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Energy and Exergy Analysis of Data Center Economizer Systems

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ABSTRACT

ENERGY AND EXERGY ANALYSIS OF DATA CENTER ECONOMIZER SYSTEMS

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Electrical consumption for data centers is on the rise as more and more of them are being built. Data center owners and operators are looking for methods to reduce energy consumption and electrical costs. One method of reducing facility costs for a chilled water plant is by adding an economizer. Most studies concerning economizer systems are conducted largely by looking at energy alone since the primary focus is reducing electrical costs. Understanding how much exergy is destroyed, where it is destroyed, and why it is destroyed provides a more complete view on how environmental impacts can be minimized while reducing energy usage.

The purpose of this study is to develop energy and exergy-based models of the most common economizer systems. A normal chiller plant without an economizer and a chiller plant with an indirect wet-side economizer (the most common type of economizer system) are compared. Results show outdoor conditions influence facility energy consumption and exergy destruction. For a chiller plant operating with an economizer, the CRAH is found to be the largest source for exergy destruction. For a chiller plant without an economizer, the chiller is the largest source for exergy destruction.

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NOMENCLATURE

A	Cross sectional area of fill pack (m^2)
AHU	Air handler unit
BHP	Brake horsepower
CD	Condensate drain
CFM	Cubic feet per minute (ft^3/min)
CHWR	Chilled water return
CHWS	Chilled water supply
COP	Coefficient of performance (W/W)
CRAC	Computer room air conditioner
CRAH	Computer room air handler
CT	Cooling tower
CWR	Condenser water return
CWS	Condenser water supply
CV	Control valve
Cp	Specific heat (btu/lbm-R) or (kJ/kg-K)
$c_{p,a}$	Specific heat of dry air at constant pressure (kJ/kg-K)
$c_{p,v}$	Specific heat of water vapor at constant pressure (kJ/kg-K)
$c_{p,w}$	Specific heat of water at constant pressure (kJ/kg-K)
dh	Differential enthalpy (kJ/kg)
dH	Differential height of cooling tower fill (m)
DWE	Direct wet-side economizer
ESP	External static pressure
EWT	Entering water temperature
Ex	Exergy
FHP	Fan horsepower
G	Dry air mass flow rate (kg/s)
H	Height of cooling tower fill pack (m)
h	Enthalpy (kJ/kg)
$h_{g,w}$	Enthalpy of water vapor (kJ/kg)
$h_{f,w}$	Enthalpy of liquid water (kJ/kg)
HUW	Humidification water
HWR	Hot water return
HWS	Hot water supply
HVAC	Heating ventilation air-conditioning
IW	Industrial water
IWE	Indirect wet-side economizer
Ka	Tower characteristic (kg/m^3-s)
kW/ton	Power consumption per ton of useful refrigeration (kW/Ton)
LWT	Leaving water temperature
L	Water mass flow rate (kg/s)

Le_f Lewis factor
LCHWR Low temperature chilled water return
LCHWS Low temperature chilled water supply
MAH Makeup air handler
ME Mechanical efficiency
MUA Makeup air handler, provides fresh air and pressurization
OAE Outside air economizer
P Pressure (psia)
Q_{cond} Rate of heat transfer at the condenser (Btu/hr)
Q_{evap} Rate of heat transfer at the evaporator (Btu/hr)
Q Energy of heat transfer (kW)
Q_{evap} Evaporator cooling load(kW)
$Q_{leak,eqv}$ Chiller evaporator heat leak (kW)
R Thermal resistance (K/kW)
RA Return air
RAH Recirculation air hander or return air handler
RH Relative humidity (%)
$s_{f,w}$ Entropy of saturated liquid water (kJ/kg-K)
SA Supply air
SHR Sensible heat ratio
T Temperature (℃ or K)
T_{cond}^{in} Condenser inlet water temperature (K)
T_{cond}^{out} Condenser outlet water temperature (K)
T_{db} Dry bulb temperature (℃)
T_{evap}^{in} Chiller evaporator inlet water temperature (K)
T_{evap}^{out} Chiller evaporator outlet water temperature (K)
T_{wb} Wet bulb temperature (℃)
TP Total pressure
V Volume of cooling tower fill pack (ft^3 or m^3)
V Volume of tower (m^3)

Greek Symbols

$\dot{X}_{in}$ Exergy rate entering system (kW)
$\dot{X}_{out}$ Exergy rate leaving system (kW)
η_{II} Second law efficiency
Ω Humidity ratio ($lb_{m,w} / lb_{m,a}$)
ΔS_T Entropy generation factor (kW/K)
Φ Relative humidity (%)
ψ Stream flow exergy
ω Humidity ratio
$\omega_{s,w}$ Humidity ratio of saturated water vapor elevated at water temperature

Subscripts

a	Air
cd	Condensate drain
cond	Condenser
cw	Condenser water
cwr	Condenser water return
cws	Condenser water supply
des	Destroyed
db	Dry-bulb
e	Exit state
evap	Evaporator
ex	Exergy
f	Fluid or liquid state
fg	Change of state from fluid to gas or vapor
i	Inlet
in	Inlet
k	Boundary
L	Latent heat component
m	Mass (lbm)
lbm	Pound mass
out	Outlet
OSA	Outside air
s	Saturated
s	Sensible heat component
T	Total heat
w	Water
wb	Wet-bulb
v	Gas or vapor state

Superscripts

in	Inlet
out	Outlet

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CHAPTER 1 INTRODUCTION

1.1 Motivation

According to an Environmental Protection Agency (EPA) report on energy efficiency in data centers, their electrical power demand could double from 2007 to 2011 in the United States (United States Environmental Protection Agency, 2007). The report indicates that electrical demand for data centers in 2006 was 1.5 percent of the total of all electrical demand in the US and will increase to 2.5 percent. The EPA report emphasizes that there are opportunities for data centers to improve their efficiencies, both on the facility and server sides of data center infrastructure. The average data center facility will use about 0.83 Watts of power for facility infrastructure, including cooling, for every 1 Watt of critical information technology (IT) power demand (Greenberg, 2007; United States Environmental Protection Agency, 2007). Chiller systems are typically the largest consumer of electrical demand, second only to critical IT demands (Koomey, 2004). The use of economizer systems is one method to reduce energy consumption for data centers by significantly lowering cooling cost.

The purpose of this study has been to develop energy and exergy-based models of one of the most common economizer systems. The models incorporate weather bin data, which will allow users to determine the energy and cost savings for the most common of economizer system for their locale. The models can also be used to determine which components result in the most exergy losses, allowing researchers to better focus their efforts to improve these systems. They can also be used to analyze the electrical energy and exergy savings under a variety of conditions such as raising the data center supply or return air temperatures or changing the cooling load. In this paper, a normal chiller plant without an economizer and a chiller plant with an indirect wet-side economizer (the most common type of economizer system) are compared.

1.2 Literature Review

The simulation of cooling system performance in the past was largely energy based. Now studies are being published that perform exergy-based analysis to determine maximum efficiency and evaluate the quality of energy conversions (Harutunian, 2003; Liu, 1994; Paulus, 2000; Wang, 2005; Wu, 2004). An exergy analysis (also called availability analysis) determines the maximum useful work than can result when a system goes through a process between two specific states or the minimum required for cooling between two states. Applying exergy balances to a system allows for a direct comparison of the amount of work potential supplied to the amount of that has been consumed (Kotas, 1995). A measurement of exergy destruction allows one to determine the work potential destroyed by each system or component due to irreversibility.

A significant amount of work has been published utilizing exergy analysis to evaluate heating ventilation air-conditioning (HVAC) system performance for general components and configurations. Reference texts have been published on the exergy method (Bejan, 2006; Bejan, Tsatsaronis, Moran, 1996) and evaluation of thermal plant efficiencies (Kotas, 1995). Moist air exergy balances and efficiency relationships have been derived for common air-conditioning processes (Dincer, Hussain, Zubair, 2004). The amount of potential energy savings from moist air in evaporative cooling (cooling towers) has been studied using the exergy method (Li, Ren, Tang, Zhang, Yang, 2001). Detailed exergy analyses have been performed on evaporative heat exchangers as a function of varying outside air conditions (Dincer, et al., 2004; Muangnoi, Asvapoositkul, Wongwises, 2007, 2008; Nianping, Chengqin, Guangfa, 2002; Qureshi, Zubair, 2003, 2006, 2007). Complete thermodynamic cycles of several types of chillers have been analyzed using exergy methods (Chen, Su, 2005, 2006, 2007; Xianguo, Guoyuan, 2007).

Inside the data center, thermal management systems have been investigated using an exergy analysis to identify local and overall inefficiencies (Shah, Carey, Bash, Patel, 2004, 2008). A case study on exergy-based optimization control strategies for computer room air conditioners (CRAC) has been compared with experimental data to show the method may improve air cooling efficiencies inside the data center (Shah, Carey, Bash, Patel, 2005a, 2005b).

Little or no work has been published that analyzes and simulates economizers for data centers using the Second Law of Thermodynamics, and more explicitly, exergy analysis. Furthermore, a sensitivity analysis to determine the effect of operating conditions, such as weather and data center cooling loads, on exergy destruction and energy usage has not been done. It has been the intent of this research to advance exergy-based analysis for economizers systems by developing specific mathematical models for data center HVAC systems. Figure 1.1 is a simple facility diagram used for this study, it represents a chiller plant without any economization features. Figure 1.2 is a simple facility diagram of a chiller plant with an integrated wet-side economizer.

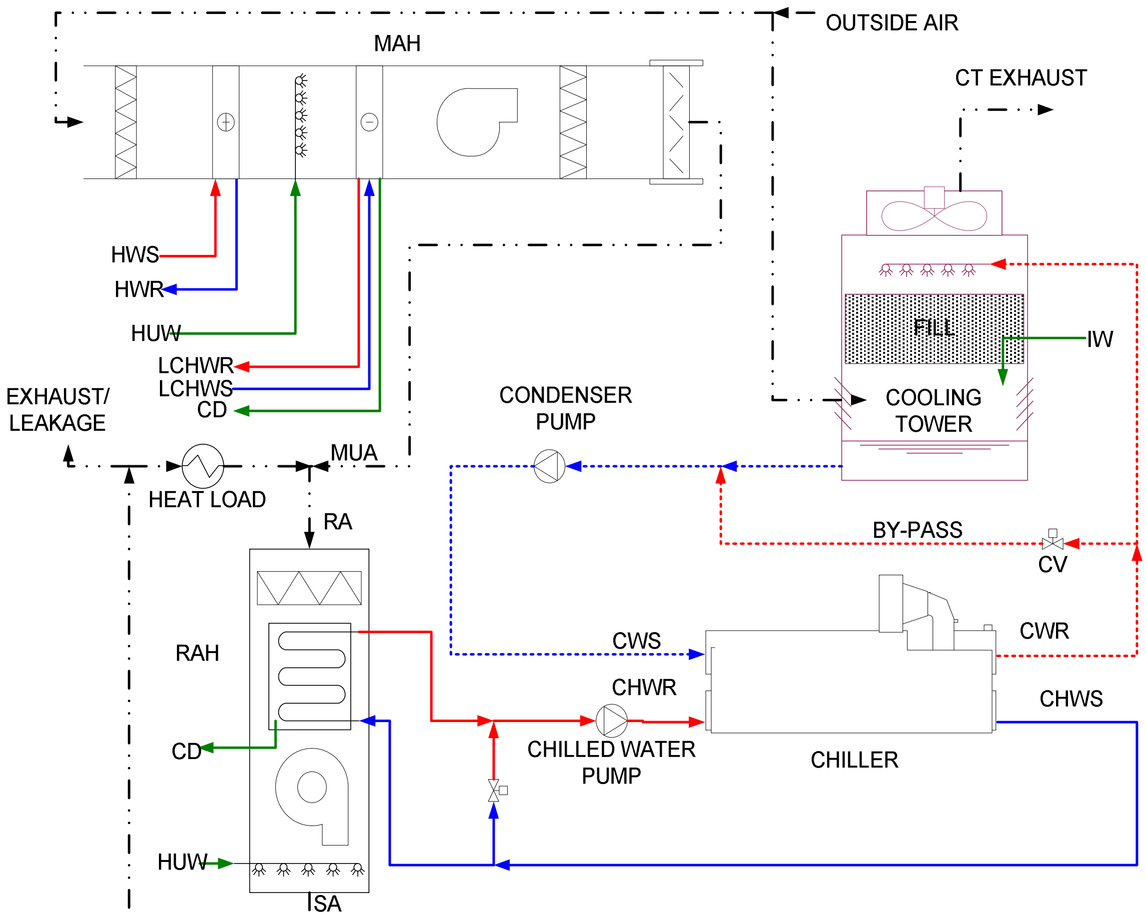


Figure 1.1: Water cooled chiller plant

For this type of integrated wet-side economizer in Figure 1.2, the heat exchanger indirectly produces chilled water or pre-cool chilled water before reaching the chiller. As servers produce heat in the data center, a return air handler (RAH) removes the heat and then rejects the unwanted heat to the chilled water system. The RAH performs many useful functions to maintain the environmental conditions inside the data center, such as filtering air contaminants, latent and sensible cooling, moving the air, and humidity controls. The chilled water is moved by a chilled water pump through the heat exchanger, water-cooled chiller and back to the RAH.

The condenser water loop has a pump that moves condenser water from the cooling tower to the heat exchanger. If the condenser water returning from the cooling tower is colder than the chilled water returning from the RAH, the heat exchanger provides a means to transfer heat to the condenser water while bypassing and therefore avoiding mixing the two water streams. Open cooling tower water is generally considered dirty because it is exposed to outdoor elements that may contain solids that cause fouling in RAH heat exchangers coils and chiller bundle tubes. In general, open loop cooling tower water is not allowed to mix with a closed loop chilled water system because of this fouling and other operational issues. The heat that the heat exchanger removes ultimately reduces the cooling load on the chiller.

The chiller utilizes a vapor refrigeration cycle to remove heat from the chilled water and then transfers the heat to the condenser water loop. Pumps move the condenser water to the cooling tower where it can then be rejected to the outside air by means of evaporative cooling. The warm condenser water cools down and then returns back to the heat exchanger and chiller. These systems operate together to remove heat from the data center to the outside air. The wet-side economizer feature, utilizing a heat exchanger to pre-cool warm chilled water return, reduces the amount of cooling required from the chiller. The make-up air handler (MAH) in Figure 1.2 provides fresh air to the data center. The MAH can also be used to positively pressurize the data center relative to other areas in the building. Dust and air infiltration is minimized. Since the MAH brings in outside air, it has components to condition the outside air before it is allowed to mix with inside air. These components provide filtration, heating, humidification control, cooling, and moving the air.

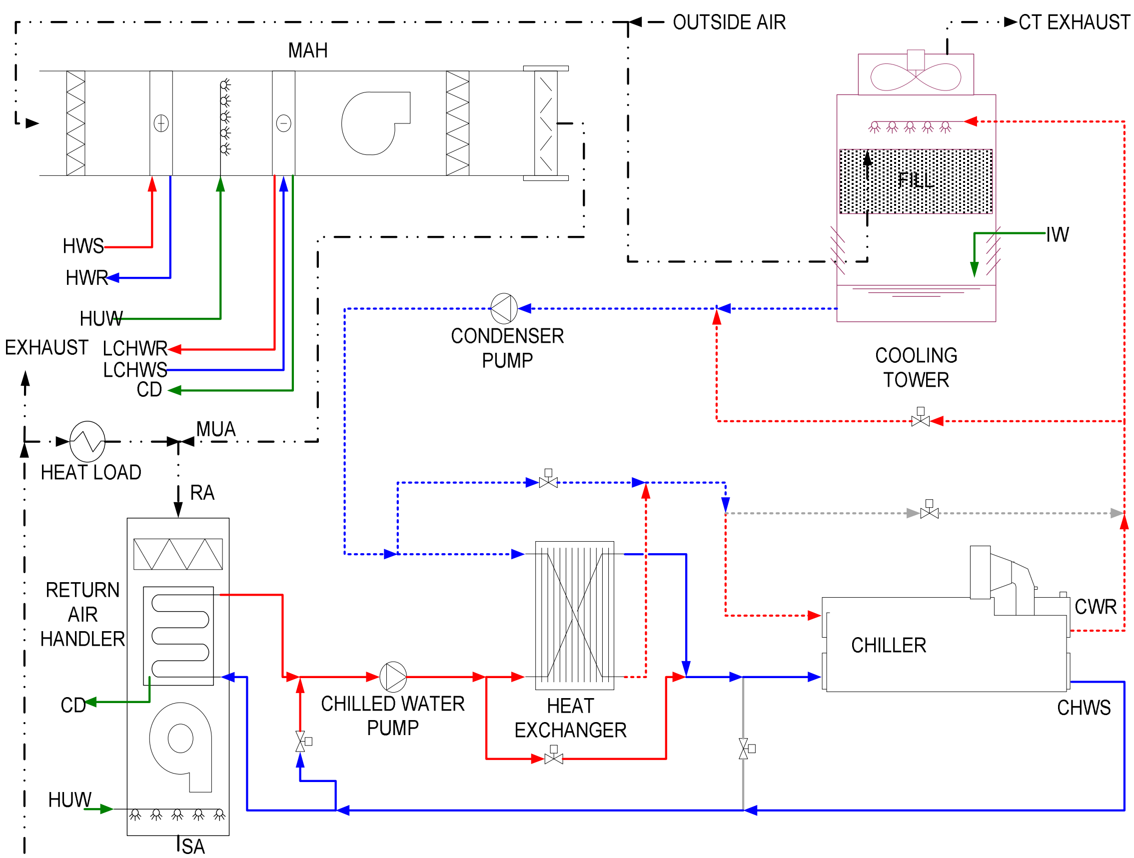


Figure 1.2: Integrated indirect wet-side economizer

Economizers, or “free-cooling,” are mechanical systems that save energy by reducing the amount of refrigeration compressor work required to provide cooling when outside air conditions are favorable. The energy consumed for a waterside economizers, only pumps and fan energy, is 15 to 20 percent of the energy required from refrigeration cooling (Standford, 2003). The use of economizers to save energy for various cooling applications is not new, and several case studies have been published (Imperatore, 1975; Starr, 1984; Telecky, 1985; Tobias, Schade, 1976; Zmeureanu, 1988). The use of economizers (free cooling) and raising supply or return air temperatures are only two of many possible methods of reducing the cooling costs for data centers (Garday, 2007; Kurkjian, Glass, Routsen, 2007; Schmidt, Beaty, Dietrich, 2007; Sorell, 2007; Tschudi, Fok, 2007). These systems may use outside air, direct wet-side evaporative heat exchangers, or indirect wet-side chilled water loops for cooling (Taras, 2005). When it is more economical to bring in fresh supply air rather than to cool the hot return air, the outside air economizer (OAE) uses partial to 100% outside air to provide cooling. In both direct wet-side (DWE) and indirect wet-side economizer (IWE) systems, a cooling tower can directly provide the production of chilled water when the outdoor wet bulb temperature is below the desired chilled water supply set point temperature. The DWE system circulates the cooling fluid directly to the cooling tower and back to the air handler, which directly or partially produces chilled water. The IWE system uses an intermediate heat exchanger between the cooling tower, chiller, and air handler, which indirectly produces chilled water. The IWE can pre-cool chilled water return prior to reaching the chiller to save energy or operate in 100 percent free-cooling mode that bypasses the chiller completely. Figure 1.3 illustrates the OAE, DWE, and IWE typical system configurations (Taras, 2005).

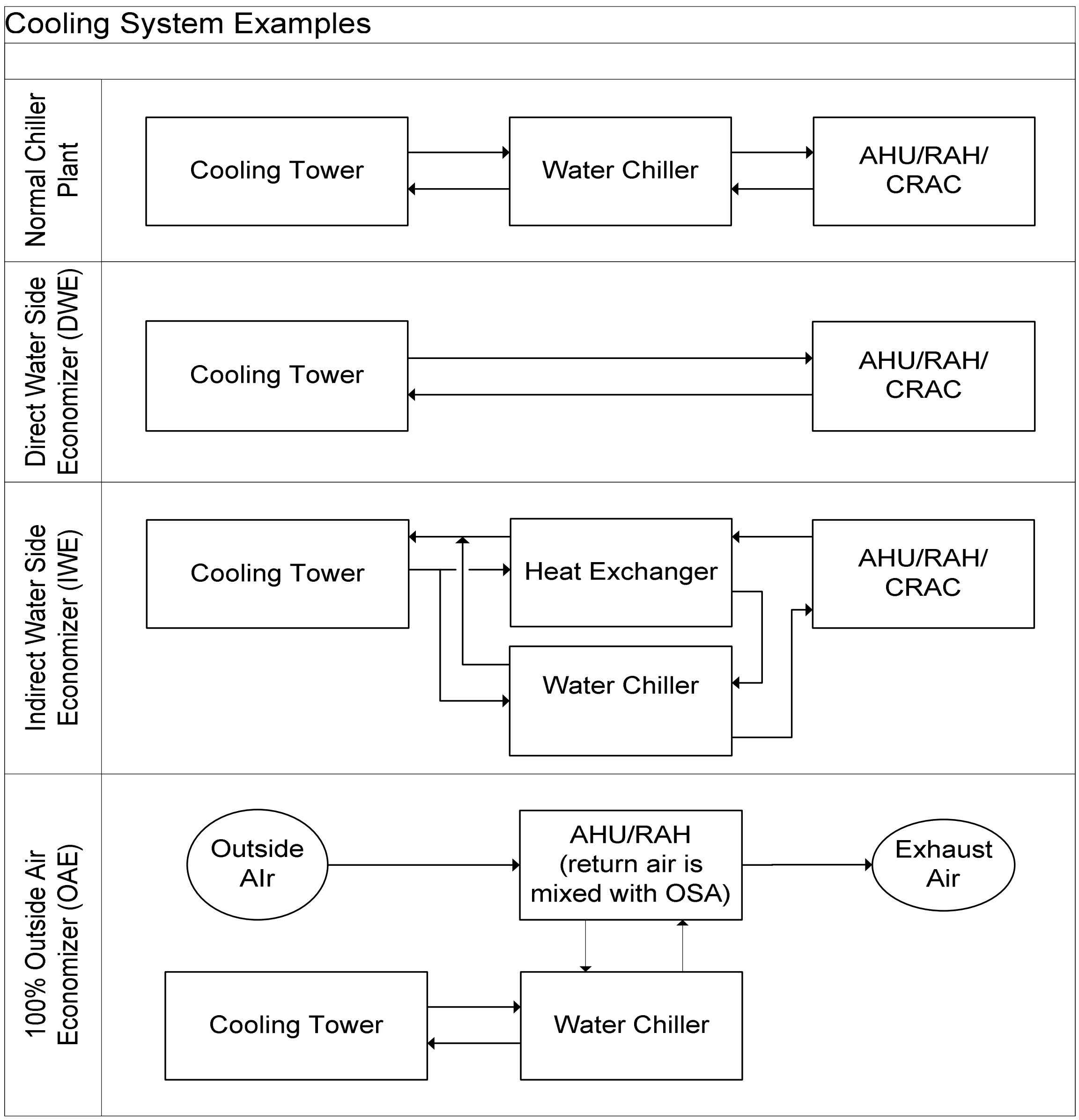


Figure 1.3: Cooling system examples for a generic chiller plant and different types of economizers

The use of these economizing systems is gaining traction worldwide because case studies have shown them to be economical (Fisk, Seppänen, Faulkner, Huang, 2005; Garday, 2007; Taras, 2005). Rising electrical rates, higher heat load densities, increasing cooling requirements, and energy conscious rebate programs are improving the payback on capital outlays required to install potentially more complicated facility systems (United States Environmental Protection Agency, 2007). Some local and state governments have adopted energy codes that require the use of economizers (Department of Planning and Development, 2006; Oregon Department of Energy, 2007).

The simulations of cooling systems' performance have been largely energy based, and now studies are being published that perform exergy-based analysis to determine maximum efficiency and evaluate the quality of energy conversions. Exergy is defined as "the maximum useful work than can be obtained as a system undergoes a process between two specific states" (Cengel, Boles, 2006). Availability analysis is another name commonly used to describe exergy analysis. Applying exergy balances to a system allows for comparison of direct measurement of the amount of work potential supplied to the amount of that has been consumed (Kotas, 1995). Exergy analysis measures the amount of work potential or the quality of different forms of energy relative to the environment. It is also used for designing, improving, and optimizing thermal fluid system designs.

