0.214
	IDEC	$0.115	$0.116	$0.127	$0.226	$0.353	$0.357	$0.320
	DEC	$1.686	$1.638	$1.622	$0.929	$0.644	$0.407	$0.341
	Zone 1A (Miami)							
	Chiller	$0.206	$0.198	$0.192	$0.181	$0.171	$0.155	$0.148
	IDEC	$0.136	$0.109	$0.106	$0.145	$0.243	$0.318	$0.298
	DEC	$1.542	$1.542	$1.526	$0.872	$0.604	$ -	$ -

APPENDIX D

COMPONENT LIFE EXPECTANCIES

Table D-1 Component life expectancies given in years.

	Miami	Phoenix	DFW	LA	Portland, San Jose	NYC, DC	Chicago	Seattle	Minn
CRAC									
Repair	9	10	15	15	16	10	20	23	23
Replace	18	19	30	30	32	20	39	46	46
Supply Fan									
Repair	9	8	9	10	10	10	11	11	12
Replace	18	17	18	19	19	20	22	23	24
Axial Flow Fan									
Repair	24	19	13	12	11	10	9	8	8
Replace	38	32	24	22	19	20	15	15	14
Centrifugal Fan									
Repair	24	19	13	12	11	10	9	8	8
Replace	38	32	24	22	19	20	15	15	14
AHU									
Repair	9	8	9	10	10	10	11	11	12
Replace	13	13	13	14	15	15	16	17	18
Evaporative Cooler									
Replace	18	19	30	30	32	20	39	46	46
Cooling Tower									
Repair	9	10	15	15	16	10	20	23	23
Replace	13	14	23	22	29	15	32	46	46
Condenser									
Repair	9	10	15	15	16	10	20	23	23
Replace	13	14	23	22	29	15	32	46	46
Circulation Pump									
Repair	4	5	8	7	10	5	16	23	23
Replace	13	14	23	22	29	15	32	46	46
Chiller-Reciprocating Water-Cooled Hermetic									
Repair	9	10	15	15	16	10	20	23	23
Replace	18	19	30	30	32	20	39	46	46
Air Filters									
Replace	1	1	1	1	1	1	1	1	1

APPENDIX E

ANNUAL WATER INVENTORY

Table E-1 Annual water inventory.

Annual Water Consumption (gal/kWh)

	Inlet Temp Condition (°C)						
	15	20	25	30	35	40	45
Zone 6A (Minneapolis)							
Chiller	0.27	0.27	0.27	0.26	0.24	0.22	0.21
IDEC	0.02	0.03	0.06	0.18	0.23	0.24	0.23
DEC	0.08	0.08	0.07	0.05	0.02	0.00	0.00
Zone 5A (NYC)							
Chiller	0.97	0.90	0.86	0.71	0.60	0.44	0.38
IDEC	0.02	0.03	0.05	0.33	0.73	0.79	0.70
DEC	1.36	1.13	1.03	0.30	0.17	0.00	0.00
Zone 5A (Chicago)							
Chiller	1.64	1.64	1.64	1.25	1.03	0.75	0.65
IDEC	0.03	0.03	0.07	0.39	0.78	0.80	0.69
DEC	1.10	1.33	1.11	0.26	0.03	0.00	0.00
Zone 4A (Washington D.C.)							
Chiller	0.98	0.91	0.86	0.72	0.60	0.45	0.39
IDEC	0.02	0.03	0.05	0.34	0.73	0.79	0.70
DEC	1.45	1.22	1.35	0.41	0.14	0.00	0.00
Zone 4C (Portland)							
Chiller	1.34	1.26	1.20	0.95	0.77	0.55	0.47
IDEC	0.01	0.04	0.10	0.46	0.83	0.80	0.70
DEC	1.24	1.50	1.53	0.85	0.57	0.28	0.06

Table E-1 Continued

Zone 3C (San Jose)							
Chiller	1.24	1.16	1.10	0.89	0.73	0.53	0.45
IDEC	0.01	0.03	0.09	0.43	0.80	0.81	0.71
DEC	1.00	1.36	1.02	0.36	0.07	0.00	0.00
Zone 2B (Phoenix)							
Chiller	1.14	1.07	1.01	0.84	0.70	0.52	0.45
IDEC	0.02	0.03	0.07	0.39	0.82	0.87	0.75
DEC	2.46	3.18	3.00	1.03	0.47	0.12	0.01
Zone 1A (Miami)							
Chiller	0.43	0.40	0.38	0.34	0.30	0.24	0.22
IDEC	0.12	0.03	0.01	0.15	0.48	0.77	0.72
DEC	1.22	1.03	1.05	0.20	0.05	0.00	0.00

APPENDIX F

OPERATIONAL LIFE CYCLE WATER INVENTORY

Table F-1 Lifecycle water inventory for the chiller plant cooling facility.

Operational Lifecycle Water Inventory (gal/kWh)		Inlet Temp Condition (°C)						
		15	20	25	30	35	40	45
	Zone 6A (Minneapolis)							
	Chiller	19.73	19.73	19.73	19.72	19.71	19.69	19.68
	Zone 5A (NYC)							
	Chiller	29.00	29.00	29.00	28.99	28.98	28.96	28.95
	Zone 5A (Newark)							
	Chiller	33.20	33.20	33.20	33.19	33.18	33.16	33.15
	Zone 5A (Chicago)							
	Chiller	20.23	20.23	20.23	20.22	20.21	20.19	20.18
	Zone 4A (Washington D.C.)							
	Chiller	38.98	38.98	38.98	38.97	38.96	38.94	38.93
	Zone 4C (Portland)							
	WU	2.49	2.49	2.49	2.48	2.47	2.45	2.44
	Zone 3C (San Jose)							
	WU	20	20	20	20	20	20	20
	Zone 2B (Phoenix)							
	WU	1	1	1	1	1	1	1
	Zone 1A (Miami)							
	Chiller	16.79	16.79	16.79	16.78	16.77	16.75	16.74
	Zone (Los Angeles)							
	Chiller	19.87	19.87	19.87	19.86	19.85	19.83	19.82
	Zone (Las Vegas)							
	Chiller	1.64	1.65	1.65	1.63	1.62	1.60	1.59

Table F-1 Continued

	Zone (Denver)							
	Chiller	1.16	1.16	1.16	1.15	1.14	1.12	1.11
	Zone (Salt Lake City)							
	Chiller	1.06	1.06	1.06	1.05	1.04	1.02	1.01
	Zone (Dallas)							
	Chiller	15.18	15.18	15.18	15.16	15.15	15.13	15.12
	Zone (San Antonio)							
	Chiller	15.18	15.18	15.18	15.17	15.16	15.14	15.13

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BIOGRAPHICAL INFORMATION

Marianna Louisa Vallejo received her bachelor degree (B.Sc.) in Mechanical Engineering from the University of Texas at Arlington in 2010. In January 2011, she began her graduate studies with the support of the Louis Stokes Alliance for Minority Participation Bridge-to-Doctorate Fellowship. Marianna's research focused on thermal design and management of IT equipment, facility cooling solutions, and sustainable development for the built environment. Marianna served as the President of the Graduate Student Senate from 2011 to 2014 and the student branch of the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) from 2013 to 2015. She was awarded the ASHRAE College of Fellows Graduate Fellowship in 2015. Marianna received her Ph.D. degree in Mechanical Engineering from the University of Texas at Arlington in December 2015.