A significant amount of work has been published utilizing exergy analysis to evaluate heating ventilation air-conditioning (HVAC) system performance for general components and configurations. Reference texts have been published on the exergy method (Bejan, 2006; Bejan, et al., 1996) and evaluating thermal plant efficiencies (Kotas, 1995). Moist air exergy balances and efficiency relationships have been derived for common air-conditioning processes (Kanoglu, Dincer, Rosen, 2007). Exergy analysis has been conducted on evaporative heat exchangers, also known as cooling towers (Qureshi, 2004). The amount of potential energy savings from moist air in evaporative cooling has been studied using exergy method (Li, et al., 2001). Detailed chiller exergy analyses have been performed on evaporative heat exchangers as a function of varying outside air conditions (Muangnoi, et al., 2007; Naphon, 2005; Nianping, et al., 2002; Qureshi, 2004; Qureshi, Zubair, 2007). Others have studied the refrigeration cycle in detail for different types of configurations. The vapor compression refrigeration plant cycle has been analyzed by trending compressor speeds and selecting different types of refrigerants (Aprea, Rossi, Greco, Renno, 2003). Complete thermodynamic cycles of several types of chillers have been analyzed using exergy methods (Chen, Su, 2005; Tsaros, 1987; Tschudi, Fok, 2007; Xianguo, Guoyuan, 2007). A modified coefficient of performance (COP) has been developed to be exergy based (Hasabnis, Bhagwat, 2007).

Inside the data center, thermal management systems have been investigated using exergy analysis to identify local and overall inefficiencies (Shah, Carey, Bash, Patel, 2003). Case studies on exergy-based optimization control strategies for computer room air conditioners (CRAC) have been compared with experimental data to show the method may improve air cooling efficiencies inside the data center (Shah, et al., 2004). Post-processing code for computational fluid dynamics (CFD) models have been created to study exergy and thermal performance in data center applications (Shah, et al., 2004). However, little work has been published that analyzes and simulates data center specific economizer systems as shown in Figure 1.2 using the 2^{nd} Law of Thermodynamics, and more explicitly, exergy analysis. Furthermore, economic sensitivity analysis of Second Law efficiencies by varying operating conditions, such as weather and varying data center cooling loads, have not been well documented. It is the intent of this thesis to advance exergy-based analysis for economizers systems by developing specific mathematical models for data center HVAC systems.

CHAPTER 2 METHODOLOGY

2.1 Simulation Overview

Each mechanical component is simulated using mass, energy, entropy, and exergy balances to show how it would perform under varying conditions. Each mechanical component is modeled as a steady-state module that produces output states with given inlet conditions, e.g., air handlers, chillers, coils, cooling towers, heat exchangers, pumps. The components are linked together and function as a complete thermal system by connecting the states through an airflow path or piping distribution. This allows each component to react to outside air conditions and with other mechanical equipment in the facility system, much as they would function in a true facility.

The main inputs to the simulation model are as follows: historical hourly weather bin data, data center heat load, operating set points, and performance characteristics of each mechanical component, such as fan and pump curves. Since temperature and humidity both must be controlled in data centers, all analyses involve moist air. Kanoglu and colleagues documented sensible cooling and heating, heating with humidification, cooling with dehumidification, evaporative cooling, and adiabatic mixing processes (Kanoglu, et al., 2007). Moist air is modeled as a combination of dry air and water vapor components using ideal gas laws.

The Engineering Equation Solver (EES) provides enthalpy and entropy values of moist air from a property database based on National Institute of Standards and Technology (NIST) JANAF thermo-chemical tables (Klein, 2007). EES is a simultaneous equation solver based on the Newton-Raphson method and is used for all simulations. This program is widely available, and commercial licenses are inexpensive. Component model details are discussed in the next section.

Exergy analysis requires choosing a dead state. The dead state occurs when the system is in equilibrium with the environment and serves as a reference point to calculate the work potential. Because data centers condition outside air and reject waste heat to the outside air, the dead state should be the current outdoor environmental conditions. An exergy analysis of each component is performed using standard practices as discussed by Bejan (Bejan, 2006) and Kanoglu et al. (Kanoglu, et al., 2007). The second-law efficiency of the entire data center can be defined using the sum of the rates of exergy entering and exiting each component, as shown below in Equation (2.1).

Second Law Efficiency (Exergy Efficiency):

$$\eta_{II} = \frac{\sum \dot{X}_{out}}{\sum \dot{X}_{in}} = 1 - \frac{\sum \dot{X}_{des}}{\sum \dot{X}_{in}} \qquad \text{Eq. (2.1)}$$

2.2 Cooling Tower Component

Cooling towers provide a means to reject condenser water heat to outside air. Muangnoi et al. (2007) outline a mathematical model for a counter-flow cooling tower where the water flows downward through the fill pack while air is induced upward by a fan as shown in Figure 2.1. An initial guess at leaving water mass flow rate and leaving condenser water temperature begins the calculation. The model iteratively solves the conservation of energy for air Eq. (2.2), conservation of mass of water Eq. (2.3) and conservation of energy for water Eq. (2.4) for steady state conditions by incrementally stepping through the height of the fill material. The model predicts exiting conditions to within 4 percent, or 1.5˚C, when compared to experimental results.

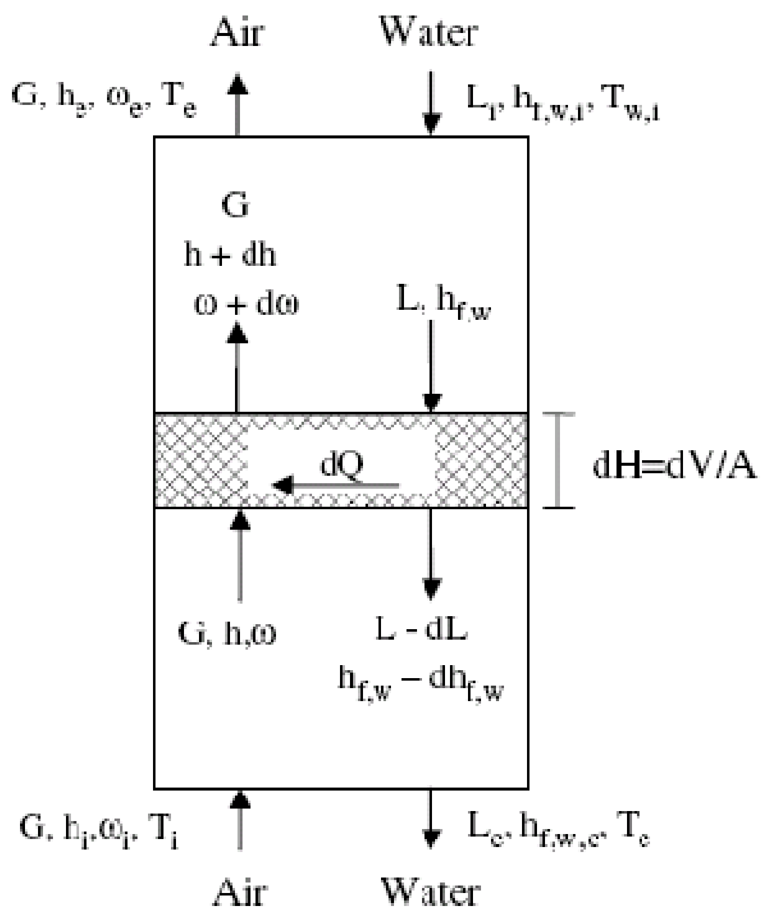


Figure 2.1: Differential element of mass and energy balance for a counter flow wet cooling tower (taken from Muangnoi et al., 2007)

$$\frac{dh}{dH} = \frac{KaA}{G}\left[Le_f c_{pa}(T_w - T) + h_{g,w}\left(\omega_{s,w} - \omega\right)\right] \qquad \text{Eq. (2.2)}$$

$$\frac{d\omega}{dH} = \frac{KaA}{G}\left(\omega_{s,w} - \omega\right) \qquad \text{Eq. (2.3)}$$

$$\frac{dT_w}{dH} = \frac{G}{Lc_{p,w}}\left(dh - h_{f,w}d\omega\right) \qquad \text{Eq. (2.4)}$$

The cooling tower model used in this study is a slightly modified version of Muangnoi's where moist airflow rate is modulated to simulate a variable speed drive fan rather than an assumed constant airflow rate. This modification allows the model to iteratively solve using different moist airflow rates until a desired leaving condenser water temperature set point is satisfied. An overall energy and mass balance is used to solve for the exit conditions, and the model iteratively solves for the exit conditions with given inlet conditions. Airflow rate is adjusted until desired condenser water temperature is achieved and energy balances are satisfied. The cooling tower characteristic parameter (Ka) is assumed to remain constant throughout the fill pack and at different inlet conditions (Muangnoi, et al., 2007, 2008). Muangnoi's results show that a constant cooling tower characteristics can be used to predict exit dry-bulb temperature to within 1.19 ℃ and exit wet-bulb tem perature to within 0.22℃ when compared to experimentally measured data (Muangnoi, et al., 2007) at varying inlet conditions. The cooling tower characteristic parameter does not change with different wet-bulb temperatures but rather with a change in the L/G ratio (water flow rate to dry air flow rate) Muangnoi's model was used to solve for the cooling tower characteristic at the summer wet-bulb design conditions and manufacturer data from GEA Power Cooling Incorporated (GEA, 2009). The Lewis factor is assumed to be unity (Muangnoi, et al., 2007). After the required air mass flow rate is determined, it is used to estimate fan brake power for the given inlet conditions. This result simulates a cooling tower with a variable speed drive (VSD) over a wide range of air inlet conditions.

If the VSD reaches 100 percent and cannot maintain leaving water temperature set point, then the leaving water temperature is adjusted upward incrementally until air velocity through the cooling tower fill pack is below 2.5 m/s. Air flow rate above 2.5m/s will exceed the brake horse power for the fan being modeled and selected cooling tower make and model. The valid range for air velocity through a fill pack is between 1.5 m/s to 2.75 m/s for the GEA counter-flow towers. This is in agreement with ASHRAE's typical limits of 300 to 700 fpm (1.5 to 3.5 m/s) (ASHRAE, 2008). Results from this modified component model are shown in Table 2.1 are nearly within +/- 0.2℃ o f Muangoi's models, which is within the uncertainty of his model when compared to experimental data.

Table 2.1: Predicted conditions for a cooling tower model

Inlet Conditions			Muangnoi et. al. Predicted Exit Air Conditions				Modified Model Predicted Exit			
$T_{db,i}$ [C]	ϕ (%) [-]	$T_{wb,i}$ [C]	$T_{db,e}$ [C]	ϕ (%) [-]	$T_{wb,e}$ [C]	G [kg/s]	$T_{db,e}$ [C]	ϕ (%) [-]	$T_{wb,e}$ [C]	G [kg/s]
32.40	40.0	21.92	34.30	98.5	34.08	0.0726	34.32	98.71	34.13	0.0713
32.40	45.0	22.98	34.08	98.0	33.78	0.0806	34.10	98.20	33.84	0.0790
32.40	50.0	23.99	33.82	97.2	33.41	0.0914	33.84	97.46	33.47	0.0894
32.40	55.0	24.96	33.51	96.0	32.92	0.1073	33.53	96.33	32.99	0.1047
32.40	60.0	25.90	33.14	94.1	32.27	0.1271	33.17	94.51	32.35	0.1297
32.40	65.0	26.81	32.73	90.8	31.36	0.1725	32.75	91.34	31.47	0.1799
32.40	70.0	27.69	32.36	84.2	29.99	0.3750	32.37	85.07	30.13	0.3470
27.00	70.0	22.78	33.83	100.0	33.83	0.0788	33.86	100.00	33.86	0.0773
28.00	70.0	23.69	33.58	99.6	33.52	0.0877	33.61	99.76	33.58	0.0859
29.00	70.0	24.60	33.28	98.2	33.02	0.1004	33.32	99.01	33.17	0.0982
30.00	70.0	25.50	32.49	97.4	32.11	0.1205	32.98	97.66	32.64	0.1174
31.00	70.0	26.41	32.58	94.6	31.79	0.1580	32.62	95.00	31.89	0.1528
32.00	70.0	27.32	32.33	88.6	30.64	0.2685	32.35	89.26	30.76	0.2449

2.3 Chiller Component

Chillers mechanically produce chilled water by removing heat from the chilled water return and rejecting it to the condenser water. The heat is removed by the cooling tower. Modeling a centrifugal chiller can quickly become complex since they typically consist of multi-stage compressors, refrigerant economizers, turning vanes, condenser and evaporator tube bundles, and temperature based throttling valves. The performance of a chiller is dependent on several operating factors such as condenser fluid temperature, refrigerant selection, and chilled water supply temperatures. Typically, negative-pressure chillers (referring to refrigerant below atmospheric pressure) operate at a peak loading of 0.5kW/ton efficiency or less as opposed to positive-pressure chillers that commonly operate at 0.55kW/ton efficiency or greater (Standford, 2003).

The capacity of the chiller is determined by a series of tests, rating requirements, and operating parameters such as 29.4 ºC (85 ºF) entering condenser water temperature and 6.7 ºC (44 ºF) exiting evaporator water temperature as shown in Table 2.2 (Air-Conditioning and Refrigeration Institute, 2003). The rated capacities found in supplier catalogs are usually certified ratings based on Air-Conditioning and Refrigeration Institute (ARI) testing requirements and not the maximum true capacity. The simulation model requires the capacity of the chiller to be specified for the given operating range.

Table 2.2: ARI standard rating conditions for a water-cooled chiller

ARI Standard Rating Conditions		
	Condenser	Evaporator
Temperature	Entering 85 ˚F	Leaving 44 ˚F
Flow Rate	3.0 gpm/ton	2.4 gpm/ton
Water-side Fouling	0.00025 hr-ft^2-˚F/Btu	0.0001 hr-ft^2-˚F/Btu

Chiller load is determined based on entering evaporator temperature and mass flow rate, and exiting chilled water set point. Chiller efficiencies may increase by 1 to 3 percent for every one degree increase of evaporator water temperature (Taras, 2005). The chiller efficiency improves when exiting chilled water evaporator temperatures can be elevated.

Performance parameters for typical centrifugal chillers can be found by curve fitting a 3rd order polynomial using Microsoft Excel as shown in Figure 2.2 and Eq

2.8 -2.9. The chart represents actual measurements from a centrifugal chiller operating with constant 21.1˚C (70 ˚F) entering condenser water and 5.6˚C (42 ˚F) exiting evaporator water. These data were the best available facility set of data to use at the time of this work. The data set included both refrigerant and water side operational data from 0 to 100 percent load. Exiting condenser water temperature and compressor power is calculated from chiller refrigeration load using Eq. (2.5) – (2.6). However, the capacity of the chiller changes with EWT and affects COP and kW/Ton performance curves. A different chiller performance model is required to adjust for varying entering condenser water temperature, exiting evaporator conditions, and cooling load dynamically. The model discussed in the next section, which is the one utilized for this work, accounts for these conditions.

$$COP = \frac{Q_L}{W} \qquad \text{Eq. (2.5)}$$

$$Q_L = \dot{m}_w c_{p,w}(EWT - LWT) \qquad \text{Eq. (2.6)}$$

$$Load = \frac{Demand}{Capacity} \qquad \text{Eq. (2.7)}$$

$$COP = 1.14\, Load^3 - 8.82\, Load^2 + 11.34\, Load + 2.26 \qquad \text{Eq. (2.8)}$$

$$\frac{kW}{Ton} = -0.66\, Load^3 + 1.98\, Load^2 - 1.85\, Load + 1.11 \qquad \text{Eq. (2.9)}$$

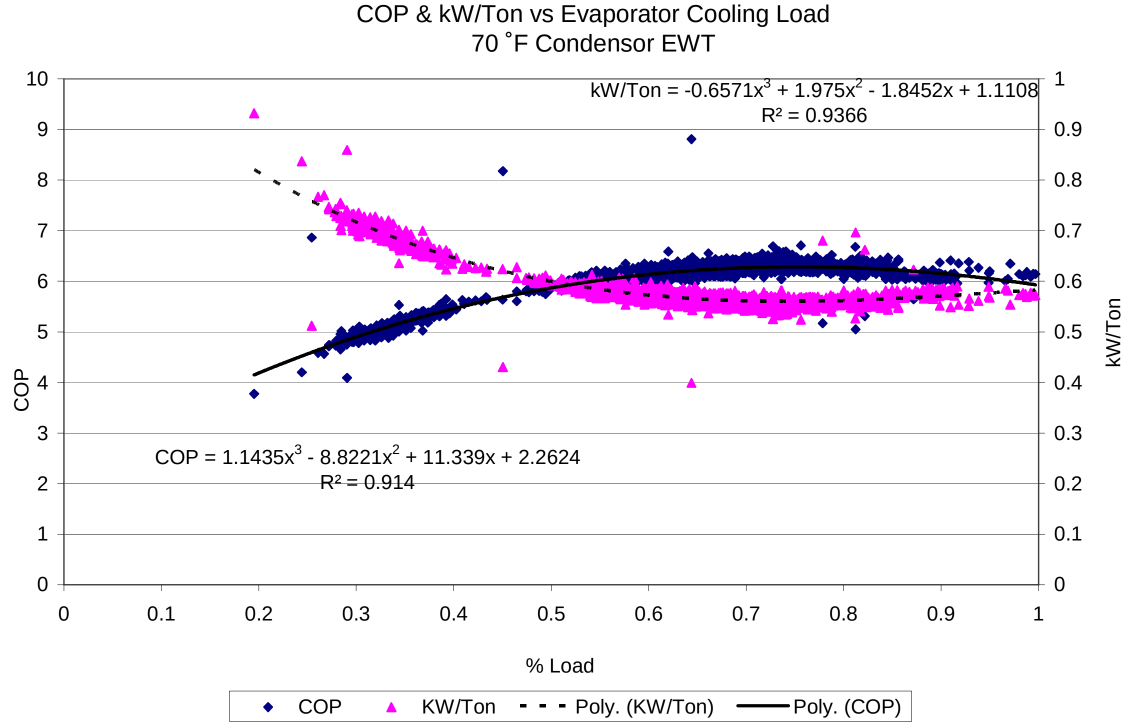


Figure 2.2: Typical chiller performance

For the simulation model, the chiller is modeled using the Simple Thermodynamic Model (STM) developed by Gordon and Ng (Gordon, Ng, 2000). The STM has been shown to predict COP within 5 percent using inputs of the inlet evaporator and condenser temperatures, the power input, and the rate of heat removal. Additional inputs include the internal entropy generation, condenser and evaporator thermal resistance, and heat leakage to the surroundings. Saththasivam and Choon (Saththasivam, Choon, 2008) have presented methods for determining these latter values for a chiller, and they have shown them to stay relatively constant for a given chiller.

Operation data were collected from a building management system (BMS) for a nominal 1,280 ton chiller with a condenser water supply temperature of 70℉, which is the same chiller used to produce Figure 2.2. STM values for a 1280 ton centrifugal chiller have been determined utilizing the same technique for curve fitting COP data and are used as inputs to the current models. Table 2.3 is the result of modeling Eq. (2.10) – (2.16) using a statistical software package, JMP (SAS, 2009), to calibrate STM values with measured chiller data. Figure 2.3 is an overlay plot of measured COP (red square) and the predicted COP (blue diamond) from the STM model. The graph shows good correlation between predicted COP by STM and the actual measured COP. The predicted COP found from the STM model is used to calculate power consumption of the chiller, Eq. (2.5) – (2.6), for varying loads. Gordon and Ng (Gordon, Ng, 2000) provides tables for other chiller sizes and types.

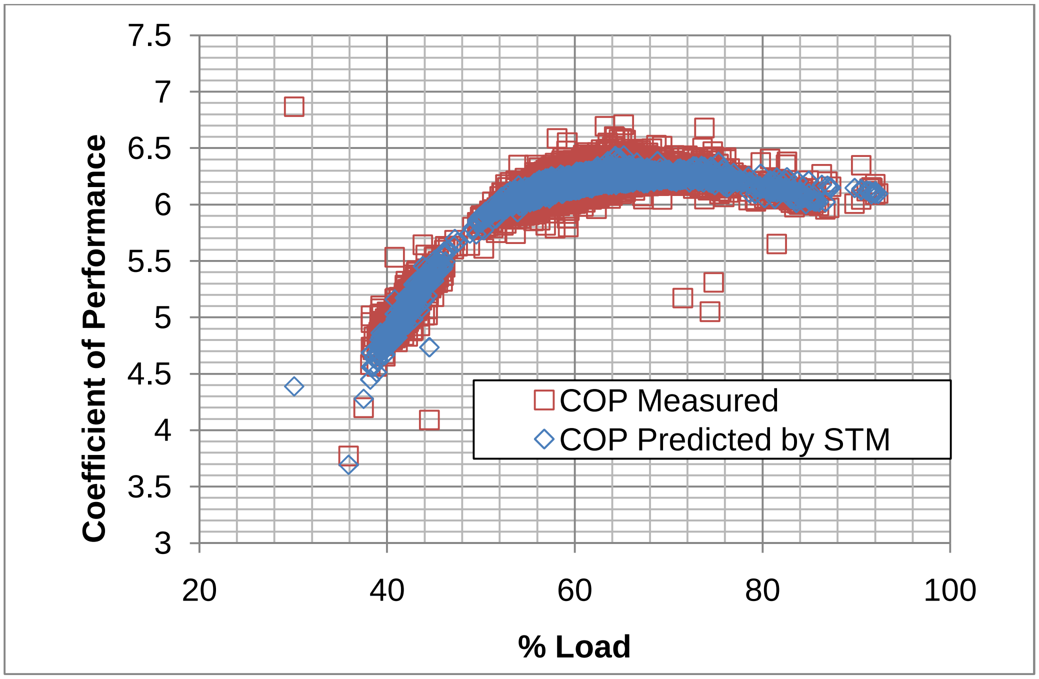


Figure 2.3: STM predicted COP verses measured COP for a 1280 ton centrifugal chiller

The STM modeling method can predict COP for several independent variables that Eq. (2.8) –(2.9) does not include. COP in Eq. (2.10) is a function of entering evaporator water temperature, entering condenser water temperature, and cooling load. A new curve would have to be generated for every possible operating condition to use Eq. (2.8)- (2.9) and would be only valid for the constant entering condenser water temperature and leaving evaporator conditions. The STM method is more applicable for this model as entering water conditions and cooling load will vary with outside air conditions.

$$\frac{T_{evap}^{in}}{T_{cond}^{in}}\left(1+\frac{1}{COP}\right)-1$$

$$= \frac{T_{evap}^{in}}{Q_{evap}}\Delta S_T + Q_{leak,eqv}\left(\frac{T_{cond}^{in}-T_{evap}^{in}}{T_{cond}^{in}Q_{evap}}\right) + R\left(\frac{Q_{evap}}{T_{cond}^{in}}\left(1+\frac{1}{COP}\right)\right) \quad \text{Eq. (2.10)}$$

$$Y = \frac{T_{evap}^{in}}{T_{cond}^{in}}\left(1+\frac{1}{COP}\right)-1 \quad \text{Eq. (2.11)}$$

$$X_1 = \frac{T_{evap}^{in}}{Q_{evap}} \quad \text{Eq. (2.12)}$$

$$X_2 = \left(\frac{T_{cond}^{in}-T_{evap}^{in}}{T_{cond}^{in}Q_{evap}}\right) \quad \text{Eq. (2.13)}$$

$$X_3 = \frac{Q_{evap}}{T_{cond}^{in}}\left(1+\frac{1}{COP}\right) \quad \text{Eq. (2.14)}$$

$$Y = X_1\Delta S_T + X_2 \mathrm{Q}_{\mathrm{leak,eqv}} + X_3 R \quad \text{Eq. (2.15)}$$

$$COP = \frac{Q_{evap}}{Power} \quad \text{Eq. (2.16)}$$

Table 2.3: Simple thermodynamic model parameters

Parameter	Estimate	Approximate Standard Error
ΔS_T	0.9372 kW/K	0.01 kW/K
R	0.0045 K/kW	2.2E-5 K/kW
$\mathrm{Q}_{\mathrm{leak,eqv}}$	-1532 kW	92 kW

2.4 Plate and Frame Heat Exchanger

The plate and frame heat exchanger in the IWE is used to pre-cool the chilled water return from the data center before entering the evaporator bundle of the chiller. The pre-cooling effect, free-cooling, reduces the amount of chiller work to produce chilled water. Two types of heat exchangers are recommended for waterside economizers; plate and frame heat exchangers and shell and tube (Standford, 2003). Plate and frame heat exchangers can achieve close approach temperatures between 1℉ and 2℉ as opposed 2 ℉ for shell and tube types. The simulation model uses a plate and frame configuration. The effectiveness-NTU method was used for modeling plate and frame heat exchangers is incorporated in the simulation model (Janna, 1998). The overall heat transfer coefficient, U_o, is assumed to remain constant for all flow rates. In reality, the overall heat transfer coefficient is a function of Reynolds number in both hot and cold water streams which will increase or decrease the heat transfer effectiveness (Croce, D'Agaro, 2002). The exit conditions for both chilled water and condenser are found through an iterative process. In this simulation, the heat exchanger is enabled when the inlet condenser water is colder than the chilled water return from the CRAH by 1.5℃.

Table 2.4 contains the assumed values for the plate and frame heat exchanger for the IWE model. It is sized to provide full free cooling capability when condenser water temperature is at 11.4 ℃ or less, which accounts for the temperature approach on the heat exchanger. At this condenser water temperature, the return chilled water temperature can be cooled down to 12.8℃ which enables the chiller to be off.

Table 2.4: Plate and frame model parameters

Parameter	Value	Description
N_s	500	Number of plates
U_o	2.2 kW/m^2-K	Overall heat transfer coefficient
A	1.5 m^2	Surface area of a single plate

A plot of predicted leaving water temperatures is shown in Figure 2.4 as a function of the temperature difference at the inlets; temperature difference between chilled water return temperature (T_w[1]) from the RAH and condenser water supply temperature (T_c[1]) from the cooling tower. Assuming that the entering chilled water return temperature remains constant, the amount of heat that can be transferred to the condenser water increases with decreasing entering condenser water supply temperature.

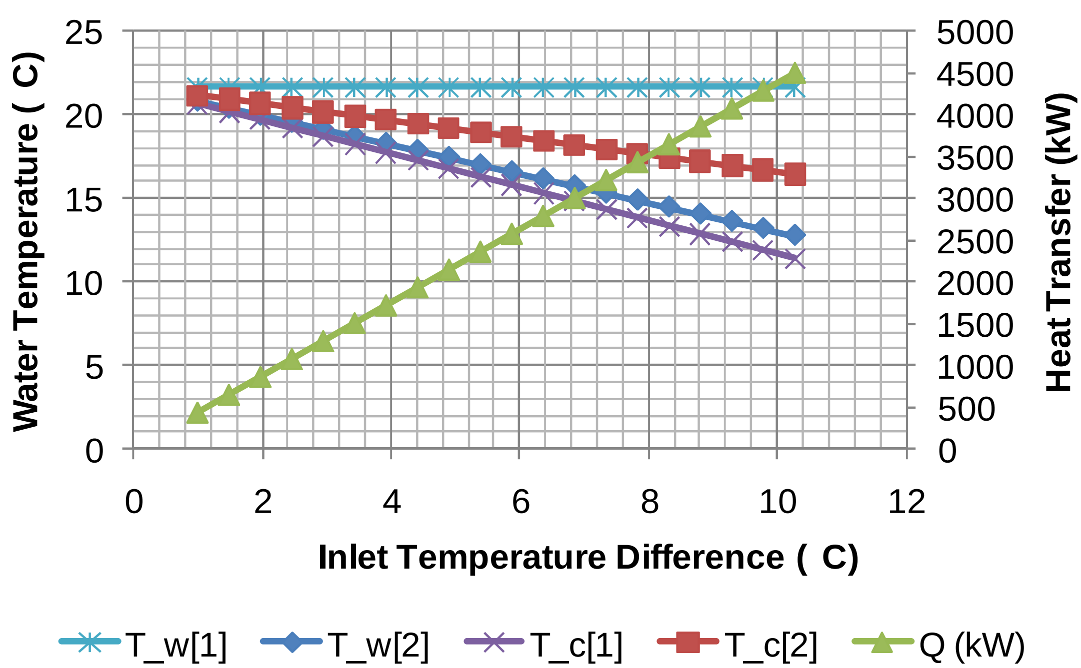


Figure 2.4: Plate and frame heat exchanger performance

2.5 Makeup Air Handler

The purpose of the make-up air handler (MUA) is to provide pressurization and fresh air to the data center (Taras, 2005). The pressurization requirement helps keep the data center clean of air particles by keeping the space differential pressure positive in relation to the surrounding space. This ensures that no outside particles infiltrate the data center space. The MUA simulation model contains a pre-filter, preheat coil, humidifier, single cooling coil, fan, final filter, and dampers, as shown in Figure 2.5. A reheat coil is not modeled since the air will be reheated by the hot return air plenum and narrow temperature and humidity control is not required. The exiting air from the MUA will mix with the hot return air from the data center. The preheat section provides sensible heating through the use of a coil. The humidification section assumes adiabatic evaporation that will cool the preheated air to the final set condition if the humidity ratio is below the set point. If needed, the humidified air is cooled to the final set point. The cooling coil also dehumidifies the air when required. Hourly weather bin data used for the fresh air in the simulation is from EnergyPlus Energy Simulation Software (United States Department of Energy, 2008) website or from HDBinWeather Software (Hana, 2008). Energy required to produce chilled water, humidification water, and hot water for the MUA are not included with the analysis and outside of the scope to study the economizer. The the MUA energy requirements are small compared to the total system energy used.

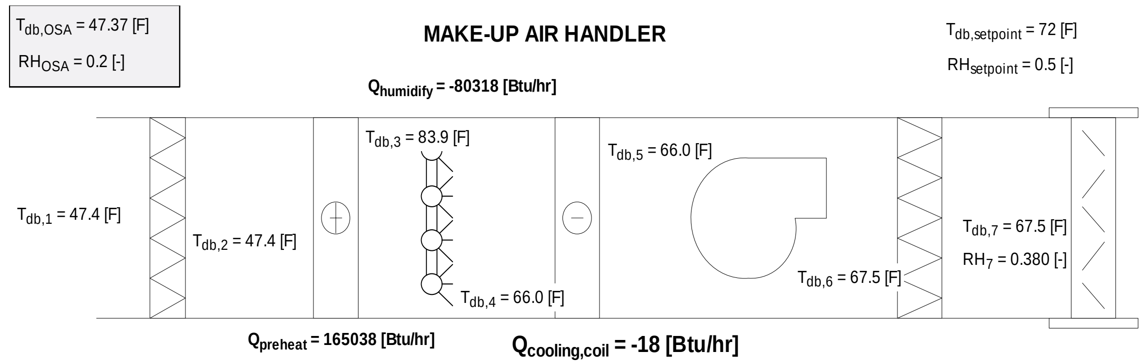


Figure 2.5: Makeup air handler schematic

2.6 Computer Room Air Handler

The primary function of the computer room air handler (CRAH) or return air handler (RAH) is to provide sensible cooling for the hot return air. However, if the surface of the cooling coil is below the dew point, dehumidification will occur. Dry, partially dry, and wet coil surfaces require different calculations, and the model follows ASHRAE's recommended calculation procedure (Owen, 2004). The RAH simulation model consists of a filter, cooling coil, fan, and humidifier sections. The simulation model for the RAH component assumes no dehumidification for a 12.8℃ (55 ℉) high temperature chilled water supply because humidity control is maintained by the MUA. The dew-point of the air is lower than 12.8℃ and therefore will not condense moisture out the air. If the coil surface is below the dew point of moist air, the dehumidification process cannot be ignored in energy and exergy balances. Chilled water demand, or flow rate and temperature, are set by this component. If the datacenter heat load increases, then the chilled water flow increases. Figure 2.6 shows a CRAH where the cooling coils are removing latent heat and estimates the amount of condensate.

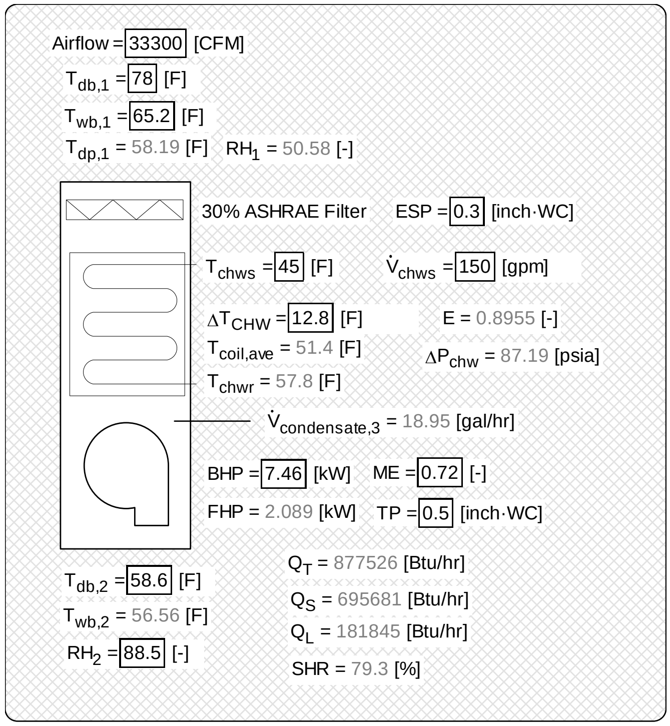


Figure 2.6: Computer room air handler (CRAH) model

2.7 General Exergy Theory - Air Side

Kanoglu et al. documented sensible cooling and heating, heating with humidification, cooling with dehumidification, evaporative cooling, and adiabatic mixing processes (Kanoglu, et al., 2007). Moist air is modeled as a combination of dry air and water vapor components using ideal gas laws. Engineering Equation Solver (EES) provides enthalpy and entropy values of moist air from a property database based on National Institute of Standards and Technology (NIST) JANAF Thermo-chemical tables (Klein, 2007). Using a property database simplifies the following equations without needing to calculate dry air and water vapor properties separately.

Exergy analysis requires choosing a dead state. The dead state occurs when the system is in equilibrium with the environment and serves as a reference point to calculate amount of work potential. Because data centers condition outside air and reject waste heat to the outside air, the dead state will be the current outdoor environmental conditions.

Mass Balance for Dry Air:

$$\sum_{in} \dot{m}_a = \sum_{out} \dot{m}_a \qquad \text{Eq. (2.17)}$$

Mass Balance for Water Vapor:

$$\sum_{in} \dot{m}_w = \sum_{out} \dot{m}_w \qquad \text{Eq. (2.18)}$$

Mass Balance for Water Vapor as a Ratio of Dry Air (Assuming no humidification or dehumidification, $\Delta\omega = 0$):

$$\sum_{in} \dot{m}_a\, \omega = \sum_{out} \dot{m}_a\, \omega \qquad \text{Eq. (2.19)}$$

$$\dot{m}_w = \dot{m}_a(\omega_{in} - \omega_{out}) \qquad \text{Eq. (2.20)}$$

Energy Balance (Assuming no work, $W = 0$):

$$Q_{in} + \sum_{in} \dot{m}h = Q_{out} + \sum_{out} \dot{m}h \qquad \text{Eq. (2.21)}$$

Entropy Balance:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = 0 \qquad \text{Eq. (2.22)}$$

$$\sum_{in} \dot{S}_{\dot{Q}} + \sum_{in} \dot{m}s - \sum_{out} \dot{S}_{\dot{Q}} - \sum_{out} \dot{m}s + \dot{S}_{gen} = 0 \qquad \text{Eq. (2.23)}$$

$$\sum_{in} \frac{\dot{Q}}{T_k} + \sum_{in} \dot{m}s - \sum_{out} \frac{\dot{Q}}{T_k} - \sum_{out} \dot{m}s + \dot{S}_{gen} = 0 \qquad \text{Eq. (2.24)}$$

where: k=boundary

Exergy Balance:

$$\sum_{in} \dot{Ex}_{\dot{Q}} + \sum_{in} \dot{m}\psi - \sum_{out} \dot{Ex}_{\dot{Q}} - \sum_{out} \dot{m}\psi - \dot{Ex}_{des} = 0 \quad \text{Eq. (2.25)}$$

$$\sum_{in} \dot{Q}\left(1 - \frac{T_0}{T_k}\right) + \sum_{in} \dot{m}\psi - \sum_{out} \dot{Q}\left(1 - \frac{T_0}{T_k}\right) - \sum_{out} \dot{m}\psi - \dot{Ex}_{des} = 0$$

$where$: Eq. (2.26)

$k = boundary$

Stream Flow Exergy:

$$\psi = h - h_0 - T_0(s - s_0) \quad \text{Eq. (2.27)}$$

Exergy Destruction:

$$\dot{Ex}_{dest} = T_0\dot{S}_{gen} \quad \text{Eq. (2.28)}$$

Second Law Efficiency (Exergy Efficiency):

$$\eta_{ex} = \frac{\dot{Ex}_{out}}{\dot{Ex}_{in}} = 1 - \frac{\dot{Ex}_{des}}{\dot{Ex}_{in}} \quad \text{Eq. (2.29)}$$

2.8 General Exergy Theory – Open Systems

In general, thermal plants are open systems and are not evaluated based on closed system models. The following set of equations generally governs most components of the thesis simulation model. In subsequent sections, each piece of mechanical equipment is uniquely modeled to show how it would perform at varying conditions.

First Law of Thermodynamics:

$$\frac{dE}{dt} - \sum_{i=0}^{n} \dot{Q}_i - \dot{W} + \sum_{in} \dot{m}h^0 - \sum_{out} \dot{m}h^0$$

$where$: Eq. (2.30)

$$h^0 = h + \frac{V^2}{2} + gz$$

Second Law of Thermodynamics:

$$\dot{S}_{gen} = \frac{dS}{dt} - \sum_{i=0}^{n} \frac{\dot{Q}_i}{T_i} - \sum_{in} \dot{m}s + \sum_{out} \dot{m}s \geq 0 \qquad \text{Eq. (2.31)}$$

Steady State Exergy Balance

$$\dot{E}x = \sum_{i=1}^{n} \left(1 - \frac{T_0}{T_i}\right) \dot{Q}_i + \sum_{in} \dot{m}(h^0 - T_0 s) - \sum_{out} \dot{m}(h^0 - T_0 s) - T_0 \dot{S}_{gen} \qquad \text{Eq. (2.32)}$$

2.9 Simulation Modeling

Each mechanical component for the IWE is modeled as a module that produces output states with given inlet conditions; e.g. air handlers, chillers, coils, cooling towers, heat exchangers, pumps, etc. The components are linked together and function as a complete thermal system by connecting the states through an airflow path or piping distribution. This allows each component to react to outside air conditions and with other mechanical equipment in the facility system, much as they would function in a true facility. Currently, a mass flow balance and corresponding temperature for each state interconnects the thesis model components. The model excludes pressure losses through the piping distribution and components because the amount of energy loss is negligible when compared to other energy losses in the system. Each state is calculated at steady-state conditions.

The main inputs to the simulation model are as follows: hourly weather bin data, data center heat load, operating set points, and performance characteristics of each mechanical component. For example, the chiller performance curve is an input to the model that will determine how much energy the chillers will consume at partial load conditions. In addition, fan and pump characteristic curves is implemented in the simulation model. The EES code is included in the appendix starting on page 69.

2.10 Data Center Heat Load

An example of the amount of heat dissipating from the data center equipment is calculated in Table 2.5 using Eq. (2.33) - (2.37). The cabinet rating, temperature, and quantity are inputs to calculate total airflow and power. The general energy balance, Eq. (2.33), is used to find unknown mass flow rate to achieve the temperature rise when power draw is also known. This method is used in the simulation model as a high-level estimate for operating conditions in the data center. Airflow and heat load serve as inputs to the return air handler to determine required chilled water flow from the chiller plant.

Table 2.5: Example of data center heat load calculations

	Cabinet Description [-]	Cabinet Rating [kW]	Temperature Rise [F]	Quantity [-]	CFM/Cabinet [ft^3/min]	Total CFM [ft^3/min]	Total Power [kW]
Row 1	HP BLADES	16.5	45	440	1160	510,532	7,260
Row 2	NET APP	7	30	20	738	14,767	140

Not all air supplied from the air handler is used to cool the servers. The servers will draw in supply air, and excess supply air will bypass the servers and return to the air handler. The air temperature delta across the RAH coil is less than the temperature rise across the servers. A bypass flow factor of 20 percent is applied to the airflow calculations to account for adiabatic mixing. The amount of air that bypasses the server does vary by air distribution design, where air is delivered by overhead or under the floor, and can be found using CFD modeling tools (Herrlin, 2005; Sorell, Abougabal, Khankari, Gandhi, Watve, 2006; Sorell, Escalante, Yang, 2005). In some cases, air is undersupplied which leads to virtually no bypass factor but the server inlet conditions are within allowable specifications. In other cases, more air is supplied than what is required by the servers to insure inlet temperatures are within the server manufacture's range or the data center owner's allowable range. The air delivered by CRAH can be 80 percent to 120 percent or more of the total server air flow.

General Energy Balance:

$$Q - \dot{m}c_p(T_{out} - T_{in}) \qquad \text{Eq. (2.33)}$$

$$\Delta T = T_{out} - T_{in} \qquad \text{Eq. (2.34)}$$

Volumetric Flow Rate Relationship with Mass and Density:

$$\dot{V} = \frac{\dot{m}}{\rho} \qquad \text{Eq. (2.35)}$$

Volumetric Flow Rate given Heat Dissipation and Change in Temperature:

$$V = \frac{Q}{\rho c_p \Delta T} \qquad \text{Eq. (2.36)}$$

Properties Evaluated at the Average Temperature:

$$T_{ave} = \frac{T_{out} + T_{in}}{2} \qquad \text{Eq. (2.37)}$$

2.11 Data Center Facility Modeling Inputs

Additional modeling assumptions have been provided in Table 2.6 that were used to generate the data set for this analysis. There are many possible permutations of a facility designs for a data center, and this represents one of many possible design options. Where possible, equipment sizes such as the cooling tower, cooling tower fans, and plate and frame heat exchangers, were cross-referenced with available supplier catalogs (GEA, 2009; Polaris, 2009).

Table 2.6: Data center facility modeling inputs

Description	Value
Location	San Francisco
Data Center Cooling Load	3.8 MW
Total Air Flow	176 m^3/s
Qty of Air Handlers	8
Fan Brake Power	25.3kW each
Chilled Water System	
Chilled Water Supply Temperature	12.8 °C, 8.9 ΔT
Chilled Water Flow	121.1 l/s
Nominal Chiller Capacity	4.5MW
Cooling Tower, Non Free Cooling	
Condenser Water Supply Temperature	17.8 °C, 5.6 ΔT
Condenser Water Flow	212 l/s
Wet-bulb design approach	5.6 °C
Cooling tower characteristic (Ka)	2.219 kg/m^3-s
Cooling Tower, Sized for Free Cooling	
Wet-bulb design approach	2.2 °C
Approximate Size Relative to Non-Free Cooling	275%
Cooling tower characteristic (Ka)	2.165 kg/m^3-s
Data Center Temperature Set point	22.5 +/- 2.5°C
Data Center Relative Humidity Set point	40-55%

There are many permutations of chilled water and condenser water systems with each of them having their advantages and disadvantages. The modeled chilled water system is based on a variable primary only flow distribution system. The condenser water is assumed to be constant flow with variable speed drives for the fans in the cooling tower. The pump head for both chilled water and condenser water are assumed to be 30.5 m for the simulation. The pump head is dependent on chilled and condenser water system design, such as pressure drop through cooling coils, chiller tube bundles, piping scheme, etc. This pump head value is only used to generate pump power curves for this simulation. The counter-flow cooling tower requires pressurized nozzles to function verses a cross-flow tower which does not need spray heads. The cooling tower could be located on a rooftop with chillers in the basement, which would require more pumping energy than if the cooling towers were located on the same elevation.

The data center environment is assumed to represent a fully utilized high-density design with hot aisle containment (Garday, 2007). The hot aisle containment allows for hotter return air temperatures and potentially increases the number of partial or full free cooling hours. Each of these assumed values can be easily changed to run the model under different conditions and for site-specific design requirements.

CHAPTER 3 RESULTS

The EES simulation model calculated energy usage and exergy destroyed for each mechanical component shown in Figure 1.1 and Figure 1.2. Energy usage for the MUA and cooling tower in San Francisco is discussed for varying inlet conditions. As the cooling tower and MUA interact with outdoor air conditions, the other mechanical equipment adjusts accordingly to the changing conditions. The energy and exergy plots demonstrate the influence of dry-bulb, wet-bulb, and relative humidity on mechanical components that do not directly interact, mix, or share boundaries with the outdoor air.

3.1 Make-up Air Handler

MUA energy requirements have been modeled and trended based on San Francisco weather bin data. Illustrated in Figure 3.1, the absolute energy of heat transferred to incoming outside air is plotted from the EES results. Each point in the chart represents the starting outside conditions for dry bulb temperature and relative humidity and then the air is conditioned to the required set point conditions exiting the MUA. The operating set point plus an allowable range permits the MUA to float with outside air conditions as an energy savings method. When the set point of 72 ±6℉ and 50 +5/-1 0 percent RH was modeled, as shown in Figure 3.1, the MUA uses the least amount of energy because OSA conditions are near the set points. Therefore, greater energy savings can be achieved when the set point operating allowable range is widened. Likewise, the energy usage will increase if the operating set points are controlled to a narrower range.

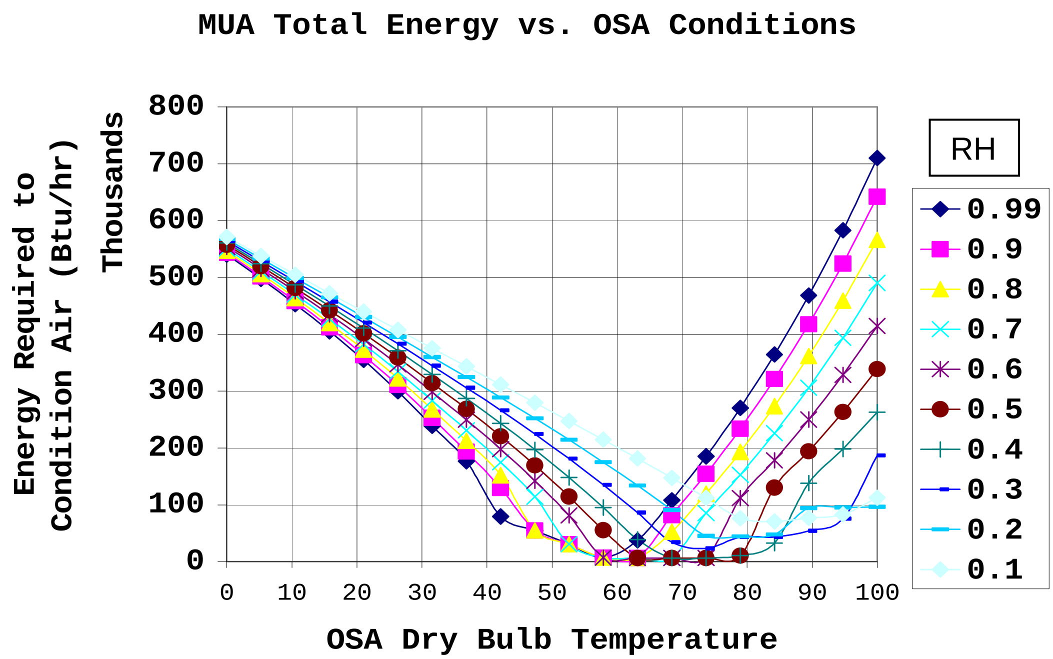


Figure 3.1: Makeup air absolute energy usage for different dry bulb and relative humidity conditions

The energy usage for a MUA can fluctuate overtime as shown in Figure 3.2, Figure 3.3, and Figure 3.4 when OSA conditions change. Figure 3.2 was generated from a single 24-hour period in the month of January. It shows total absolute MUA energy is influenced by wet bulb temperature throughout the 24-hour period. Early morning hours require preheating and humidification. As temperature rises in mid-day, the MUA energy usage decreases. In the afternoon hours, the OSA temperatures decrease and MUA energy usage increases.

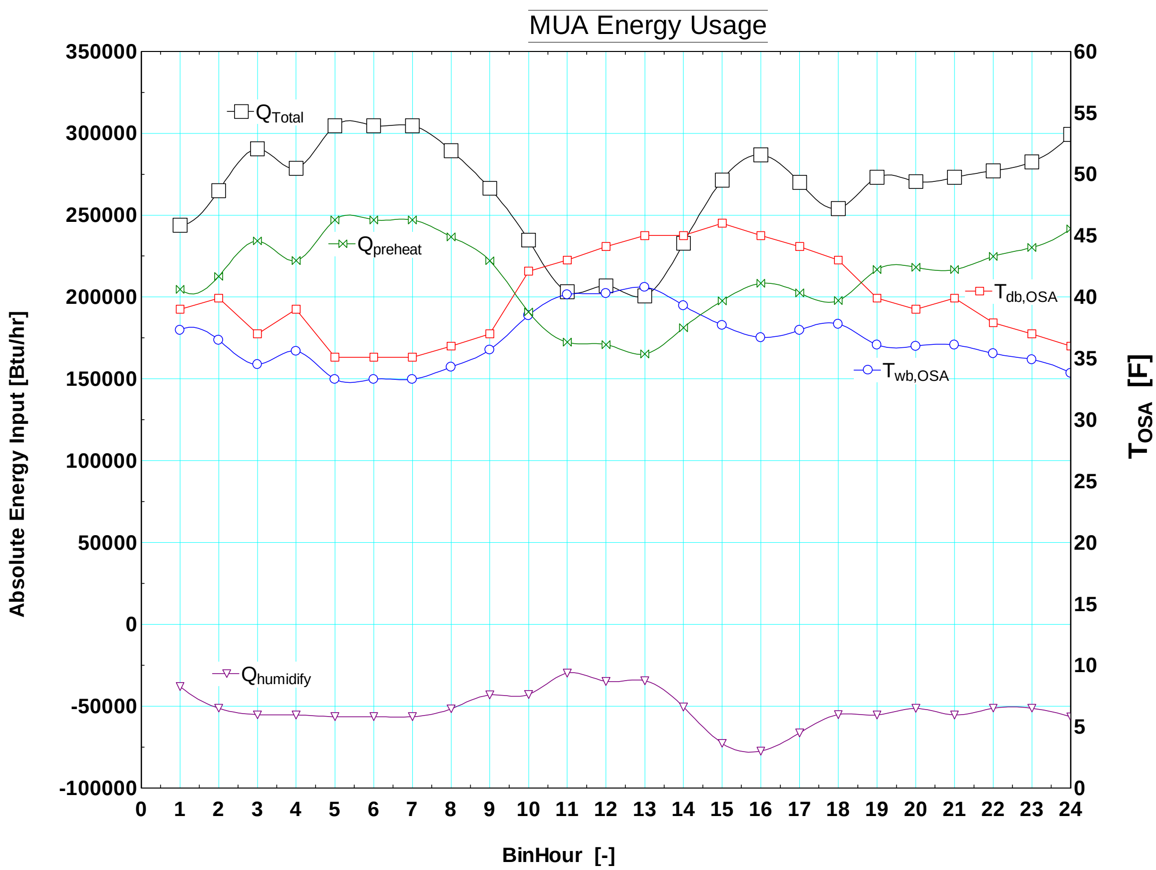


Figure 3.2: Makeup air handler absolute total energy usage for a 24-hour period with varying wet bulb conditions

In Figure 3.3, the MUA model was evaluated from 1 to 8760 hours of the year based on San Francisco weather bin data. The computation time for the model exceeds three hours as it solved for each bin hour for only the MUA unit. The computation time increases to 50 hours as more calculations state steps are included into the simulation model, representing interaction with other mechanical components. The graph qualitatively shows MUA energy requirements reduced between the bin hours of 3500 and 7000. This indicates opportunities to use 100 percent OSA for free-cooling most of the year.

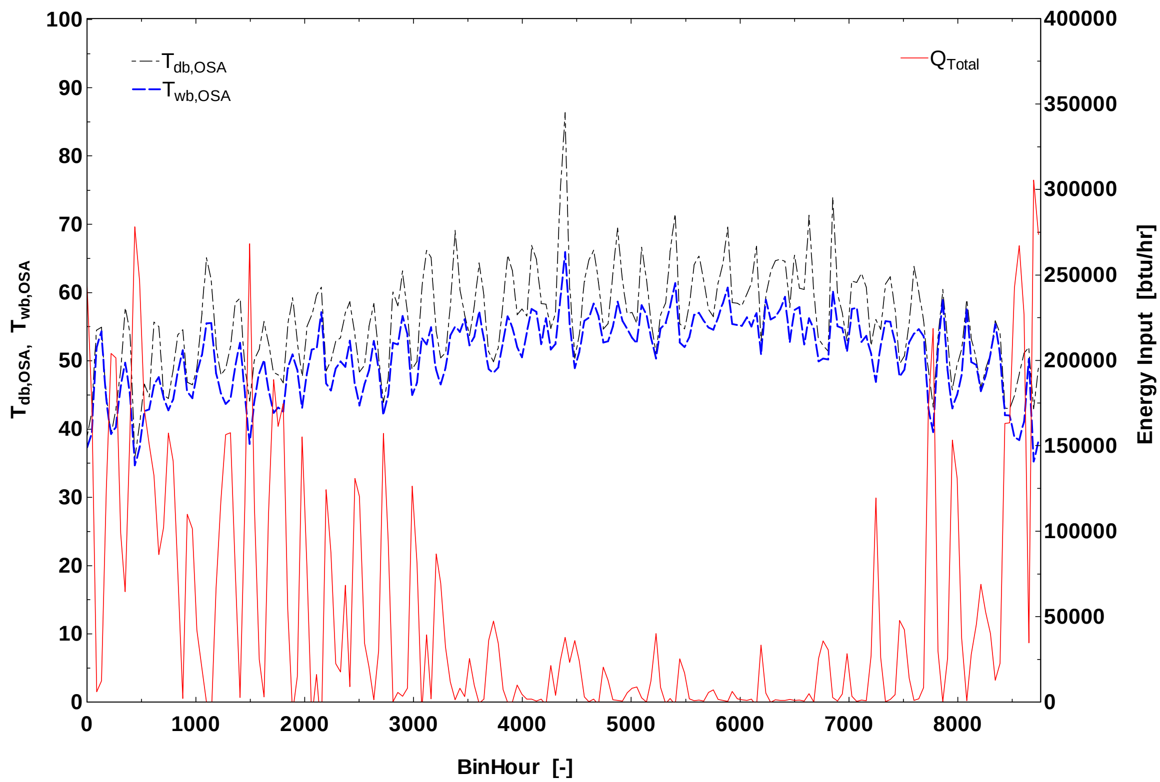


Figure 3.3: San Francisco weather bin data and projected MUA energy usage

No preheat or humidification is required to achieve the desired set point being modeled for the warmest day in San Francisco as shown in Figure 3.4. As dry bulb and wet bulb temperatures rise, the MUA is cooling more and increasing the chiller demand. When temperatures reach 55 ℉ at app roximately 10:00PM, the MUA preheats the air but does not require humidification. The plot also shows that there are hours when the MUA does not need to condition the air but uses only fan energy.

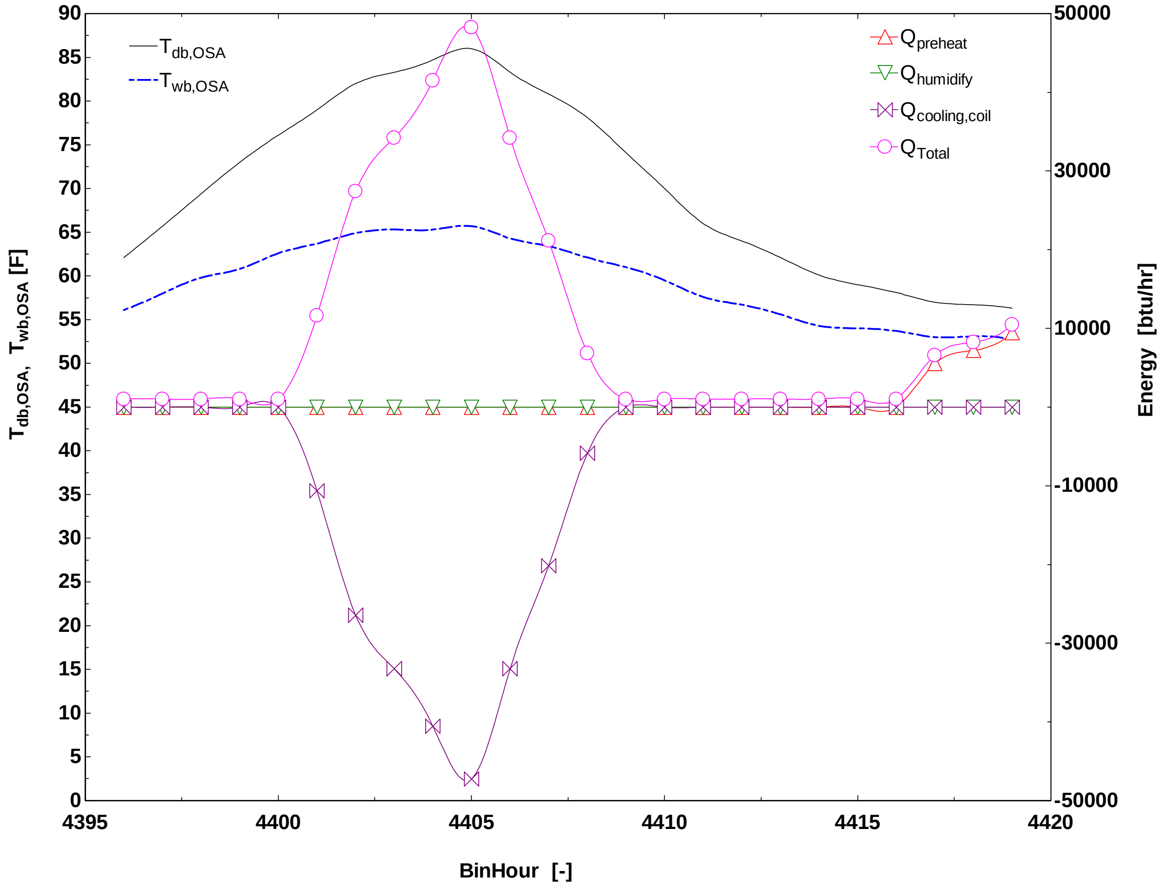


Figure 3.4: MUA energy transfer on warmest day in San Francisco

3.2 Cooling Tower

The cooling tower fan energy usage was modeled and was found to vary with relative humidity as shown in Figure 3.5. The less humid the air, the easier it is for adiabatic vaporization to occur and requires less air flow. As a result, fan brake horsepower consumption is reduced. When humidity increases, it becomes more difficult to evaporate water and therefore more airflow is needed to evaporate the condenser water. Control logic must be modeled to regulate fan speed based on exiting condenser water temperature and entering OSA wet-bulb temperatures. The wet-bulb temperature represents the coldest state that the condenser water can achieve with evaporative cooling. If the performance of heat transfer is known for the cooling tower, then the fan and pumping energy can be optimized for lowest energy usage and maximum efficiency.

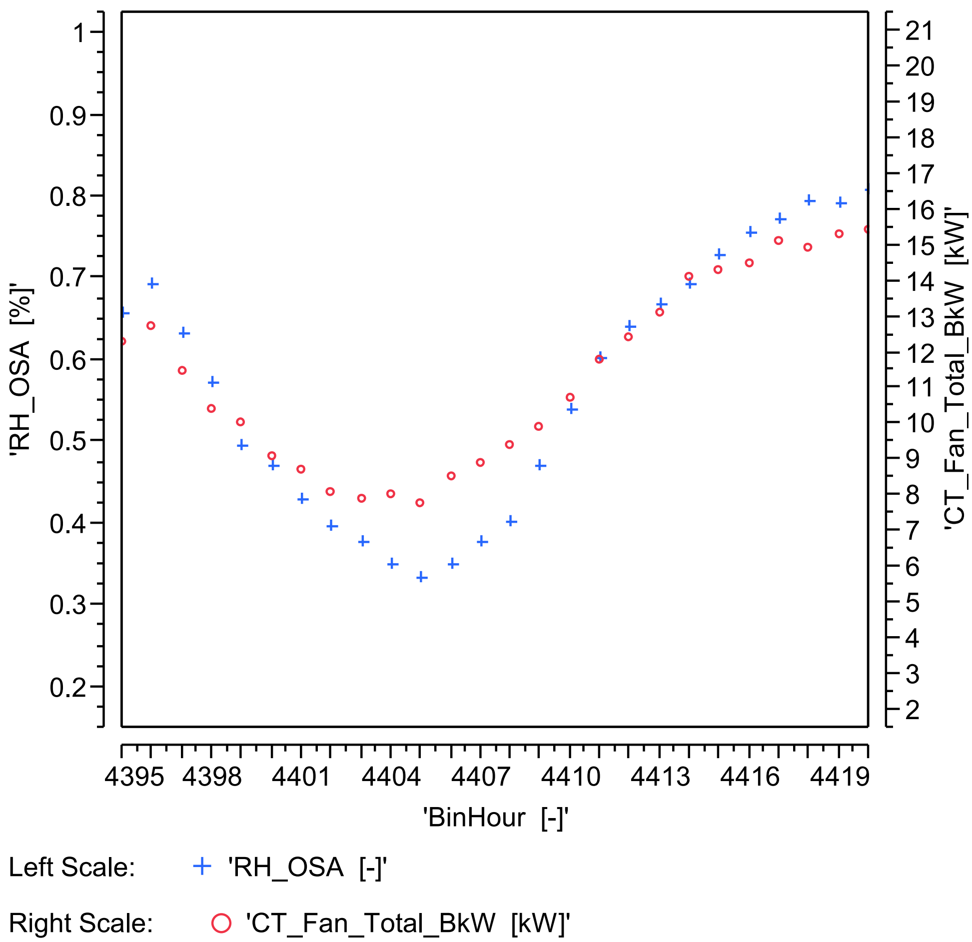


Figure 3.5: Evaporative cooling tower energy usage on warmest day for San Francisco

In both the MUA and cooling tower models, energy consumption varies with outside air conditions. These two systems are located at the boundary where mechanical equipment interacts with varying outdoor conditions. The simulation models respond to a change in environmental condition.

3.3 Economization Hours

The following charts and graphs are the result of running the normal chiller plant and IWE models developed using Engineering Equation Solver (EES) (Klein, 2007). A statistical software package, JMP , was utilized to analyze the data set and determine which parameters have the most influence for a given variable. Hourly weather bin data used in the simulation is based on San Francisco, California, USA.

Based on the results of the IWE model and assumed operating set points, there are 2659 hours/year of full free cooling (chiller completely bypassed) and 6063 hrs/year of partial free cooling. The remaining 38 hours indicate no partial free cooling is possible because the leaving condenser water temperature from the cooling tower is greater than the chilled water return temperature from the CRAH. If the chilled water set point were lower, i.e. 7.2 °C versus 12.7°C, then the number of full free cooling hours is reduced to around 322 hours/year, 7334 partial free cooling hours/year, and full load on the chiller for 1104 hours/year. Therefore, choosing the right set points, elevating chilled water return temperatures, can significantly influence the number of available economizer hours.

3.4 Energy Consumption

Table 3.1 estimates the power consumption for this system design scenario if the economizer were off throughout the year. The data center load was assumed constant. However, in real world operation this would vary. Since the data center load was constant in the simulation, the pumping power requirements for both chilled water and condenser water is constant. If the condenser water system were a variable flow design, we would expect to see more significant changes in power consumption as the controls would adjust VFD frequency to maintain leaving water temperature with varying wet-bulb conditions. The condenser water system modeled in this design was a constant flow with the cooling tower fans ramping up and down to maintain leaving condenser water temperature. 15 percent of the total power consumed by the chiller plant is in pumping. This shows that there is further opportunity to reduce pump energy or select a different condenser water system design such as variable flow. The cooling tower fan power consumption amounts to less than 1 percent of the power. Therefore, reducing the amount of work for the vapor compression chillers is more important than reducing power consumption for pumps and fans. If the cooling tower fill volume and fill height were smaller, the fan power consumption would increase because the tower would work harder to maintain leaving condenser water temperature.

Table 3.1: Basic chiller plant estimated power consumption (kWh)

Bin Month	Hours	CHW PMP	CRAH	Cooling Tower	CW PMP	MAH	Chiller	DC Cooling Load	COP for Chiller Plant
January	744	21,182	20,237	5,258	55,934	4,161	388,371	2,834,640	6.0
February	672	19,132	18,278	6,909	50,521	3,758	351,045	2,560,320	6.0
March	744	21,182	20,237	6,398	55,934	4,161	388,389	2,834,640	6.0
April	720	20,498	19,584	7,894	54,130	4,027	376,161	2,743,200	6.0
May	744	21,182	20,237	10,769	55,934	4,161	390,183	2,834,640	5.9
June	720	20,498	19,584	11,874	54,130	4,027	378,637	2,743,200	5.9
July	744	21,182	20,237	13,775	55,934	4,161	393,290	2,834,640	5.9
August	744	21,182	20,237	14,509	55,934	4,161	394,243	2,834,640	5.8
September	720	20,498	19,584	13,884	54,130	4,027	382,604	2,743,200	5.8
October	744	21,182	20,237	12,825	55,934	4,161	391,384	2,834,640	5.9
November	720	20,498	19,584	9,342	54,130	4,027	376,168	2,743,200	6.0
December	744	21,182	20,237	6,010	55,934	4,161	388,404	2,834,640	6.0
	8,760	249,397	238,272	119,448	658,577	48,995	4,598,879	33,375,600	5.9

The estimated energy consumption for the IWE model is shown in Table 3.2. The same EES model was used to generate Table 3.1 with the exception of the economizer mode being enabled and a larger cooling tower. One important item to note is that the energy used to produce chilled water, heating water, and humidification for the MAH is not included with the MAH values provided in Table 3.1. They are assumed to be supplied from different systems and currently are outside the scope of the economizer system being studied. In general, the energy consumption for the MAH is much smaller than the chiller plant and is ignored. Figure 3.4 shows the cooling load for the MAH to be about 4.2 Tons (15kW) , which is less than 5 percent of the power consumed by one chiller in this simulation model. The power represented in Table 3.1 for the MAH is only fan power. Future revisions of the EES model may be revised to add more detail for a complete energy model of an operating facility, including compressed air for controls and boilers for hot water. The chilled water produced for the MAH requires a colder supply temperature, i.e., less than 7.2℃ for dehumidification, verses 12.7 ℃ sensible only cooling. The chilled water supply temperature for this analysis assumes 12.7 ℃.

Table 3.2: Estimated power consumption with an integrated wet-side economizer (kWh)

Bin Month	Hours	CHW PMP	CRAH	Cooling Tower	CW PMP	MAH	Chiller	DC Cooling Load	COP for Chiller Plant
January	744	21,162	20,237	7,804	55,934	4,161	44,007	2,834,640	21.3
February	672	19,117	18,278	9,730	50,521	3,758	94,077	2,560,320	14.4
March	744	21,163	20,237	10,084	55,934	4,161	70,661	2,834,640	17.5
April	720	20,483	19,584	11,328	54,130	4,027	112,402	2,743,200	13.6
May	744	21,170	20,237	11,488	55,934	4,161	177,635	2,834,640	10.5
July	744	21,174	20,237	10,226	55,934	4,161	242,296	2,834,640	8.5
August	744	21,175	20,237	9,720	55,934	4,161	249,386	2,834,640	8.3
June	720	20,489	19,584	10,837	54,130	4,027	211,920	2,743,200	9.1
September	720	20,492	19,584	9,274	54,130	4,027	244,675	2,743,200	8.2
October	744	21,173	20,237	10,951	55,934	4,161	221,567	2,834,640	9.0
November	720	20,486	19,584	10,329	54,130	4,027	137,192	2,743,200	12.1
December	744	21,164	20,237	8,949	55,934	4,161	71,226	2,834,640	17.6
	8,760	249,249	238,272	120,719	658,577	48,995	1,877,044	33,375,600	11.3

3.5 Energy Efficiency

COP is calculated for the chiller plant as the total data center cooling load divided by total power consumed by the chiller plant. An average yearly COP was found to be 11.3 for the IWE design and 5.9 without an economizer. As expected, power consumption to provide useful cooling is reduced in colder months compared to the summer months because the economizer is pre-cooling the return water prior to reaching the chiller, effectively reducing amount of compressor work for the chiller. Cooling tower fan energy is increased in the

IWE because more air is required to provide cold condenser water temperature with a smaller approach temperature to the wet-bulb. Approach temperature is the difference between leaving condenser water temperature and entering wet-bulb temperature.

A common metric used to measure chilled water plant efficiency is kW/ton or electrical power required per unit of useful cooling. It is the sum of all electrical power used by the chiller plant divided by the amount of useful cooling generated in the evaporator of the chiller. The lower the kW/ton or kW/kW results in higher energy efficiency. Figure 3.6 is a plot of calculated energy efficiency as a function of dry-bulb temperature for all 8760 hours of the year.

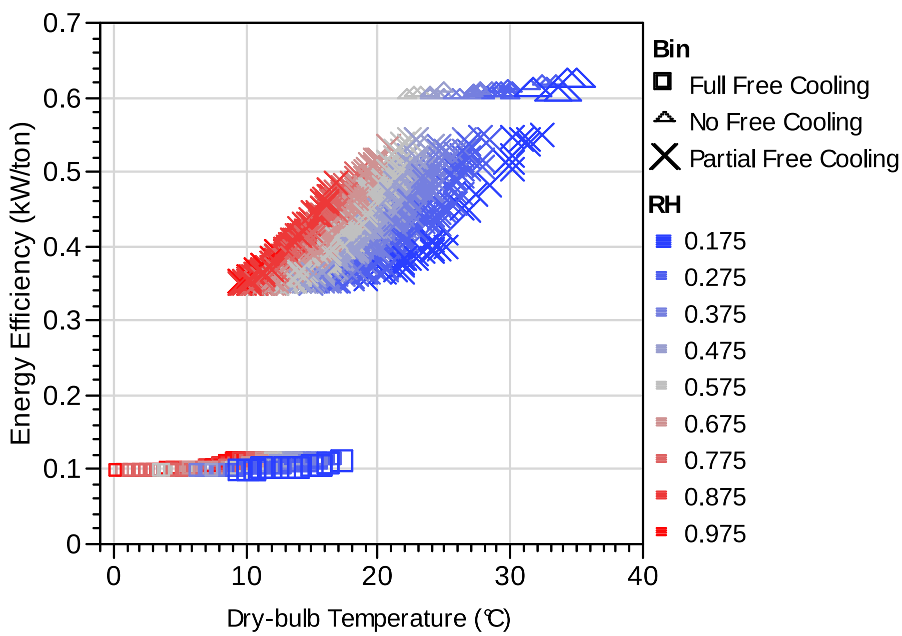


Figure 3.6: Chiller plant with an integrated indirect wet-side economizer energy efficiency performance (kW/ton) versus outside air dry-bulb temperature

Figure 3.6 shows that the economizer is very efficient when the economizer is enabled, full or partial. In full free cooling mode, only the pump and fan energy is used to transport heat from the data center to the environment. The energy efficiency was less than 0.15kW/ton in full free cooling mode. This value represents the best case scenario if nothing else in the chiller plant were required to operate. The 0.15kW/ton does not include ancillary loads to support the chiller plant, such as conditioning the chiller room (if required), controls, refrigerant oil heaters, and redundant chillers in standby. Ideally, full free cooling is the preferred mode of operation year round, but it is not physically possible because chilled water cannot be produced without the assistance of the vapor compression chiller.

Figure 3.7 is a plot of calculated energy efficiency as a function of wet-bulb temperature. Figure 3.7 shows the same data as Figure 3.6. The wet-bulb temperature can be found using dry-bulb temperature and relative humidity. It shows that there is a strong relationship between wet-bulb temperature and chiller plant kW/ton performance. The wet-bulb temperature can be used to predict evaporative chiller plant performance better than knowing dry-bulb temperature and relative humidity as separate variables.

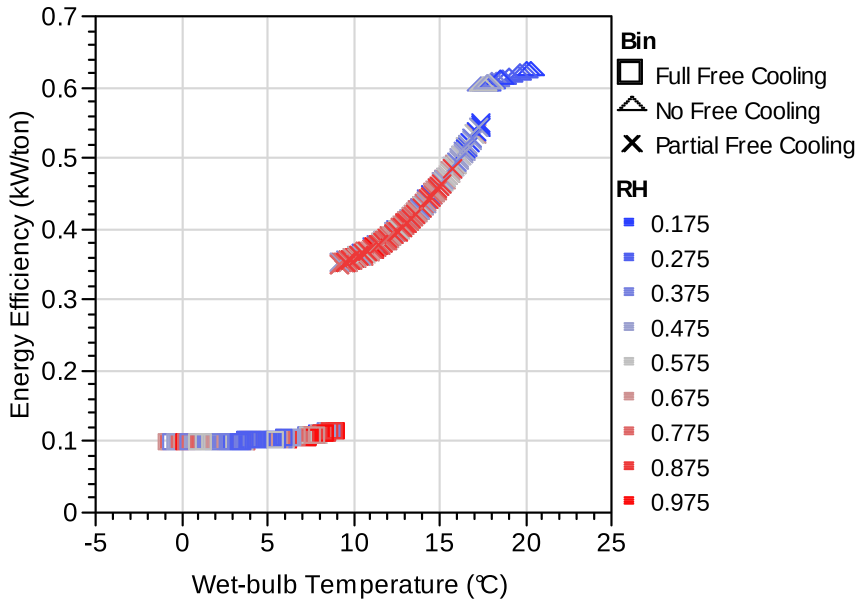


Figure 3.7: Chiller plant with an integrated indirect wet-side economizer energy efficiency performance (kW/ton) versus outside air wet-bulb temperature

As the system transitions from full free cooling to partial free cooling, the chillers are turned on. Typically, chiller manufactures required a minimum of 25 percent cooling load on the chiller to prevent the chiller from cycling on/off prematurely. Shown in Figure 3.7 at 9 ℃, the kW/t on does not transition smoothly from 0.15kW/ton efficiency value to nearly 0.35kW/ton because of the chiller power cycling on or off. When the system transitions from partial free cooling to no free cooling, there is an abrupt change in energy efficiency. This may be the result of not enough data points, 38 bin hours at that outdoor condition where no partial free cooling is possible.

The partial free cooling mode allows some of the heat load to bypass the chiller through the heat exchanger and is directly rejected to the cooling tower.

When wet-bulb temperature is less than 9.2 ℃, the ch illers can be turned off and chilled water can be indirectly produced by the cooling tower. The heat from the chilled water return is rejected to the condenser water in the plate and frame heat exchanger. When the wet-bulb temperature is around 4°C, then the cooling tower fans are running at very low speed or are in the off state. This operating point will change based on the selection of the cooling tower. In this simulation, partial free cooling yielded 0.35kW/ton to 0.55kW/ton efficiency.

The data is binned to show the mode of operation for when the chiller plant is in full free cooling (low kW/ton), partial free cooling (medium kW/ton), and no free cooling (high kW/ton). In addition, a color scale is provided outdoor relative humidity (RH) for each bin hour. Without the RH scale, it would be difficult to interpret what was causing the variations in energy efficiency for a given dry-bulb temperature. The RH scale shows that relative humidity has an influence on energy being consumed for a given dry-bulb temperate. The reason is that the cooling tower operation dependent on the evaporation process. The lower the humidity, the more that air can theoretically absorb moisture and remove heat from the condenser water.

The chilled water plant energy consumption increases non-linearly with increasing wet-bulb temperature during the partial free cooling mode of operation shown in Figure 3.7. Increasing leaving condenser water temperature from the cooling tower will increase the chiller power consumption because the compressor has to work harder to elevate the refrigerant temperature so that it can condense at the warmer cooling tower water temperature. The condenser water temperature set point increases with the wet bulb temperature.

The no free cooling mode of operation shows that chilled water plant kW/ton generally increases linearly with both dry-bulb and wet-bulb temperature. For the San Francisco climate, the relative humidity is low at these higher temperatures and therefore more evaporation can occur without requiring as much cooling tower fan power.

A comparison of the chiller plant model without IWE produces similar results but is shown to be less energy efficient on a kW/ton metric. The lowest kW/ton for this configuration is just above 0.6 kW/ton verses 0.1 kW/ton with IWE. If the condenser water temperature were lowered to 18.3℃, i nstead of 21.1℃, the chiller plant without IWE would have an improved performance to 0.58kW/ton.

3.6 Exergy Destruction

Minimizing exergy destruction leads to improving the second law efficiency. The exergy destroyed is calculated in for each of the major mechanical components as shown in without the economizer operating. If the boundary box were drawn around the individual component, the exergy destruction values represent energy that was not recovered or recycled. Reducing the amount of exergy being destroyed will lead to improving overall Second Law Efficiencies. The MAH operates less ideally in the colder months because increased preheating and humidification requirements. The cooling tower destroys less exergy in drier and warmer months from August to September but is much smaller compared to the other systems. If the climate were very humid and/or very cold and dry, the amount of exergy being destroyed by the MAH would contribute more to the total exergy being destroyed for the entire facility. The CRAH is relatively constant through the year and fluctuates because of the changing reference dead state. The dead state for the moist air comparisons is the outdoor air conditions for dry-bulb temperature and humidity. The dead state for the chilled water or condenser waters states was referenced at the dry-bulb temperature. This assumes that the water in the pipes does not evaporate and approach the wet-bulb temperature, although this assumption would not apply for water in the cooling tower that is in direct contact with the air.

Table 3.3: Exergy destroyed without an economizer (kWh)

Bin Month	Chiller	CRAH	Cooling Tower	MAH
January	322,381	204,731	73,866	6,428
February	293,614	186,007	58,148	4,255
April	315,058	199,711	61,745	4,134
March	324,044	205,708	68,346	5,283
May	327,328	207,081	57,940	3,628
June	317,739	200,797	52,955	3,093
July	329,073	208,085	52,423	3,050
August	329,751	208,115	50,301	3,012
September	319,163	201,644	49,826	2,909
October	328,610	207,631	54,389	3,242
November	315,598	199,759	59,081	4,136
December	323,212	204,980	70,882	5,781
Total	3,845,571	2,434,248	709,901	48,951

The exergy destruction is less for the wet-side economizer than a chiller plant without an economizer. The significant difference is the reduction of chiller power contributing to exergy entering the system and a colder condenser water temperature leaving the cooling tower. In the IWE model, the CRAH destroys more exergy on a yearly average than the chiller. This occurs because of the large temperature differences between the air and chilled water as heat is transferred from the air to the chilled water, and the chiller power consumption is reduced because of the economizer. Under these conditions, improving the efficiency of the CRAH becomes as important as reducing cooling load on the chiller as found in the energy only analysis. The exergy results show reducing exergy destruction with CRAH is an area of important focus to minimize exergy being destroyed if the chiller plant already has an economizer.

In both models, the total exergy destroyed for the CRAH is nearly the same. The differences are from a minor change in chilled water temperature return below exiting the plate and frame heat exchanger. In some cases, the chilled water supply temperature was colder than in the model with the economizer turned off. The exergy for the cooling tower is less in the IWE model because of the closer approach assumption for leaving condenser water temperature and its system temperature is operating closest to the outside air conditions; exergy entering on the airside of the cooling tower is zero because it is already at the dead state. In the model without the economizer, the leaving condenser water was constant for the majority of the year with the exception when the condenser water temperature was adjusted to a higher leaving water temperature set point because the approach exceeded the design of the cooling tower. The heat exchanger destroys exergy also because the heat transfer process generates entropy. Most of the entering exergy is recovered but the temperature differences between the condenser water and chilled water still causes exergy to be destroyed. This indicates that heat transfer between fluids across boundaries should be reduced as much as possible in the facility design or as differential temperature between hot and cold should approach zero degrees, which is not practical because of a very large heat exchanger. Some exergy destroyed because of heat transfer is not that important since in reality it will never be recovered. This may be an indicator that for the same climate, a direct wet-side economizer might destroy less exergy because the chiller and heat exchanger are eliminated during full free cooling mode. The exergy destroyed by the MAH in the IWE is exactly the same as the model with the economizer turned off.

Table 3.4: Exergy destroyed with an economizer (kWh)

Bin Month	Chiller	CRAH	Cooling Tower	Heat Exchanger	MAH
January	41,657	207,755	36,554	29,388	6,428
February	88,239	187,700	29,367	23,972	4,255
March	66,959	208,196	34,796	28,677	5,283
April	105,475	201,286	33,608	25,334	4,134
May	162,948	207,773	32,832	20,348	3,628
June	193,148	200,898	30,644	16,364	3,093
July	217,121	208,085	31,383	12,945	3,050
August	221,699	208,115	29,610	11,063	3,012
September	216,564	201,663	30,833	10,243	2,909
October	201,089	207,764	32,625	15,917	3,242
November	126,939	200,984	31,389	22,170	4,136
December	67,388	207,395	35,400	28,327	5,781
Total	1,709,226	2,447,614	389,041	244,748	48,951

Figure 3.8 shows that exergy destruction increases with lower humidity, but second law efficiency increases as well. Although counter-intuitive, this is

because the sum of the rates of exergy entering the components has increased more than the increase in rate of exergy destroyed, leading to an increase in efficiency. More cooling tower water is being evaporated, which increases the amount of makeup water exergy entering the system, but fan energy has been reduced. The exergy destroyed in these figures is for the CRAH, MAH cooling tower, cooling tower makeup, data center air leakage, chiller, and heat exchanger.

The exergy destruction is plotted for all bin hours. Exergy destruction is minimized for the system in Figure 3.8 when the cooling tower exergy destruction is at its minimum. This occurs when the wet-bulb temperature is near 9.2℃ and when the economizer is in full free cooling mode. When the wet-bulb temperature is greater than 9.2℃, the economizer swi tches from full to partial free cooling. The higher relative humidity corresponds to a larger wet-bulb temperature at the same dry-bulb temperature. The current condenser water temperature adjusts its set point to a higher temperature because the leaving cooling tower water is unable to maintain leaving water temperature. This occurs because the evaporation process is not removing enough heat to adequately cool down the leaving water to the set point. In partial free cooling, the exergy destruction increases non-linearly with increasing dry-bulb similarly for the same reason as that shown than in Figure 3.8 for the partial free cooling mode. Increasing the temperature of the condenser water entering the chiller also increases the work for the chiller and therefore energy efficiency decreases.

The STM model accounts for efficiency losses with elevating condenser water temperature and decreasing chilled water return temperature. Figure 3.8 also shows that it actually may be beneficial to switch over from partial free cooling to no free cooling mode sooner to reduce the amount of exergy being destroyed.

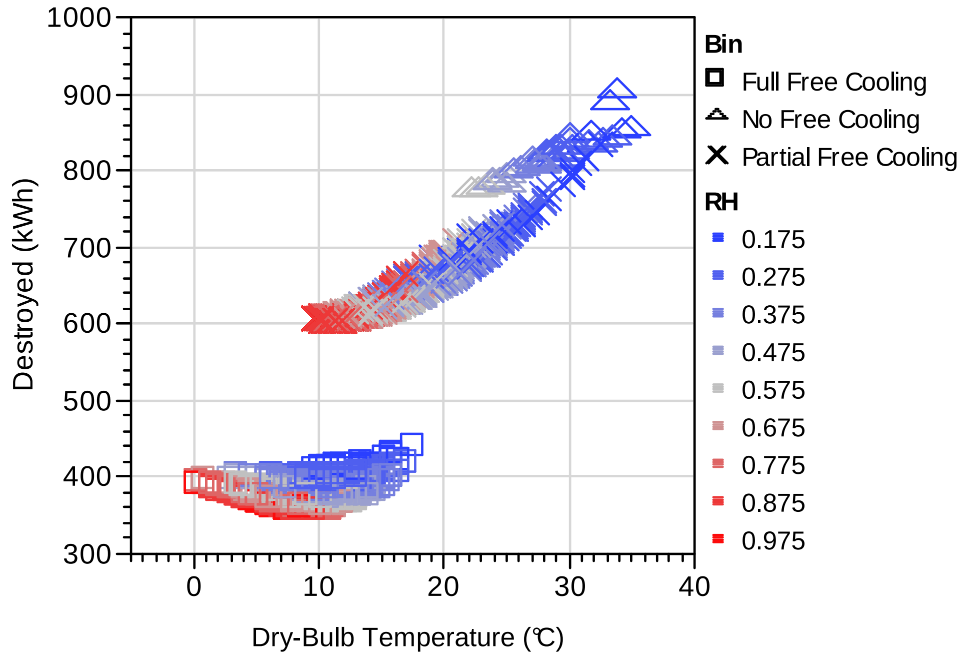


Figure 3.8: Total exergy destroyed versus outdoor dry-bulb temperature for chilled water plant with IWE

The 2nd Law efficiency is calculated for each bin hour of the year in Figure 3.9. Each mode of operation for the economizer has its most efficient and least efficient operating point. The full free cooling mode has its best efficiency of 56 percent when moist air conditions are cold and dry, 15.6℃ dry-bulb and 20 percent RH. When the economizer switches from full to partial free cooling mode at about 9℃ wet-bulb, system efficiency noticeably dro ps because the chiller is operating to produce chilled water to meet the desired chilled water supply temperature set point. When no free cooling is possible at higher wet-bulb temperature, the average efficiency drops again but this represents only 38 hours of data points. Further research is required to show that an economizer can be controlled by maximizing 2nd Law Efficiency, but it may be easier to use exergy destroyed calculations to optimize the economizer system.

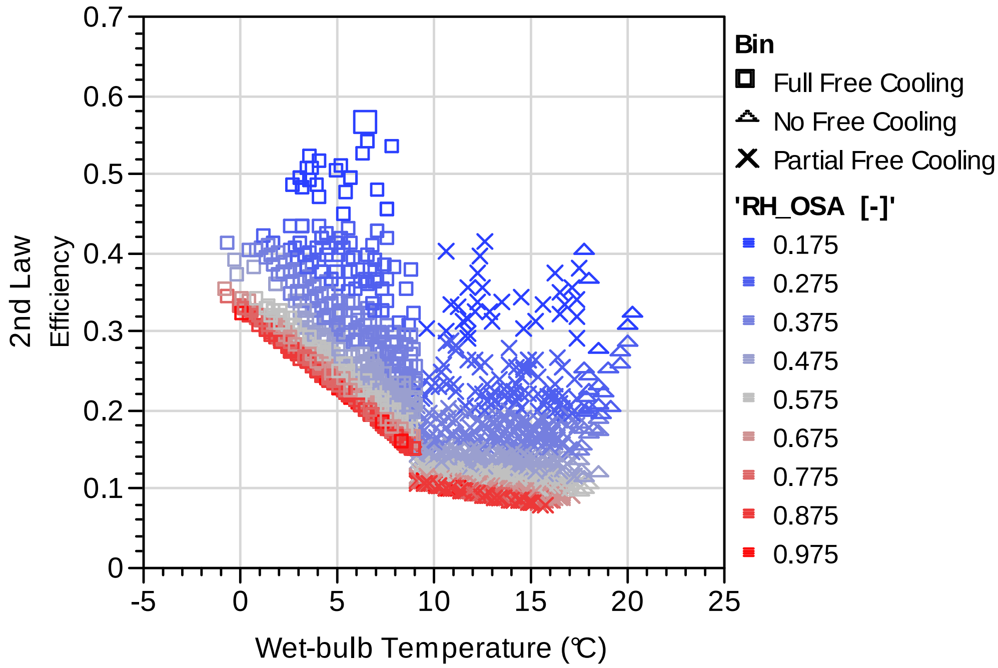


Figure 3.9: 2nd Law efficiency plot for wet-bulb temperature for chilled water plant with IWE

CHAPTER 4 CONCLUSION

A simulation model for predicting the response of a chiller plant with an economizer was developed to investigate where the most exergy was destroyed and where most energy was consumed for every hour of the year. For a chiller plant operating with an economizer, the CRAH was the largest source for exergy destruction. For a chiller plant operating without an economizer, the chiller was the largest source for exergy destruction. In addition, it was shown that Second Law efficiencies would vary significantly throughout a given year in San Francisco.

According to the 1st Law of Thermodynamics, reducing the amount of parasitic losses, such as friction, will make mechanical components more energy efficient. Since electrical power consumed is exergy destroyed, reducing overall energy consumption will minimize exergy destruction.

Second Law analysis will indicate when, where, and how the exergy is destroyed. Reducing the amount of heat transfer between fluids and reducing the temperature difference should reduce the amount of exergy destroyed in the system since entropy is a function of energy transferred and effective boundary temperatures. For the outside air economizer, indoor air is directly mixed with outside air. This eliminates one more energy transfer between system boundaries. This may be an indication that a system with an outside air economizer will yield the least amount of exergy destroyed.

4.1 Future Work

Additional simulation models should be developed to compare energy and exergy results other designs, such as OAE and DWE, for other climates. The exergy analysis may show which type of economizer is suited best for a given climate. The programming environment that EES provides, easy access to fluid properties, allows engineers to quickly evaluate designs and predict performance of a system under varying conditions. Listed below are a few areas that could improve the simulation model:

1. Modify cooling coil algorithms for partial and wet surfaces to account for the dehumidification process.
2. Modify plate and frame heat exchanger calculations to adjust heat transfer coefficients based on changes with the Reynold’s number, because flow rates and temperatures are not constant.
3. Modify MUA and RAH for OSA economizer operation.
4. Expand simulation for a DWE system.
5. Expand energy and exergy analysis for supporting systems that were excluded from this analysis such as the hot water system.
6. Improve the solver calculation method to use a technique other than the Newton-Raphson technique.

REFERENCES

Air-Conditioning and Refrigeration Institute (2003) ARI 550/590-2003: performance rating of water chilling packages using the vapor compression cycle. Retrieved from http://www.ari.org

Aprea, C., Rossi, F. d., Greco, A., Renno, C. (2003) Refrigeration plant exergetic analysis varying the compressor capacity. *International Journal of Energy Research 27*(7), 653-669.

ASHRAE (2008) HVAC systems and equipment *ASHRAE Handbook* (pp. 39.19).

Bejan, A. (2006) *Advanced engineering thermodynamics* (3rd ed.): John Wiley & Sons, Inc.

Bejan, A., Tsatsaronis, G., Moran, M. (1996) *Thermal design and optimization*: John Wiley & Sons, Inc.

Cengel, Y. A., Boles, M. A. (2006) *Thermodynamics: an engineering approach, 5th edition*: McGraw Hill.

Chen, C. K., Su, Y. F. (2005) Application of exergy method to an irreversible inter-cooled refrigeration cycle. *Proceedings of the Institution of Mechanical Engineers, Part A, Power & Energy 219*(8), 661-667.

Chen, C. K., Su, Y. F. (2006) Exergetic efficiency optimization of a refrigeration system with multi-irreversibilities. *Proceedings of the Institution of Mechanical Engineers, Part C, Journal of Mechanical Engineering Science 220*(8), 1179-1187.

Chen, C. K., Su, Y. F. (2007) Application of exergy method to a two-stage irreversible combined refrigeration system. *Proceedings of the Institution of Mechanical Engineers, Part C, Journal of Mechanical Engineering Science 221*(7), 807-813.

Croce, G., D'Agaro, P. (2002) Numerical analysis of forced convection in plate and frame heat exchangers. *International Journal of Numerical Methods for Heat and Fluid Flow 12*, 756-771.

Department of Planning and Development (2006) *Washington State Non-residential Energy Code* Retrieved from http://www.seattle.gov/DPD/Codes/Energy_Code/Nonresidential/Chapter_14/default.asp.

Dincer, I., Hussain, M. M., Zubair, S. M. (2004) A feasibility study of using thermal energy storage in a conventional air-conditioning system. *International Journal of Energy Research 28*(11), 955-967.

Fisk, W. J., Seppänen, O., Faulkner, D., Huang, J. (2005) Economic benefits of an economizer system: energy savings and reduced sick leave. *ASHRAE Transactions 111*(2), 673-679.

Garday, D. (2007) Reducing data center energy consumption with wet side economizers *Intel Information Technology White Paper*.

GEA (2009). Cooling tower calculator Retrieved 1/18/2009, from http://gea-energytechnology.com/opencms/opencms/gem/en/calculators/CT_Calculator.html

Gordon, J. M., Ng, K. C. (2000) *Cool thermodynamics: the engineering and physics of predictive, diagnostic and optimization methods for cooling systems*: Cambridge International Science Publishing.

Greenberg, S. (2007) Data center energy use, metrics, and rating systems. *ENERGY STAR Industry Focus Discussion: Development of Energy Performance Benchmark for Data Centers,* (October 31, 2007). Retrieved from http://www.energystar.gov/ia/partners/prod_development/downloads/DC_Energy_Metrics.pdf

Hana, B. (2008) HDBinWeather: Hands Down Software.

Harutunian, V. (2003) *Functional model and second law analysis method for energy efficient process design: applications in HVAC systems design.* Ph.D., The University of Texas at Austin, Texas, United States. Retrieved from http://proquest.umi.com/pqdweb?did=765218911&

Hasabnis, Y. C., Bhagwat, S. S. (2007) Performance evaluation of ammonia absorption refrigeration cycle based on exergetic coefficient of performance. *International Journal of Exergy 4*(1), 19-37.

Herrlin, M. K. (2005) *Rack cooling effectiveness in data centers and telecom central offices: the rack cooling index (RCI).* Paper presented at the American Society of Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE 2005 Annual Meeting, June 25, 2005 - June 29, 2005, Denver, CO, United States.

Imperatore, T. (1975) Proven ways to save energy in commercial buidlings. *Heating, Piping & Air Conditioning 47*(5), 48-53.

Janna, W. S. (1998) *Design of fluid thermal systems* (2nd ed.): PWS Publishing Company.

Kanoglu, M., Dincer, I., Rosen, M. A. (2007) Exergy analysis of psychrometric processes for HVAC&R applications. *ASHRAE Transactions 113*(2), 172-180.

Klein, S. A. (2007) Engineering Equations Solver (EES): F-Chart Software. Retrieved from www.fchart.com

Koomey, J. G. (2004, March 2, 2004) *Data center power use: a review of the historical data.* Paper presented at the IBM Austin Conference on Energy-Efficient Design, Austin, Texas.

Kotas, T. J. (1995) *The exergy method of thermal plant analysis*: Krieger Publishing Company.

Kurkjian, C., Glass, J., Routsen, G. (2007) Efficiency plus reliability. *ASHRAE Journal 49*(12), 26-31.

Li, N., Ren, C., Tang, G., Zhang, G., Yang, J. (2001) *Exergy analysis of moist air and energy saving potential in HVAC by evaporative cooling or energy recovery.* Paper presented at the Proceedings of the International Conference on Energy Conversion and Application (ICECA'2001), Wuhan, China.

Liu, G. (1994) *Second law analysis of thermal systems.* Ph.D., University of Nevada, Reno, Reno, Nevada, United States. Retrieved from http://proquest.umi.com/pqdweb?did=741188181&

Muangnoi, T., Asvapoositkul, W., Wongwises, S. (2007) An exergy analysis on the performance of a counterflow wet cooling tower. *Applied Thermal Engineering 27*(5-6), 910-917.

Muangnoi, T., Asvapoositkul, W., Wongwises, S. (2008) Effects of inlet relative humidity and inlet temperature on the performance of counterflow wet cooling tower based on exergy analysis. *Energy Conversion and Management 49*(10), 2795-2800.

Naphon, P. (2005) Study on the heat transfer characteristics of an evaporative cooling tower. *International Communications in Heat and Mass Transfer 32*(8), 1066-1074.

Nianping, L., Chengqin, R., Guangfa, T. (2002) Principles of exergy analysis in HVAC and evaluation of evaporative cooling schemes. *Building and Environment 37*(11), 1045-1055.

Oregon Department of Energy (2007) *Oregon non-residential building energy code*. Retrieved from http://www.oregon.gov/ENERGY/CONS/Codes/docs/16-Economizers.pdf.

Owen, M. S. (2004) Air-cooling and dehumidifying coils *HVAC Systems and Equipment* (2004 ed.). Atlanta, Georgia, United States: ASHRAE.

Paulus, D. M., Jr. (2000) *Second law applications in modeling, design and optimization.* Ph.D., Marquette University, Wisconsin, United States. Retrieved from http://proquest.umi.com/pqdweb?did=727833381&

Polaris (2009). Polaris plate heat exchangers Retrieved 02/05/2009, from http://www.polarisphe.com/pages.aspx?pid=29&name=Plate-Frame-Heat-Exchangers

Qureshi, B. A. (2004) *Design, rating and exergy analysis of evaporative heat exchangers.* M.S., King Fahd University of Petroleum and Minerals (Saudi Arabia), Saudi Arabia. Retrieved from http://proquest.umi.com/pqdweb?did=997886281&

Qureshi, B. A., Zubair, S. M. (2003) Application of exergy analysis to various psychrometric processes. *International Journal of Energy Research 27*(12), 1079-1094.

Qureshi, B. A., Zubair, S. M. (2006) An improved non-dimensional model of wet-cooling towers. *Proceedings of the Institution of Mechanical Engineers, Part E (Journal of Process Mechanical Engineering) 220*(E1), 31-41.

Qureshi, B. A., Zubair, S. M. (2007) Second-law-based performance evaluation of cooling towers and evaporative heat exchangers. *International Journal of Thermal Sciences 46*(2), 188-198.

SAS (2009) JMP statistical discovery software (Version 8.0.2). Retrieved from www.jmp.com

Saththasivam, J., Choon, N. K. (2008) Predictive and diagnostic methods for centrifugal chillers. *ASHRAE Transactions 114*(1), 282-287.

Schmidt, R., Beaty, D., Dietrich, J. (2007) Increasing energy efficiency in data centers. *ASHRAE Journal 49*(12), 18-24.

Shah, A. J., Carey, V. P., Bash, C. E., Patel, C. D. (2003) *Exergy analysis of data center thermal management systems.* Paper presented at the Proceedings of the ASME Advanced Energy Systems Division - 2003, Washington, DC., United States.

Shah, A. J., Carey, V. P., Bash, C. E., Patel, C. D. (2004) *An exergy-based control strategy for computer room air-conditioning units in data centers.* Paper presented at the Proceedings of the ASME Heat Transfer Division - 2004, Anaheim, CA, United States.

Shah, A. J., Carey, V. P., Bash, C. E., Patel, C. D. (2005a) *Exergy-based optimization strategies for multi-component data center thermal management: part I, analysis.* Paper presented at the Proceedings of the ASME/Pacific Rim Technical Conference and Exhibition on Integration and Packaging of MEMS, NEMS, and Electronic Systems: Advances in Electronic Packaging 2005, San Francisco, CA, United States.

Shah, A. J., Carey, V. P., Bash, C. E., Patel, C. D. (2005b) *Exergy-based optimization strategies for multi-component data center thermal management: part II, application and validation.* Paper presented at the Proceedings of the ASME/Pacific Rim Technical Conference and Exhibition on Integration and Packaging of MEMS, NEMS, and Electronic Systems: Advances in Electronic Packaging 2005, San Francisco, CA, United States.

Shah, A. J., Carey, V. P., Bash, C. E., Patel, C. D. (2008) Exergy analysis of data center thermal management systems. *Journal of Heat Transfer 130*.

Sorell, V. (2007) OA economizers for data centers. *ASHRAE Journal 49*(12), 32-37.

Sorell, V., Abougabal, Y., Khankari, K., Gandhi, V., Watve, A. (2006) *An analysis of the effects of ceiling height on air distribution in data centers.* Paper presented at the 2006 Winter Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE, January 21, 2006 - January 25, 2006, Chicago, IL, United States.

Sorell, V., Escalante, S., Yang, J. (2005) *Comparison of overhead and underfloor air delivery systems in a data center environment using CFD modeling.* Paper presented at the American Society of Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE 2005 Annual Meeting, June 25, 2005 - June 29, 2005, Denver, CO, United States.

Standford, H. W. (2003) *HVAC water chillers and cooling towers: fundamentals, application, and operation*: Marcel Dekker, Inc.

Starr, G. E. (1984) Computer facility 'free' cooling. *Heating, Piping & Air Conditioning 56*(5), 99-102.

Taras, M. F. (2005) Is economizer cycle justified for AC applications. *ASHRAE Journal 47*(7), 38-44.

Telecky, D. J. (1985). *Free cooling - wet side*. Paper presented at the New Directions in Energy Technology, Proceedings of the 7th World Energy Engineering Congress, Atlanta, GA, USA.

Tobias, J. R., Schade, G. R. (1976) *Energy savings in building by using an air economiser and night setback.* Paper presented at the 1976 Joint Automatic Control Conference, West Lafayette, IN, United States.

Tsaros, T. L. (1987) *Exergy analysis of heat pumps and heat pump heat exchangers.* M.S., University of Massachusetts Lowell, Massachusetts, United States. Retrieved from http://proquest.umi.com/pqdweb?did=754448801&Fmt=7&clientId=17867&RQT=309&VName=PQD

Tschudi, W., Fok, S. (2007) Best practices for energy-efficient data centers identified through case studies and demonstration projects. *ASHRAE Transactions 113*(1), 450-456.

United States Department of Energy. (2008). EnergyPlus energy simulation software weather data. Retrieved April 11, 2008 http://www.eere.energy.gov/buildings/energyplus/

United States Environmental Protection Agency (2007) *Report to congress on server and data center energy efficiency public law 109-431*. Retrieved from http://www.energystar.gov/ia/partners/prod_development/downloads/EPA_Datacenter_Report_Congress_Final1.pdf.

Wang, W. (2005) *A simulation-based optimization system for green building design.* Ph.D., Concordia University (Canada), Canada. Retrieved from http://proquest.umi.com/pqdweb?did=932404551&

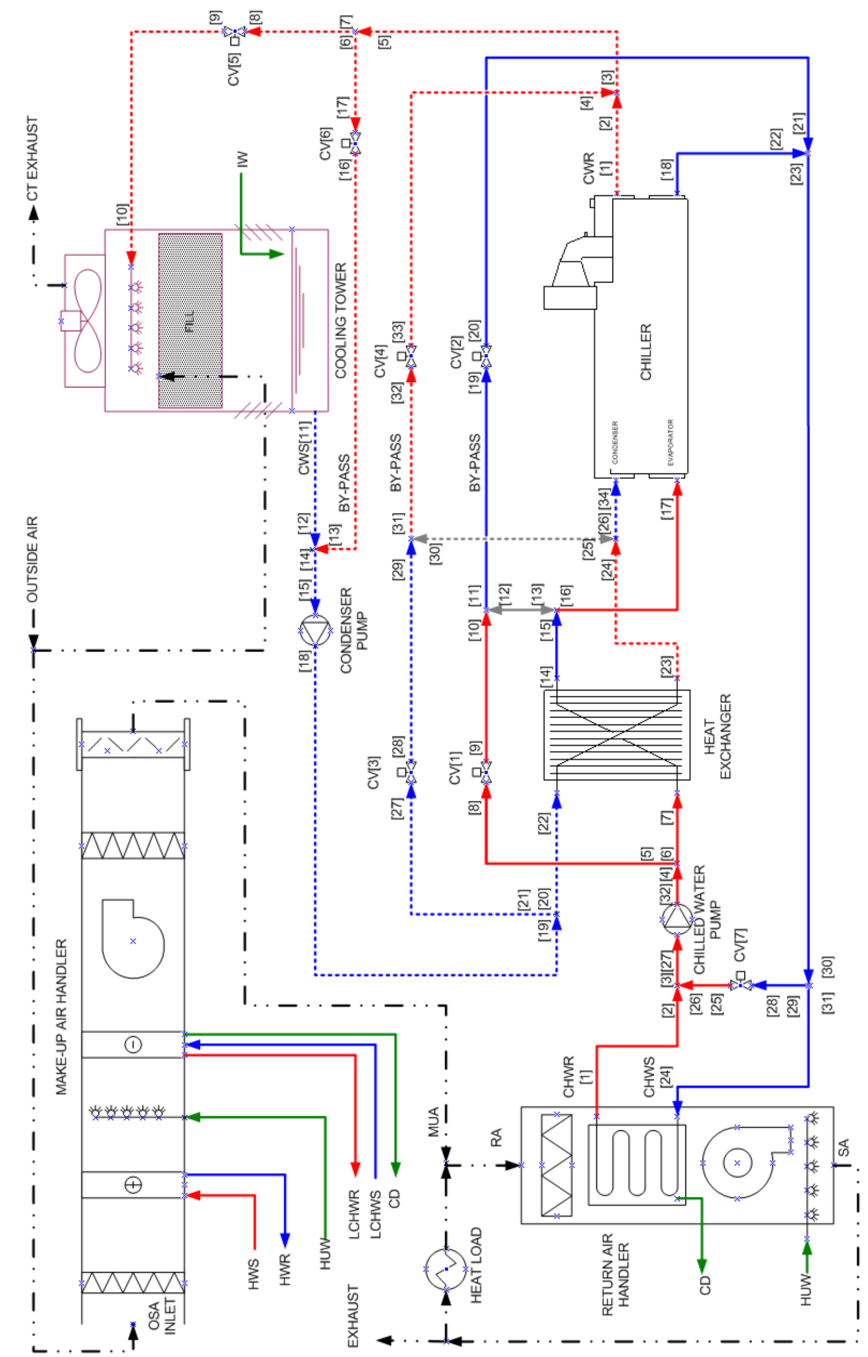
Wu, X. (2004) *Second law analysis of residential heating systems.* M.A.Sc., Concordia University (Canada), Canada. Retrieved from http://proquest.umi.com/pqdweb?did=845743421&

Xianguo, L., Guoyuan, M. (2007) Exergetic optimization of a key design parameter in heat pump systems with economizer coupled with scroll compressor. *Energy Conversion and Management 48*(4), 1150-1159.

Zmeureanu, R. (1988) Energy savings in HVAC systems in Montreal due to natural cooling. *International Journal of Ambient Energy 9*(2), 75-82.

APPENDICES

A.1 Indirect Wet-side Economizer (IWE)



A.2 EES CODE

```
$UnitSystem SI MASS DEG KPA C KJ
$TABSTOPS 0.25 0.5 0.75 1.0 1.25 in

"! -------------------------BEGIN CHILLER FUNCTIONS-------------------------"

PROCEDURE CHILLER2(T_chws_setpoint,T_db_OSA, T_wb_OSA, P_OSA,T_evap_i,T_cond_i,
m_dot_chw, DELTAS_T,Q_leak_eqv,R, m_dot_cw,Chiller_KW_Demand_Trigger:
T_cond_e,T_evap_e,
Power_chillers,Ex_dot_dest,Ex_dot_in,Ex_dot_out,eta_chiller,X_dot_chw[1..2],X_dot_cw[1..2],Q_
evap_demand)
	"Saththasivam, J., & Kim Choon, N. (2008). Predictive and Diagnostic Methods for
Centrifugal Chillers. ASHRAE Transactions, 114(1), 282-287"
	"Units must be in Kelvin"
	T_evap_e:=convertTemp(C,K,T_chws_setpoint)
	cp_e:=Cp(Water,T=T_chws_setpoint,x=0)

	Q_evap_demand:=m_dot_chw*cp_e*(T_evap_i-T_evap_e)
	N_Demand:=Ceil(Q_evap_demand/Chiller_KW_Demand_Trigger)

	Q_evap:=Q_evap_demand/N_Demand

	COP_pred:=((1+(T_evap_i/Q_evap)*DELTAS_T+Q_leak_eqv*(T_cond_i-
T_evap_i)/(T_cond_i*Q_evap))/(T_evap_i/T_cond_i-R*Q_evap/T_cond_i)-1)^(-1)

	Power_chiller:=Q_evap/COP_pred

	Q_cond:=(Q_evap+Power_chiller)*N_Demand
	cp_c:=Cp(Water,T=convertTemp(K,C,T_cond_i),x=0)

	T_cond_e:=Q_cond/(m_dot_cw*cp_c)+T_cond_i

	DELTAT_cond:=T_cond_e-T_cond_i

	Power_chillers:=Power_chiller*N_Demand

	"! Exergy Calculations"
	"Condensor Water Side"
	T_0:=convertTemp(C,K,T_wb_OSA)
	T_cw[1]:=convertTemp(K,C,T_cond_i)
	T_cw[2]:=convertTemp(K,C,T_cond_e)
	x_cw[1]:=ENTHALPY(WATER,T=T_cw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
	x_cw[2]:=ENTHALPY(WATER,T=T_cw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))
	X_dot_cw[1]:=m_dot_cw*x_cw[1]
	X_dot_cw[2]:=m_dot_cw*x_cw[2]

	"Chilled Water Side"
	T_chw[1]:=convertTemp(K,C,T_evap_i)
	T_chw[2]:=convertTemp(K,C,T_evap_e)
```

```
      x_chw[1]:=ENTHALPY(WATER,T=T_chw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_chw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
      x_chw[2]:=ENTHALPY(WATER,T=T_chw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_chw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))
      X_dot_chw[1]:=m_dot_chw*x_chw[1]
      X_dot_chw[2]:=m_dot_chw*x_chw[2]

      "Exergy Destroyed by Chiller - Water-side Only"
      Ex_dot_dest:=(X_dot_chw[1]-X_dot_chw[2])+(X_dot_cw[1]-X_dot_cw[2])+Power_chillers
      Ex_dot_in:=Power_chillers+X_dot_chw[1]+X_dot_cw[1]
      Ex_dot_out:=X_dot_chw[2]+X_dot_cw[2]
      eta_chiller:=ETA1(Ex_dot_out,Ex_dot_in)

END
PROCEDURE CHILLER(T_chws_setpoint,T_db_OSA, T_wb_OSA,
P_OSA,T_evap_i,T_cond_i,m_dot_chw, DELTAS_T,Q_leak_eqv,R,
m_dot_cw,Chiller_KW_Demand_Trigger: T_cond_e,T_evap_e,
Power_chillers,Ex_dot_dest,Ex_dot_in,Ex_dot_out,eta_chiller,X_dot_chw[1..2],X_dot_cw[1..2],Q_
evap_demand)
      IF (T_evap_i<=convertTemp(C,K,T_chws_setpoint)) OR (m_dot_chw<10[kg/s])  THEN
            T_cond_e:=T_cond_i
            T_evap_e:=T_evap_i
            Power_chillers:=0[kW]
            Q_evap_demand:=0[kW]
            "! Exergy Calculations"
            "Condensor Water Side"
            T_0=convertTemp(C,K,T_wb_OSA)
            T_cw[1]=convertTemp(K,C,T_cond_i)
            T_cw[2]=convertTemp(K,C,T_cond_e)
            x_cw[1]=ENTHALPY(WATER,T=T_cw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
            x_cw[2]=ENTHALPY(WATER,T=T_cw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))
            X_dot_cw[1]=0[kW]"m_dot_cw*x_cw[1]"
            X_dot_cw[2]=0[kW]"m_dot_cw*x_cw[2]"

            "Chilled Water Side"
            T_chw[1]=convertTemp(K,C,T_evap_i)
            T_chw[2]=convertTemp(K,C,T_evap_e)
            x_chw[1]=ENTHALPY(WATER,T=T_chw[1],x=0)-
ENTHALPY(WATER,T=T_wb_OSA,x=0)-T_0*(Entropy(Water,T=T_chw[1],x=0)-
Entropy(Water,T=T_wb_OSA,x=0))
            x_chw[2]=ENTHALPY(WATER,T=T_chw[2],x=0)-
ENTHALPY(WATER,T=T_wb_OSA,x=0)-T_0*(Entropy(Water,T=T_chw[2],x=0)-
Entropy(Water,T=T_wb_OSA,x=0))
            X_dot_chw[1]=0[kW]
            X_dot_chw[2]=0[kW]

            "Exergy Destroyed by Chiller - Water-side Only"
            Ex_dot_dest=0[kW]
            Ex_dot_in=0[kW]
            Ex_dot_out=0[kW]
            eta_chiller=1[-]
```

```
	ELSE
		CALL CHILLER2(T_chws_setpoint,T_db_OSA, T_wb_OSA,
P_OSA,T_evap_i,T_cond_i, m_dot_chw, DELTAS_T,Q_leak_eqv,R,
m_dot_cw,Chiller_KW_Demand_Trigger: T_cond_e,T_evap_e,
Power_chillers,Ex_dot_dest,Ex_dot_in,Ex_dot_out,eta_chiller,X_dot_chw[1..2],X_dot_cw[1..2],Q_
evap_demand)
	ENDIF

END

"! --------------------------END CHILLER FUNCTIONS--------------------------"

"! --------------------------BEGIN MAH FUNCTIONS--------------------------"
FUNCTION HUMIDIFY(omega_1, P1,T_db1,T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
“Calculates how much humidity is required for MAH”
	omega|star=0[-]
	omega_setpoint_min=HUMRAT(AIRH2O,T=T_db_setpoint_low,r= RH_setpoint_low,P=P1)
	omega_setpoint_inter=HUMRAT(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)
	omega_setpoint_max=HUMRAT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	T_wb_setpoint_min=WETBULB(AIRH2O,T=T_db_setpoint_low, r=RH_setpoint_low,P=P1)
	T_wb_setpoint_inter=WETBULB(AIRH2O,T=T_db_setpoint_high, r=RH_setpoint_low,P=P1)
	T_wb_setpoint_max=WETBULB(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	T_wb=WETBULB(AIRH2O,T=T_db1,w=omega_1,P=P1)

	"Only humidify if W_osa is below minimum."
	IF (omega_1<=omega_setpoint_min) THEN
		IF(T_wb<=T_wb_setpoint_inter) THEN
			omega_2=HUMRAT(AIRH2O,B=T_wb,r= RH_setpoint_low,P=P1)
		ELSE
			IF(T_wb<=T_wb_setpoint_max) THEN
				omega_2=HUMRAT(AIRH2O,T=T_db_setpoint_high,B=T_wb,P=P1)
			ELSE
				IF(T_wb>T_wb_setpoint_max) THEN
					omega_2=omega_setpoint_min
				ENDIF
			ENDIF
		ENDIF
	ELSE
		omega_2=0[-]
	ENDIF
	omega|star=omega_2-omega_1
	"Error Cache"
	IF(omega|star<0) THEN
		omega|star=0
	ENDIF

	HUMIDIFY:=omega|star
END

FUNCTION DEHUMIDIFY(omega_1,P1,T_db, T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
{Calculates how much dehumidification is required for MAH}
```

```
	h_max=ENTHALPY(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	dp_max=DEWPOINT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	q_s_max=ENTHALPY(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)-
ENTHALPY(AIRH2O,T=dp_max,r=1,P=P1)
	h_min=ENTHALPY(AIRH2O,T=T_db_setpoint_low,r= RH_setpoint_low,P=P1)
	dp_min=DEWPOINT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	q_s_min=ENTHALPY(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)-
ENTHALPY(AIRH2O,T=dp_min,r=1,P=P1)

	ADP_min=5.6[C]
	ADP_max=dp_max
	BF=0.2[-]
	h_1=ENTHALPY(AIRH2O,T=T_db,w=omega_1,P=P1)
	dp_1=DEWPOINT(AIRH2O,T=T_db,w=omega_1,P=P1)
	h|star_1=ENTHALPY(AIRH2O,T=dp_1,w=omega_1,P=P1)
	q|star_s=h_1-h|star_1

	CF=1[-]-BF
	ADP=(T_db_setpoint_high-BF*T_db)/CF

	h_adp=ENTHALPY(AIRH2O,T=ADP_min,r= 1,P=P1)
	q_t_max=h_max-h_adp
	q_t_min=h_min-h_adp
	SHR_min=q_s_min/q_t_min
	SHR_max=q_s_max/q_t_max

	omega_setpoint_max=HUMRAT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	"Only dehumidify if W_osa> Wmax"
		IF(omega_1>omega_setpoint_max) THEN
			omega_2=omega_setpoint_max
		ELSE
			omega_2=omega_1
		ENDIF
	DEHUMIDIFY:=omega_2-omega_1
END

FUNCTION T_PREHEAT(omega_1, P1,T_db1,T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
	{Calculates how much preheat is required for MAH}
	T_min=14.4[C]

	omega_setpoint=HUMRAT(AIRH2O,T=T_db_setpoint,r=RH_setpoint,P=P1)
	omega_setpoint_min=HUMRAT(AIRH2O,T=T_db_setpoint_low,r= RH_setpoint_low,P=P1)
	omega_setpoint_inter=HUMRAT(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)
	omega_setpoint_max=HUMRAT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)

	T_wb_setpoint=WETBULB(AIRH2O,T=T_db_setpoint,w=omega_setpoint,P=P1)
	T_wb_setpoint_min=WETBULB(AIRH2O,T=T_db_setpoint_low, r=RH_setpoint_low,P=P1)
	T_wb_setpoint_max=WETBULB(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
	T_wb=WETBULB(AIRH2O,T=T_db1,w=omega_1,P=P1)

	IF((omega_1<=omega_setpoint_min) AND (T_wb<=T_wb_setpoint_min)) THEN
```

```
            T_db|star=TEMPERATURE(AIRH2O,B=T_wb_setpoint_min,w=omega_1,P=P1)
        ELSE
            IF((omega_1<=omega_setpoint_max) AND (T_db1<=T_min)) THEN
                T_db|star=T_min
            ELSE
                T_db|star=T_db1 "no preheat"
            ENDIF
        ENDIF
        T_PREHEAT:=T_db|star
    END

    FUNCTION T_COOL(omega_1, P1, T_db1, T_db_setpoint, T_db_setpoint_low,
    T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
        {Calculates how much cooling is required for MAH}

        omega_setpoint=HUMRAT(AIRH2O,T=T_db_setpoint,r=RH_setpoint,P=P1)
        omega_setpoint_min=HUMRAT(AIRH2O,T=T_db_setpoint_low,r= RH_setpoint_low,P=P1)
        omega_setpoint_inter=HUMRAT(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)
        omega_setpoint_max=HUMRAT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)

        T_wb_setpoint=WETBULB(AIRH2O,T=T_db_setpoint,w=omega_setpoint,P=P1)
        T_wb_setpoint_min=WETBULB(AIRH2O,T=T_db_setpoint_low, r=RH_setpoint_low,P=P1)
        T_wb_setpoint_inter=WETBULB(AIRH2O,T=T_db_setpoint_high, r=RH_setpoint_low,P=P1)

        T_wb_setpoint_max=WETBULB(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)

        T_dp_max=DEWPOINT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
        omega_setpoint_inter=HUMRAT(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)


        ADP_min=5.6[C]
        ADP_max=T_dp_max
        BF=0.2[-]

        CF=1[-]-BF
        ADP=(T_db_setpoint_high-BF*T_db1)/CF

        "W < Wmax, Then process is SHR=1"
        IF(omega_1<=omega_setpoint_max) THEN
            IF((omega_1>=omega_setpoint_inter) AND (T_db1 >=T_db_setpoint_high)) THEN

                T_db|star=T_db_setpoint_high
            ELSE
                T|star=TEMPERATURE(AIRH2O,w=omega_1,r= RH_setpoint_low,P=P1)
                IF((omega_1>=omega_setpoint_min) AND (T_db1 >=T|star)) THEN
                    T_db|star=T|star
                ELSE
                    T_db|star=T_db1
                ENDIF
            ENDIF
        ELSE
            IF(omega_1>omega_setpoint_max) THEN
                T_db|star=T_dp_max
```

```
                { NEED TO ASSUME >75% RH then start Condensing and Cooling
                        T_db|star2=ADP+BF*(T_db1+ADP)}
            ELSE
                T_db|star=T_db1
            ENDIF
        ENDIF
        T_COOL:=T_db|star
END

SUBPROGRAM
T_DEHUMIDIFYSOLVER(T_db1,omega_setpoint_max,ADP_min,ADP_max,P1:T_db|star2)
{Finds Temperature for after dehumidifying for MAH}
        T_db|star2=ADP_min+BF*(T_db1-ADP_min)

        ADP_max=DewPoint(AirH2O,T=T_db|star2,w=omega_setpoint_max,P=P1)
END

FUNCTION T_HUMIDIFY(omega_1, P1, T_db1, T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
        {Calculates temperature for humidificiation for preheat coil in the MAH}
        omega_setpoint=HUMRAT(AIRH2O,T=T_db_setpoint,r=RH_setpoint,P=P1)
        omega_setpoint_max=HUMRAT(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)
        omega_setpoint_min=HUMRAT(AIRH2O,T=T_db_setpoint_low,r= RH_setpoint_low,P=P1)
        omega_setpoint_inter=HUMRAT(AIRH2O,T=T_db_setpoint_high,r= RH_setpoint_low,P=P1)

        T_wb_setpoint=WETBULB(AIRH2O,T=T_db_setpoint,w=omega_setpoint,P=P1)
        T_wb_setpoint_min=WETBULB(AIRH2O,T=T_db_setpoint_low, r=RH_setpoint_low,P=P1)
        T_wb_setpoint_inter=WETBULB(AIRH2O,T=T_db_setpoint_high, r=RH_setpoint_low,P=P1)

        T_wb_setpoint_max=WETBULB(AIRH2O,T=T_db_setpoint_high,r=RH_setpoint_high,P=P1)

        T_wb=WETBULB(AIRH2O,T=T_db1,w=omega_1,P=P1)

        IF(omega_1<=omega_setpoint_min) THEN
            IF (T_wb<=T_wb_setpoint_inter) THEN
                T_db|star=TEMPERATURE(AIRH2O,B=T_wb,r= RH_setpoint_low,P=P1)
            ELSE
                IF(T_wb<=T_wb_setpoint_max) THEN
                    T_db|star=TEMPERATURE(AIRH2O,B=T_wb,w=omega_setpoint_inter,P=P1)
                ELSE
                    IF(T_wb>T_wb_setpoint_max) THEN

        T_db|star=TEMPERATURE(AIRH2O,B=T_wb,w=omega_setpoint_min,P=P1)
                    ELSE
                        T_db|star=T_db1 "no humidification"
                    ENDIF
                ENDIF
            ENDIF
        ELSE
            T_db|star=T_db1 "no humidification"
        ENDIF
        T_HUMIDIFY:=T_db|star
END
```

```
MODULE MAH(T_db_OSA, T_wb_OSA, P_OSA, T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high,
V_dot_air,BkW_design,V_dot_design,ME_design: T_db[1..7],w[1..7],Q_preheat, Q_humidify,
Q_cooling_coil,Q_fan,
BkW,Ex_dot_dest_preheat,eta_heating,Ex_dot_dest_humid,eta_humidify,Ex_dot_dest_cooling,et
a_cooling,MAH_Ex_dot_dest,MAH_Ex_dot_in,MAH_Ex_dot_out,MAH_eta_II,MAH_X_dot_air[1..
2],MAH_X_dot_hw[1..2], MAH_X_dot_huw[1],MAH_X_dot_chw[1..2],MAH_X_dot_cd[1])

	"V_dot_air=4000[ft^3/min]"
	rho_OSA=DENSITY(AIRH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	m_dot_air=V_dot_air*rho_OSA
	w_setpoint=HUMRAT(AIRH2O,T=T_db_setpoint,r=RH_setpoint,P=P_OSA)

	T_wb_setpoint=WETBULB(AIRH2O,T=T_db_setpoint,w=w_setpoint,P=P_OSA)
	h_setpoint=ENTHALPY(AIRH2O,T=T_db_setpoint,w=w_setpoint,P=P_OSA)

	"! Exergy Calculation"
	T_0=convertTemp(C,K,T_db_OSA)
	P_0=P_OSA
	w_0=HUMRAT(AIRH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	RH_0=RELHUM(AIRH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	R_a=0.287[kJ/kg-K]
	R_v=0.461[kJ/kg-K]

	"1. Inlet"
	T_db[1]=T_db_OSA
	P[1]=P_OSA
	RH[1]=RELHUM(AIRH2O,T=T_db[1],B=T_wb[1],P=P[1])
	T_dp[1]=DEWPOINT(AIRH2O,T=T_db[1],B=T_wb[1],P=P[1])
	w[1]=HUMRAT(AIRH2O,T=T_db[1],B=T_wb[1],P=P[1])
	T_wb[1]=T_wb_OSA
	h[1]=ENTHALPY(AIRH2O,T=T_db[1],B=T_wb[1],P=P[1])

	"! Exergy Calculation"   "Per Bejan and Kanoglu et.al."
	c_p_a=Cp(Air,T=T_db[1])
	c_p_w=Cp(Water,T=T_db[1],x=1)
	T_wb_s[1]=T_wb[1]
	s[1]=Entropy(AirH2O,T=T_db[1],B=T_wb_s[1],P=P[1])
	T[1]=convertTemp(C,K,T_db[1])
	x[1]=RND((c_p_a+w[1]*c_p_w)*T_0*(T[1]/T_0-1-
LN(T[1]/T_0))+(1+1.608*w[1])*R_a*T_0*LN(P[1]/P_0)+R_a*T_0*((1+1.608*w[1])*LN((1+1.608*w_
0)/(1+1.608*w[1]))+1.608*w[1]*LN(w[1]/w_0)),3)
	X_dot[1]=m_dot_air*x[1]


	"2. Prefilter"
	T_db[2]=T_db[1]
	RH[2]=RH[1]
	P[2]=P[1]
	T_dp[2]=DEWPOINT(AIRH2O,T=T_db[2],w=w[2],P=P[2])
	w[2]=w[1]
	h[2]=ENTHALPY(AIRH2O,T=T_db[2],w=w[2],P=P[2])
```

```
T_wb[2]=WETBULB(AIRH2O,T=T_db[2],w=w[2],P=P[2])

"! Exergy Calculation"
T_wb_s[2]=T_wb[2]
s[2]=Entropy(AirH2O,T=T_db[2],B=T_wb_s[2],P=P[2])
T[2]=convertTemp(C,K,T_db[2])
x[2]=RND((c_p_a+w[2]*c_p_w)*T_0*(T[2]/T_0-1-LN(T[2]/T_0))+(1+1.608*w[2])*R_a*T_0*LN(P[2]/P_0)+R_a*T_0*((1+1.608*w[2])*LN((1+1.608*w_0)/(1+1.608*w[2]))+1.608*w[2]*LN(w[2]/w_0)),3)
X_dot[2]=m_dot_air*x[2]

"3. Preheat"
P[3]=P[2]
w[3]=w[2]
T_db[3]=T_PREHEAT(w[3],P[3],T_db[2],T_db_setpoint, T_db_setpoint_low, T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
h[3]=ENTHALPY(AIRH2O,T=T_db[3],w=w[3],P=P[3])
RH[3]=RELHUM(AIRH2O,T=T_db[3],w=w[3],P=P[3])
T_dp[3]=DEWPOINT(AIRH2O,T=T_db[3],w=w[3],P=P[3])
T_wb[3]=WETBULB(AIRH2O,T=T_db[3],w=w[3],P=P[3])
Q_preheat=m_dot_air*(h[3]-h[2])

"Determine flow rate for Hot water assuming 20F difference between supply and return"
"This should be enhanced using NTU or LMTD method"
"Assumed T_wb_OSA as reference point"
T_hw[1]=convertTemp(F,C,180[F])
T_hw[2]=convertTemp(F,C,180[F]-20[F])
c_p_hw=Cp(Water,T=T_hw[1],x=0)
Q_preheat=m_dot_hw*c_p_hw*(T_hw[1]-T_hw[2])
T_wb_0=convertTemp(C,K,T_wb_OSA)

x_hw[1]=ENTHALPY(WATER,T=T_hw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-T_0*(Entropy(Water,T=T_hw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
x_hw[2]=ENTHALPY(WATER,T=T_hw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-T_0*(Entropy(Water,T=T_hw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))

X_dot_hw[1]=m_dot_hw*x_hw[1]
X_dot_hw[2]=m_dot_hw*x_hw[2]

"! Exergy Calculation"
T_wb_s[3]=T_wb[3]
s[3]=Entropy(AirH2O,T=T_db[3],B=T_wb_s[3],P=P[3])
T[3]=convertTemp(C,K,T_db[3])
x[3]=RND((c_p_a+w[3]*c_p_w)*T_0*(T[3]/T_0-1-LN(T[3]/T_0))+(1+1.608*w[3])*R_a*T_0*LN(P[3]/P_0)+R_a*T_0*((1+1.608*w[3])*LN((1+1.608*w_0)/(1+1.608*w[3]))+1.608*w[3]*LN(w[3]/w_0)),3)
X_dot[3]=m_dot_air*x[3]

"Exergy Destroyed by Preheating Air - Sensible Only"
Ex_dot_dest_preheat=RND(X_dot[2]+X_dot_hw[1]-X_dot[3]-X_dot_hw[2],3)
Ex_dot_out_preheat=RND(X_dot[3]+X_dot_hw[2],3)
Ex_dot_in_preheat=RND(X_dot[2]+X_dot_hw[1],3)
eta_heating=ETA1(Ex_dot_out_preheat,Ex_dot_in_preheat)
```

```
	"4. Humidifier - Spray"
	"Cooling with Humidification"
	T_w=12.8[C]
	T_db[4]=T_HUMIDIFY(w[3],P[3],T_db[3],T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
	w[4]=w[3]+HUMIDIFY(w[3],P[3],T_db[3],T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
	P[4]=P[3]
	h[4]=ENTHALPY(AIRH2O,T=T_db[4],w=w[4],P=P[4])
	h_fg=ENTHALPY(WATER,T=T_w,x=0)-ENTHALPY(WATER,T=T_w,x=1)
	Q_humidify=m_dot_air*HUMIDIFY(w[3],P[3],T_db[3],T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)*h_fg
	RH[4]=RELHUM(AIRH2O,T=T_db[4],w=w[4],P=P[4])
	T_dp[4]=DEWPOINT(AIRH2O,T=T_db[4],w=w[4],P=P[4])
	T_wb[4]=WETBULB(AIRH2O,T=T_db[4],w=w[4],P=P[4])

	"! Exergy Calculation"
	T_wb_s[4]=T_wb[4]
	s[4]=Entropy(AirH2O,T=T_db[4],B=T_wb_s[4],P=P[4])
	T[4]=convertTemp(C,K,T_db[4])
	x[4]=RND((c_p_a+w[4]*c_p_w)*T_0*(T[4]/T_0-1-
LN(T[4]/T_0))+(1+1.608*w[4])*R_a*T_0*LN(P[4]/P_0)+R_a*T_0*((1+1.608*w[4])*LN((1+1.608*w_
0)/(1+1.608*w[4]))+1.608*w[4]*LN(w[4]/w_0)),3)
	x_w[4]=RND(ENTHALPY(WATER,T=T_w,x=0)-ENTHALPY(WATER,T=T_db_OSA,x=1)-
T_0*Entropy(Water,T=T_w,x=0)+T_0*Entropy(Water,T=T_db_OSA,x=1)-R_v*T_0*LN(RH_0),3)
	X_dot_w[4]=m_dot_air*(w[4]-w[3])*x_w[4]
	X_dot[4]=m_dot_air*x[4]

	"Exergy Destroyed by Adiabic Evaporative Cooling"
	Ex_dot_dest_humid=RND(X_dot[3]+X_dot_w[4]-X_dot[4],3)
	Ex_dot_in_humid=RND(X_dot[3]+X_dot_w[4],3)
	Ex_dot_out_humid=RND(X_dot[4],3)
	eta_humidify=ETA1(Ex_dot_out_humid,Ex_dot_in_humid)

	"5. Cooling Coil + Dehumidification"
	T_CC_ADP=convertTemp(F,C,42[F])
	T_db[5]=T_COOL(w[4],P[4],T_db[4], T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
	P[5]=P[4]
	w[5]=w[4]+DEHUMIDIFY(w[4],P[4],T_db[4], T_db_setpoint, T_db_setpoint_low,
T_db_setpoint_high, RH_setpoint, RH_setpoint_low,  RH_setpoint_high)
	h[5]=ENTHALPY(AIRH2O,T=T_db[5],w=w[5],P=P[5])
	RH[5]=RELHUM(AIRH2O,T=T_db[5],w=w[5],P=P[5])
	T_dp[5]=DEWPOINT(AIRH2O,T=T_db[5],w=w[5],P=P[5])
	T_wb[5]=WETBULB(AIRH2O,T=T_db[5],w=w[5],P=P[5])
	Q_cooling_coil=m_dot_air*(h[5]-h[4])+m_dot_air*(w[5]-
w[4])*(ENTHALPY(WATER,T=T_dp[5],x=0)-ENTHALPY(WATER,T=T_CC_ADP,x=0))
	c_p_cw=Cp(Water,T=T_CC_ADP,x=0)
	T_cw[1]=convertTemp(F,C,42[F])
	T_cw[2]=convertTemp(F,C,42[F]+12[F]) "assume 12F rise"
	-Q_cooling_coil=m_dot_cw*c_p_cw*(convertTemp(C,K,T_cw[2])-convertTemp(C,K,T_cw[1]))
```

```
	"! Exergy Calculation"
	T_wb_s[5]=T_wb[5]
	s[5]=Entropy(AirH2O,T=T_db[5],B=T_wb_s[5],P=P[5])
	T[5]=convertTemp(C,K,T_db[5])
	x[5]=RND((c_p_a+w[5]*c_p_w)*T_0*(T[5]/T_0-1-
LN(T[5]/T_0))+(1+1.608*w[5])*R_a*T_0*LN(P[5]/P_0)+R_a*T_0*((1+1.608*w[5])*LN((1+1.608*w_
0)/(1+1.608*w[5]))+1.608*w[5]*LN(w[5]/w_0)),3)
	X_dot[5]=m_dot_air*x[5]

	x_cw[1]=ENTHALPY(WATER,T=T_cw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
	x_cw[2]=ENTHALPY(WATER,T=T_cw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))

	X_dot_cw[1]=RND(m_dot_cw,3)*x_cw[1]
	X_dot_cw[2]=RND(m_dot_cw,3)*x_cw[2]

	x_w[5]=ENTHALPY(WATER,T=T_CC_ADP,x=0)-ENTHALPY(WATER,T=T_db_OSA,x=1)-
T_0*Entropy(Water,T=T_CC_ADP,x=0)+T_0*Entropy(Water,T=T_db_OSA,x=1)-
R_v*T_0*LN(RH_0)

	X_dot_w[5]=RND(m_dot_air*(w[5]-w[4])*x_w[5],3)

	"Exergy Destroyed by Cooling Air"
	Ex_dot_dest_cooling=RND(X_dot[4]+X_dot_cw[1]-X_dot_cw[2]-X_dot[5]-X_dot_w[5],3)
	Ex_dot_in_cooling=RND(X_dot[4]+X_dot_cw[1],3)
	Ex_dot_out_cooling=RND(X_dot_cw[2]+X_dot[5]+X_dot_w[5],3)
	eta_cooling=ETA2(Ex_dot_out_cooling,Ex_dot_in_cooling)

	"6. Fan"
	V_dot_air[6]=m_dot_air/Density(AirH2O,T=T_db[5],w=w[5],P=P[5])
	"BkW_design=7.5[hp]*convert(hp,kW)""!LINK TO MAH SETTINGS"
	"V_dot_design=4000[cfm]*convert(cfm,m^3/s)""!LINK TO MAH SETTINGS"
	"ME_design=0.93[-]""!LINK TO MAH SETTINGS"
	AkW=ME_design*BkW_design*(V_dot_air[6]/V_dot_design)^3"!LINK TO MAH SETTINGS"
	BkW=BkW_design*(V_dot_air[6]/V_dot_design)^3"!LINK TO MAH SETTINGS"
	DELTAT_FkW=(BkW-AkW)/(m_dot_air*c_p_a)"!LINK TO MAH SETTINGS"

	T_db[6]=convertTemp(K,C,convertTemp(C,K,T_db[5])+DELTAT_FkW)
	w[6]=w[5]
	P[6]=P[5]

	RH[6]=RELHUM(AIRH2O,T=T_db[6],w=w[6],P=P[6])
	h[6]=ENTHALPY(AIRH2O,T=T_db[6],w=w[6],P=P[6])
	Q_fan=m_dot_air*(h[6]-h[5])
	T_dp[6]=DEWPOINT(AIRH2O,T=T_db[6],w=w[6],P=P[6])
	T_wb[6]=WETBULB(AIRH2O,T=T_db[6],w=w[6],P=P[6])

	"! Exergy Calculation"
	T_wb_s[6]=T_wb[6]-.0001[C]"offset T_wb  by -.01C because EES can not solve when
Tdb=Twb for entrophy"
	s[6]=Entropy(AirH2O,T=T_db[6],B=T_wb_s[6],P=P[6])
	T[6]=convertTemp(C,K,T_db[6])
```

```
	x[6]=(c_p_a+w[6]*c_p_w)*T_0*(T[6]/T_0-1-LN(T[6]/T_0))+(1+1.608*w[6])*R_a*T_0*LN(P[6]/P_0)+R_a*T_0*((1+1.608*w[6])*LN((1+1.608*w_0)/(1+1.608*w[6]))+1.608*w[6]*LN(w[6]/w_0))
	X_dot[6]=m_dot_air*x[6]
	"Exergy Destroyed by Cooling Air "
	Ex_dot_dest_fan=RND(X_dot[6]-X_dot[5],3)
	Ex_dot_in_fan=RND(X_dot[5]+BkW,3)
	Ex_dot_out_fan=RND(X_dot[6]+X_dot_w[5],3)
	eta_fan=ETA1(Ex_dot_out_fan,Ex_dot_in_fan)

	"7. Final Filter"
	T_db[7]=T_db[6]
	P[7]=P[6]
	w[7]=w[6]
	T_dp[7]=DEWPOINT(AIRH2O,T=T_db[7],w=w[7],P=P[7])
	RH[7]=RELHUM(AIRH2O,T=T_db[7],w=w[7],P=P[7])
	h[7]=ENTHALPY(AIRH2O,T=T_db[7],w=w[7],P=P[7])
	T_wb[7]=WETBULB(AIRH2O,T=T_db[7],w=w[7],P=P[7])

	MAH_Ex_dot_dest=Ex_dot_dest_fan+Ex_dot_dest_cooling+Ex_dot_dest_humid+Ex_dot_dest_preheat+BkW
	MAH_Ex_dot_in=X_dot[1]+X_dot_hw[1]+X_dot_w[4]+X_dot_cw[1]+BkW
	MAH_Ex_dot_out=X_dot[6]+X_dot_hw[2]+X_dot_cw[2]+X_dot_w[5]
	MAH_eta_II=ETA1(MAH_Ex_dot_out,MAH_Ex_dot_in)
	MAH_X_dot_air[1..2]=[X_dot[1],X_dot[6]]
	MAH_X_dot_hw[1..2]=[X_dot_hw[1],X_dot_hw[2]]
	MAH_X_dot_huw[1]=X_dot_w[4]
	MAH_X_dot_chw[1..2]=[X_dot_cw[1],X_dot_cw[2]]
	MAH_X_dot_cd[1]=X_dot_w[5]

END
"! --------------------------END MAH FUNCTIONS--------------------------"
"! --------------------------BEGIN  CRAH  FUNCTIONS--------------------------"

MODULE CRAH(T_db_OSA, T_wb_OSA, P_OSA,Airflow,omega_RA,T_RA,T_SA,T_chws,P_RA,P_chw,DELTAT_CHW,CRAH_Qty,CRAH_AirFLow,CRAH_BkW:m_dot_chws,T_chwr,Total_BkW,Ex_dot_dest_cooling,Ex_dot_in_cooling,Ex_dot_out_cooling,eta_cooling,CRAH_X_dot_air[1..2],CRAH_X_dot_chw[1..2])
	{
	Things to do
		1) Convert logic for precision conditioning,
		2) Add bypass flow around coil
		3) Add ESP (~0.3 in WC) and SP (0.5 in WC) calcs to estimate BHP
		4) Add Pressure drop accross filter - 30% ASHRAE
		5) Add Humidification
	}
	"! Exergy Calculation"
	T_0=convertTemp(C,K,T_db_OSA)
	P_0=P_OSA
	w_0=HUMRAT(AIRH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	RH_0=RELHUM(AIRH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	R_a=0.287[kJ/kg-K]
	R_v=0.461[kJ/kg-K]
```

```
"Air Side Properties - State 1"
T_db[1]=T_RA
w[1]=omega_RA
P[1]=P_RA
RH[1]=RELHUM(AIRH2O,T=T_db[1],w=w[1],P=P[1])
T_dp[1]=DEWPOINT(AIRH2O,T=T_db[1],w=w[1],P=P[1])
h[1]=ENTHALPY(AIRH2O,T=T_db[1],w=w[1],P=P[1])
rho_ra[1]=Density(AirH2O,T=T_db[1],w=w[1],P=P[1])
FlowRate_ra=Airflow
m_dot_air=FlowRate_ra*rho_ra[1]
Q[1]=m_dot_air*h[1]

"! Exergy Calculation"   "Per Bejan and Kanoglu et.al."
c_p_a=Cp(Air,T=T_db[1])
c_p_w=Cp(Water,T=T_db[1],x=1)
s[1]=Entropy(AirH2O,T=T_db[1],w=w[1],P=P[1])
T[1]=convertTemp(C,K,T_db[1])
x[1]=RND((c_p_a+w[1]*c_p_w)*T_0*(T[1]/T_0-1-LN(T[1]/T_0))+(1+1.608*w[1])*R_a*T_0*LN(P[1]/P_0)+R_a*T_0*((1+1.608*w[1])*LN((1+1.608*w_0)/(1+1.608*w[1]))+1.608*w[1]*LN(w[1]/w_0)),3)
X_dot[1]=m_dot_air*x[1]

"State 2"
P[2]=P[1]
T_db[2]=T_SA-DELTAT_motor
w[2]=w[1] "Assume no dehumidification"
h[2]=Enthalpy(AirH2O,T=T_db[2],w=w[2],P=P[2])
Q[2]=m_dot_air*h[2]
Q_coil=Q[1]-Q[2]-Q_m

"! Exergy Calculation"   "Per Bejan and Kanoglu et.al."
s[2]=Entropy(AirH2O,T=T_db[2],w=w[2],P=P[2])
T[2]=convertTemp(C,K,T_db[2])
x[2]=RND((c_p_a+w[2]*c_p_w)*T_0*(T[2]/T_0-1-LN(T[2]/T_0))+(1+1.608*w[2])*R_a*T_0*LN(P[2]/P_0)+R_a*T_0*((1+1.608*w[2])*LN((1+1.608*w_0)/(1+1.608*w[2]))+1.608*w[2]*LN(w[2]/w_0)),3)
X_dot[2]=m_dot_air*x[2]

"Motor Heat"
BkW_per_CRAH=CRAH_BkW*((FlowRate_ra/CRAH_Qty)/(CRAH_AirFLow*convert(L/s,m^3/s)))^3
ME=0.72[-]
AkW=ME*BkW_per_CRAH
FkW=BkW_per_CRAH*(1-ME)
Q_m=FkW*CRAH_Qty
Total_BkW=CRAH_Qty*BkW_per_CRAH
DELTAT_motor=BkW_per_CRAH*(1-ME)/(c_p_a*m_dot_air/CRAH_Qty)
"Q_T=Q_coil+Q_m"

"Chilled Water Flow"
P_chw[1]=P_chw
"P_chw[2]=P_chw[1]+DELTAP_chw"
```

```
	T_chwr=T_chws+DELTAT_CHW
	cp_chw=Cp(Water,T=T_chws,x=0)
	Q_coil=m_dot_chws*(Enthalpy(Water,T=T_chwr,x=0)-Enthalpy(Water,T=T_chws,x=0))

	"! Exergy Calculation"	"Per Bejan and Kanoglu et.al."
	T_cw[1]=T_chws
	T_cw[2]=T_chwr
	x_cw[1]=ENTHALPY(WATER,T=T_cw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
	x_cw[2]=ENTHALPY(WATER,T=T_cw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))
	X_dot_cw[1]=m_dot_chws*x_cw[1]
	X_dot_cw[2]=m_dot_chws*x_cw[2]

	T_CC_ADP=T_cw[1]
	x_w[3]=ENTHALPY(WATER,T=T_CC_ADP,x=0)-ENTHALPY(WATER,T=T_db_OSA,x=1)-
T_0*Entropy(Water,T=T_CC_ADP,x=0)+T_0*Entropy(Water,T=T_db_OSA,x=1)-
R_v*T_0*LN(RH_0)
	X_dot_w[3]=RND(m_dot_air*(w[2]-w[1])*x_w[3],3)

	CRAH_X_dot_air[1]=X_dot[1]
	CRAH_X_dot_air[2]=X_dot[2]
	CRAH_X_dot_chw[1]=X_dot_cw[1]
	CRAH_X_dot_chw[2]=X_dot_cw[2]
	"Exergy Destroyed by Cooling Air"
	Ex_dot_dest_cooling=(X_dot[1]-X_dot[2])+(X_dot_cw[1]-X_dot_cw[2])+Total_BkW
	Ex_dot_out_cooling=X_dot_cw[2]+X_dot[2]+X_dot_w[3]
	Ex_dot_in_cooling=X_dot[1]+X_dot_cw[1]+Total_BkW
	eta_cooling=ETA2(Ex_dot_dest_cooling,Ex_dot_in_cooling)
	"rho_chw=Density(Water,T=T_chws,P=P_chw[1])"
	"V_dot_chws=(m_dot_chws/rho_chw)*convert(ft^3/hr,gpm)"

	"Curve Fitted Pressure drop as a function of GPM, based on FH600C"
	"DELTAP_chw=0.0087*V_dot_chws^1.8386"

END

"! -------------------------END  CRAH  FUNCTIONS-------------------------"

"! -------------------------BEGIN WEATHER BIN FUNCTIONS-------------------------"
PROCEDURE WEATHERBINDATAMAXTEMP(P_OSA: T_db_OSA_max,
T_wb_OSA_max,RH_OSA_max, BinMonth$, BinDay,BinHourMax)
	{Retrieves selected data from lookup table and calculates RH}
	i:=0
	NRows:=8760
	T_db_OSA_max:=-999
	T_wb_OSA_max:=-999
	Repeat
		i:=i+1
		T_db_OSA=Lookup('WEATHER',i,'Dry Bulb')
		IF(T_db_OSA>T_db_OSA_max) THEN
			T_db_OSA_max:=T_db_OSA
			T_wb_OSA_max:=Lookup('WEATHER',i,'Wet Bulb')
```

```
                RH_OSA_max:=RelHum(AirH2O,T=T_db_OSA,B=T_wb_OSA_max,P=P_OSA)
                BinMonth$:=Lookup$('WEATHER',i,'Month')
                BinDay:=Lookup('WEATHER',i,'Day')
                BinHourMax:=i
            ENDIF
        until (i>=NRows)

END
PROCEDURE WEATHERBINDATA(City$,BinHour, P_OSA: T_db_OSA, T_wb_OSA,RH_OSA,
BinMonth$, BinDay)
        {Retrieves selected data from lookup table and calculates RH
            Data from Table is in ENG units. This program converts to SI}
        T_db_OSA:=convertTemp(F,C,Lookup(City$,BinHour,'Dry Bulb'))
        T_wb_OSA:=convertTemp(F,C,Lookup(City$,BinHour,'Wet Bulb'))
        IF (T_db_OSA<=T_wb_OSA) THEN "EES CANNOT SOLVE IF TWB>=TDB"
            T_wb_OSA:=T_db_OSA-.0001[C]
        ENDIF
        RH_OSA:=RelHum(AirH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
        BinMonth$:=Lookup$(City$,BinHour,'Month')
        BinDay:=Lookup(City$,BinHour,'Day')
END
FUNCTION GETALTITUDEDATA(City$)
        {Retrieves selected data from lookup table and returns altitude}
        i=0
        N=15
        Repeat
            i:=i+1
            CityName$=Lookup$('CITYLIST',i,'City')
            IF (CityName$=City$) THEN
                GETALTITUDEDATA:=Lookup('CITYLIST',i,'Altitude')
            ENDIF
        Until (i>=N)
END
FUNCTION BAROMETRICPRESSURE(ALTITUDE)
        { ASHRAE 2005 FUNDAMENTALS CHAPTER 6., equation 6.3 }
        BAROMETRICPRESSURE:=14.696*(1-6.8754*10^(-6)*ALTITUDE)^(5.2559)
END
"! --------------------------END WEATHER BIN FUNCTIONS--------------------------"

"! --------------------------BEGIN COOLING LOAD CALCULATIONS-------------------------"
PROCEDURE DATACENTERLOAD(NRows, T_db_setpoint, P_OSA, RH_setpoint: Airflow,
HeatLoad,Racks)
        {
        Calculates and sets values for the DC equipment lookup table
        Future: Add PDU Heat loads, Lighting, and people.
        }

        P=P_OSA
        cp=Cp(AirH2O,T=T_db_setpoint,r=RH_setpoint,P=P)
        rho=Density(AirH2O,T=T_db_setpoint,r=RH_setpoint,P=P)

        i:=0
        Airflow:=0
```

```
        HeatLoad:=0
    Racks:=0
        repeat
      i:=i+1

                Power=Lookup('DC Equipment',i,'Cabinet Rating')
                DeltaT=Lookup('DC Equipment',i,'Temperature Rise')
                Qty=Lookup('DC Equipment',i,'Quantity')

        Lookup('DC Equipment',i,'Airflow/Cabinet')=Power/(rho*cp*DeltaT)
                Lookup('DC Equipment',i,'Total Airflow')=Power*Qty/(rho*cp*DeltaT)
                Lookup('DC Equipment',i,'Total Power')=Power*Qty
                Airflow:=Airflow+Power/(rho*cp*DeltaT)*Qty
                HeatLoad:=HeatLoad+Power*Qty
                Racks:=Racks+Qty
    until (i>=NRows)
END
"! -------------------------END COOLING LOAD CALCULATIONS-------------------------"
"! -------------------------BEGIN EXERGY FUNCTIONS-------------------------"
FUNCTION FLOWEXERGY(T_o, T_wb_o, P_o,s,h)
        omega_o=HumRat(AirH2O,T=T_o,B=T_wb_o,P=P_o)
        h_o=ENTHALPY(AirH2O,T=T_o,B=T_wb_o,P=P_o)
        s_o=Entropy(AirH2O,T=T_o,B=T_wb_o,P=P_o)
        {Flow Exergy}
        FLOWEXERGY:=(h-h_o)-ConvertTemp(F,R,T_o)*(s-s_o)
END

FUNCTION WATERFLOWEXERGY(T_o, T_wb_o, P_o,T)
        h_o=ENTHALPY(Water,T=T_wb_o,P=P_o)
        s_o=Entropy(Water,T=T_wb_o,P=P_o)
        h=ENTHALPY(Water,T=T,x=0)
        s=Entropy(Water,T=T,x=0)
        {Flow Exergy}
        WATERFLOWEXERGY:=(h-h_o)-ConvertTemp(F,R,T_o)*(s-s_o)
END
"! -------------------------END EXERGY FUNCTIONS-------------------------"

"! -------------------------BEGIN COOLING TOWER FUNCTIONS-------------------------"

"! -------------------------BEGIN COOLING TOWER FUNCTIONS-------------------------"

PROCEDURE
Muangnoi(L,T_db_i,T_wb_i,T_w_i,T_cw_setpoint,A,Ka,P1,Le,Z,N:G,T_db_e,T_wb_e,T_w_e,X_d
es,X_w_i,X_w_e,X_air_i,X_air_e,Q_rejected,IW_makeup)
{!      An exergy analysis on the performance of a counterflow wet cooling tower
        T. Muangnoi et al. / Applied Thermal Engineering 27 (2007) 910-917}

        "Set Step Size to intgrate through Tower Hieght"
        dZ=Z/(N-1[-])

        "Specific Heats"
        cp_a=Cp(Air,T=T_db_i) "Dry Air"
        cp_w=Cp(Water,T=T_w_i,x=0) "Water"
```

```
cp_v=Cp(Water,T=T_db_i,x=1) "Water Vapor"
T_w_inlet=T_w_i
T_db[1]=T_db_i
T_wb[1]=T_wb_i
w[1]=HumRat(AirH2O,T=T_db_i,B=T_wb_i,P=P1)
h[1]=Enthalpy(AirH2O,T=T_db_i,B=T_wb_i,P=P1)
rh[1]=RelHum(AirH2O,T=T_db_i,B=T_wb_i,P=P1)
rho[1]=Density(AirH2O,T=T_db_i,B=T_wb_i,P=P1)

"Convergence Criteria"
eta|star_L=5E-9[kg/s]
eta|star_G=5E-9[kg/s]
MaxIter=100

"Exergy Calculations"
"Restricted Dead State"
T_0=T_db_i
P_0=P1
rh_0=RelHum(AirH2O,T=T_db_i,B=T_wb_i,P=P_0)
w_0=HumRat(AirH2O,T=T_db_i,r=rh[1],P=P_0)

h_f_0=Enthalpy(Water,T=T_0,x=0)
s_f_0=Entropy(Water,T=T_0,x=0)
"Gas Constants"
R_v=0.4615[kJ/kg-K] "Water Vapor"
R_a=0.28703[kJ/kg-K] "Dry Air"

"Dead State"
"T_00=T_wb_i"
"rh_00=0.8[-]"
rh_00=rh_0
"w_00=HumRat(AirH2O,T=T_0,r=rh_00,P=P_0)"
w_00=w_0
"Initial Guess"
L[1]=L-.01[kg/s]
Z[1]=0
j=1
REPEAT
    Z[j+1]=Z[j]+dZ
    j=j+1
UNTIL (j>N)

T_cw_setpoint_new=T_cw_setpoint
    "LG=2"
"Initital Guess at G"
G_max=A*rho[1]*1.75[m/s]

"REPEAT"
    k=1
    T_w[1]=T_cw_setpoint_new
    Q_w=L*cp_w*(T_w_i-T_w[1])
    h[N]=Enthalpy(AirH2O,T=T_w_i,B=T_w_i,P=P1)
    G=Q_w/(h[N]-h[1])
```

```
{IF G_guess>G_max THEN
      G=G_max
ELSE
      G=G_guess
ENDIF}
REPEAT

      i=1

      REPEAT

            j=1

            REPEAT

                  "Saturated humidity ratio elevated at T_w"
                  w_s_w[j]=HumRat(AirH2O,T=T_w[j],B=T_w[j],P=P1)
                  "Saturated enthalpy of water vapor elevated at T_w"
                  h_g_w[j]=Enthalpy(Water,T=T_w[j],x=1)

                  "Equation 5: Change of Air Enthalpy"
                  dh[j]=(Ka*A)/G*(Le*cp_a*(T_w[j]-T_db[j])+h_g_w[j]*(w_s_w[j]-w[j]))*dZ
                  h[j+1]=h[j]+dh[j]

                  "Equation 6: Change of Humidity Ratio"
                  dw[j]=(Ka*A)/G*(w_s_w[j]-w[j])*dZ
                  w[j+1]=w[j]+dw[j]

                  "Equation 7: Change of Water Temperature"
                  h_f_w=Enthalpy(Water,T=T_w[j],x=0)
                  "cp_w=Cp(Water,T=T_w[j],x=0)"
                  dT_w[j]=G/(L[j]*cp_w)*(dh[j]-h_f_w*dw[j])
                  T_w[j+1]=T_w[j]+dT_w[j]

                  T_db[j+1]=Temperature(AirH2O,h=h[j+1],w=w[j+1],P=P1)
                  T_wb[j+1]=WetBulb(AirH2O,T=T_db[j+1],w=w[j+1],P=P1)

                  {T_wb[j+1]=WetBulb(AirH2O,h=h[j+1],w=w[j+1],P=P1)
                  T_db[j+1]=Temperature(AirH2O,B=T_wb[j+1],w=w[j+1],P=P1)}

                  rh[j+1]=RelHum(AirH2O,T=T_db[j+1],B=T_wb[j+1],P=P1)
                  L[j+1]=L[j]+G*(w[j+1]-w[j])

                  "Exergy Calculations"
                  T0=convertTemp(C,K,T_0)

                  "Water Side"
                  s_f_w[j]=Entropy(Water,T=T_w[j],x=0)
                  X_w[j]=L[j]*((h_f_w-h_f_0)-T0*(s_f_w[j]-s_f_0)-R_v*T0*ln(rh_00))

                  "Air Side"
                  T=convertTemp(C,K,T_db[j])
                  T0=convertTemp(C,K,T_0)
```

```
                    X_air[j]=G*((cp_a-w[j]*cp_v)*(T-T0-
T0*ln(T/T0))+R_a*T0*((1+1.608*w[j])*ln((1+1.608*w_00)/(1+1.608*w[j]))+1.608*w[j]*ln(w[j]/w_00))
)

                    {IF (T_w[j]>T_w_i) THEN
                        T_w[j]=T_w_i
                    ENDIF}

                    j=j+1

                UNTIL(j>N)

                dL_error=L-L[N]
                L[1]=L[1]+dL_error
                i=i+1

            UNTIL (abs(L[N]-L)<=eta|star_L)

            T_error=T_w[N]-T_w_i
            Q_error=L*cp_w*T_error
            dh_error=h[1]-h[N]
            dw_error=w[1]-w[N]
            h_f_w=Enthalpy(Water,T=average(T_w[N],T_w[1]),x=0)
            G_error=Q_error/(dh_error-h_f_w*dw_error)
            "G_error=Q_error/dh_error"

            G=G+G_error/2

            IF G>G_Max THEN
                G=G_Max
                G_error=0
            ENDIF

            k=k+1
        UNTIL ((ABS(G_error)<=eta|star_G) OR (k>MaxIter))
            {Reset Leaving Water Temperature because Fan is Maxed out}

        "T_cw_setpoint_new=T_w[1]+0.1[C]"
        {IF (T_cw_setpoint_new+1.5)>T_w_i THEN
            G=G_Max
            T_w_i=T_w_inlet+2
            T_cw_setpoint_new=T_cw_setpoint
        ENDIF}
    "UNTIL (G<G_Max)"

    X_w_i=X_w[N]
    X_w_e=X_w[1]
    X_air_i=X_air[1]
    X_air_e=X_air[N]
    X_des=((X_w_i-X_w_e)+(X_air_i-X_air_e))

    T_db_e=T_db[N]
    T_wb_e=T_wb[N]
```

```
        T_w_e=T_w[1]; T_w_N=T_w[N]
        CT_RANGE=T_w_N-T_w_e
        CT_INLET_ERROR=T_w_N-T_w_inlet
        Q_rejected=G*(h[N]-h[1])
        IW_makeup=L-L[1]
END

Procedure
COOLINGTOWER(L,T_db_i,T_wb_i,P_OSA,Ka,Cell_Area,Fill_Depth,Cell_Qty,Cell_BkW,T_w_i,
T_cw_setpoint,CT_Approach_Min:G,T_w_e,X_des,X_w_i,X_w_e,X_air_i,X_air_e,X_dot_IW,CT_
Fan_Power,IW_makeup)
        {Adjusts T_cw Setpoints until Massflow is below a predetermined threshold of assumed
2.5m/s max face velocity }

        NewT_cw_setpoint=T_cw_setpoint

        IF (T_wb_i+CT_Approach_Min)>T_cw_setpoint THEN
                NewT_cw_setpoint=T_wb_i+CT_Approach_Min
        ENDIF

        A=Cell_Area*Cell_Qty
        N=10 "Number of Descrete Points"
        Le=1.00[-]
        Z=Fill_Depth
        IF (T_w_i>(NewT_cw_setpoint+2)) THEN
                CALL
Muangnoi(L,T_db_i,T_wb_i,T_w_i,NewT_cw_setpoint,A,Ka,P_OSA,Le,Z,N:G,T_db_e,T_wb_e,T_
w_e,X_des,X_w_i,X_w_e,X_air_i,X_air_e,Q_rejected,IW_makeup)
        ELSE
                G=0; T_db_e=T_db_i; T_wb_e=T_wb_i; T_w_e=NewT_cw_setpoint
                X_des=0; X_w_i=0; X_w_e=0; X_air_i=0; X_air_e=0; Q_rejected=0; IW_makeup=0;
        ENDIF
        "Fan Power Calculations - Fan Simularity Laws"
        G_ACM=G/Density(AirH2O,T=T_db_i,B=T_wb_i,P=P_OSA)

        "Airflow Per Cell"
        CellACM=G_ACM/Cell_Qty
        Cell_MaxACM=Cell_Area*1.75[m/s]
        "Power Per Cell"
        Cell_Actual_BkW=Cell_BkW*(CellACM/Cell_MaxACM)^3
        G_ACM_CFM=CellACM*convert(m^3/s,cfm)
        CT_Fan_Power=Cell_Actual_BkW*Cell_Qty

        T_0=convertTemp(C,K,T_db_i)
        T_iw=convertTemp(F,C,55[F])
        x_iw=ENTHALPY(WATER,T=T_iw,x=0)-ENTHALPY(WATER,T=T_db_i,x=0)-
T_0*(Entropy(Water,T=T_iw,x=0)-Entropy(Water,T=T_db_i,x=0))
        X_dot_IW=IW_makeup*x_iw
        X_des=X_des+CT_Fan_Power

END
"! -------------------------END  COOLING TOWER FUNCTIONS-------------------------"
```

```
"! --------------------------BEGIN PLATE AND FRAME HEAT EXCHANGER FUNCTIONS---------------
-----------"

PROCEDURE
PlateFrameHX(N_s,T_db_OSA,T_w_i,T_c_i,m_dot_w,m_dot_c:T_w_o,T_c_o,q,Ex_dot_dest,Ex_
dot_in,Ex_dot_out,eta_HX)
	FLOWCHECK=min(m_dot_w,m_dot_c)
	IF (FLOWCHECK>1) AND (T_w_i>T_c_i) THEN
		CALL PlateFrameHX2(N_s,T_w_i,T_c_i,m_dot_w,m_dot_c:T_w_o,T_c_o,q)

		"! Exergy Calculations"
		"Condensor Water Side"
		T_0=convertTemp(C,K,T_db_OSA)"Wet bulb is the coldest temperature that water can
achieve"
		T_cw[1]=T_c_i;		T_cw[2]=T_c_o; m_dot_cw=m_dot_c
		x_cw[1]=ENTHALPY(WATER,T=T_cw[1],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[1],x=0)-Entropy(Water,T=T_db_OSA,x=0))
		x_cw[2]=ENTHALPY(WATER,T=T_cw[2],x=0)-ENTHALPY(WATER,T=T_db_OSA,x=0)-
T_0*(Entropy(Water,T=T_cw[2],x=0)-Entropy(Water,T=T_db_OSA,x=0))
		X_dot_cw[1]=m_dot_cw*x_cw[1]
		X_dot_cw[2]=m_dot_cw*x_cw[2]

		"Chilled Water Side"
		T_chw[1]=T_w_i;		T_chw[2]=T_w_o;  m_dot_chw=m_dot_w
		x_chw[1]=ENTHALPY(WATER,T=T_chw[1],x=0)-
ENTHALPY(WATER,T=T_db_OSA,x=0)-T_0*(Entropy(Water,T=T_chw[1],x=0)-
Entropy(Water,T=T_db_OSA,x=0))
		x_chw[2]=ENTHALPY(WATER,T=T_chw[2],x=0)-
ENTHALPY(WATER,T=T_db_OSA,x=0)-T_0*(Entropy(Water,T=T_chw[2],x=0)-
Entropy(Water,T=T_db_OSA,x=0))
		X_dot_chw[1]=m_dot_chw*x_chw[1]
		X_dot_chw[2]=m_dot_chw*x_chw[2]

		"Exergy Destroyed by HX"
		Ex_dot_dest=(X_dot_chw[1]-X_dot_chw[2])+(X_dot_cw[1]-X_dot_cw[2])
		Ex_dot_in=X_dot_chw[1]+X_dot_cw[1]
		Ex_dot_out=X_dot_chw[2]+X_dot_cw[2]
		eta_HX=ETA2(Ex_dot_dest,Ex_dot_in)

	ELSE
		T_w_o:=T_w_i
		T_c_o:=T_c_i
		q:=0[kW]
		Ex_dot_dest:=0[kW]
		Ex_dot_in:=0[kW]
		Ex_dot_out:=0[kW]
		eta_HX:=1[-]
	ENDIF

END

PROCEDURE PlateFrameHX2(N_s,T_w_i,T_c_i,m_dot_w,m_dot_c:T_w_o,T_c_o,q)
```

```
"! Counter Flow Plate and Frame Calculations Using Modified LMTD Method 1/1 Pass system"

    {Initial Guess}
    T_w_o=convertTemp(F,C,55); T_c_o=convertTemp(F,C,61.6)

    Repeat
        {A. Fluid Properties}
        {Warm Fluid}
        "T_bar_w=59.8[F]"
        T_bar_w=average(T_w_i, T_w_o)
        rho_w=Density(Water,T=T_bar_w,x=0);     C_p_w=Cp(Water,T=T_bar_w,x=0)

        {Cold Fluid}
        T_bar_c=average(T_c_i, T_c_o)
        rho_c=Density(Water,T=T_bar_c,x=0);     C_p_c=Cp(Water,T=T_bar_c,x=0)

        {H. Capacitance Rates}
        C_dot_w=m_dot_w*c_p_w
        C_dot_c=m_dot_c*c_p_c
        C_dot_min=min(C_dot_w,C_dot_c)
        C_dot_max=max(C_dot_w,C_dot_c)

        R=C_dot_c/C_dot_w

        {B. Plate Dimensions and Properties}
        A_o=16.076[ft^2]*convert(ft^2,m^2)
        U_o=367[btu/hr-ft^2-F]*convert(btu/hr-ft^2-F,kW/m^2-K)

        {I. NTU and Correction Factors}
        NTU=(U_o*A_o*N_s)/C_dot_min
        F=(1-0.0166*NTU)

        {J. Outlet Temperatures}
        E_counter=EXP(((U_o*A_o*N_s*F)*(R-1))/C_dot_c)

        T_w_o_new=(T_w_i*(R-1)-R*T_c_i*(1-E_counter))/(R*E_counter-1)
        T_c_o=(T_w_i-T_w_o_new)/R+T_c_i
        T_error=ABS(T_w_o_new-T_w_o)
        T_w_o=(T_w_o_new-T_w_o)/2+T_w_o
        q_w=C_dot_w*(T_w_i-T_w_o_new)
        q_c=C_dot_c*(T_c_o-T_c_i)
        q_error=q_c-q_w
        q=q_w
    UNTIL T_error<1E-5[C]

END
"! -------------------------END PLATE AND FRAME HEAT EXCHANGER FUNCTIONS------------------
--------"
"! -------------------------BEGIN GENERAL FUNCTIONS-------------------------"
Function RND(x,d)
    {x:=value to round, d:=number of digits; EES round feature only rounds to integer values}
    RND:=ROUND(x*10^d)/(10^d)
End
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Function ETA2(Xdes,Xin)
	{Checks to see if Exergy in is not zero to prevent division by zero}
	IF (RND(Xin,3)=<0) Then
		ETA2:=1
	ELSE
		ETA2:=1-Xdes/Xin
	ENDIF
End
Function ETA1(Xout,Xin)
	{Checks to see if Exergy in is not zero to prevent division by zero}
	IF (RND(Xin,3)=<0) Then
		ETA1:=1
	ELSE
		ETA1:=Xout/Xin
	ENDIF
End
"! --------------------------END  GENERAL FUNCTIONS--------------------------"

"! --------------------------BEGIN  ECONOMIZER FUNCTIONS--------------------------"
FUNCTION HXTCV(T_CW,T_CHW)
{Bypass valving for Heat Exchanger. 1.5[C] is the minimum approach for the HX}
	IF((T_CW+1[C])<T_CHW) THEN
		HXTCV:=0
	ELSE
		HXTCV:=1
	ENDIF
End
FUNCTION CHTCV(T_CHW,T_chws_setpoint)
{Bypass valving for Chiller. 1.0[C] is assumed for a deadband range}
	IF((T_CHW+1[C])<T_chws_setpoint) THEN
		CHTCV:=1
	ELSE
		CHTCV:=0
	ENDIF
End
"! --------------------------END  ECONOMIZER FUNCTIONS--------------------------"

"! --------------------------BEGIN PROCEEDURE-------------------------"

	"Call Modules and Subprograms before main programs"
	CALL
MAH(T_db_OSA,T_wb_OSA,P_OSA,T_db_setpoint,T_db_setpoint_low,T_db_setpoint_high,
RH_setpoint,RH_setpoint_low,RH_setpoint_high,MakeUpAir_Min,MAH_BkW_Design,MAH_V_do
t_design,MAH_ME_design: MAH_T_db[1..7],MAH_omega[1..7],MAH_Q_preheat,
MAH_Q_humidify,
MAH_Q_cooling_coil,MAH_Q_fan,MAH_FAN_BkW,MAH_X_preheat,MAH_eta_heating,MAH_X_
humidify,MAH_eta_humidify,MAH_X_Cooling,MAH_eta_cooling,MAH_X_dest,MAH_X_in,MAH_
X_out,MAH_eta_II,MAH_X_dot_air[1..2],MAH_X_dot_hw[1..2],
MAH_X_dot_huw[1],MAH_X_dot_chw[1..2],MAH_X_dot_cd[1])

	CALL CRAH(T_db_OSA, T_wb_OSA, P_OSA,Airflow_RA, MAH_omega[7], T_RA, T_SA,
CRAH_T_chws, P_RA, P_chw, DELTAT_CHW, CRAH_Qty, CRAH_AirFLow, CRAH_BkW:
CRAH_m_dot_chwr,
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CRAH_T_chwr,CRAH_Total_BkW,CRAH_X_dest,CRAH_X_in,CRAH_X_out,CRAH_eta_II,CRA
H_X_dot_air[1..2],CRAH_X_dot_chw[1..2])

	CALL CHILLER(T_chws_setpoint,T_db_OSA, T_wb_OSA, P_OSA,T_evap_i,T_cond_i,
m_dot_chw, DELTAS_T,Q_leak_eqv,R, m_dot_cw,Chiller_KW_Demand_Trigger:
T_cond_e,T_evap_e,
Power_chiller,CH_X_dest,CH_X_in,CH_X_out,CH_eta_II,CH_X_dot_chw[1..2],CH_X_dot_cw[1..
2],CH_Q_evap_demand)

	CALL
COOLINGTOWER(L,T_db_i,T_wb_i,P_OSA,Ka,Cell_Area,Fill_Depth,Cell_Qty,Cell_BkW,T_w_i,
T_cw_setpoint,CT_Approach_Min:G,T_w_e,CT_X_dot_des,CT_X_dot_w_i,CT_X_dot_w_e,CT_X
_dot_air_i,CT_X_dot_air_e,X_dot_IW,CT_Fan_Total_BkW,IW_makeup)

	CALL PlateFrameHX(N_s,T_db_OSA,T_w_i_HX,
T_c_i_HX,m_dot_w_HX,m_dot_c_HX:T_w_o_HX,T_c_o_HX,HX_Q,HX_X_dot_dest,HX_X_dot_i
n,HX_X_dot_out,HX_eta_II)

	"! Counterflow Heat Exchanger"
	m_dot_w_HX=m_dot_chw[22];		m_dot_c_HX=m_dot_cw[22]
	T_w_i_HX=T_chw[22];			T_c_i_HX=T_cw[22]
	T_w_o_HX=T_chw[23];		T_c_o_HX=T_cw[23]
	N_s=500;

	"! Primary  Input Parameters"
	"Lookup Bin Data for Weather- Link to Param table"
	"BinHour=149""4405"
	BinHour=4400

	CALL WEATHERBINDATA(City$,BinHour, P_OSA: T_db_OSA, T_wb_OSA,RH_OSA,
BinMonth$, BinDay)
	Altitude=GETALTITUDEDATA(City$)*convert(ft,m)
	P_OSA=BAROMETRICPRESSURE(Altitude*convert(m,ft))*convert(psia,kPa)
	w_OSA=HumRat(AirH2O,T=T_db_OSA,B=T_wb_OSA,P=P_OSA)
	"! Economizer Settings"
	T_cw_setpoint=CONVERTTEMP(F,C,52.5[F])"SETCWTEMP(T_WB_OSA,T_wb_FFC,T_wb
_Approach)"

	"! Data Center Conditioned Space Parameters"
	{"Altitude=15[ft];"
	Area=1000 [m^2] "including CRU/RAH spaces"
	Pressurization=9[(m^3/hr)/m^2];}
	MakeUpAir_Min=(Pressurization*Area)*Convert(m^3/hr,m^3/s)
	"Lighting=12.92 [W/m^2];"			Q_lighting=Lighting*Area

	"! Data Center Setpoints"
	{T_db_setpoint=22.5[C];		T_db_setpoint_low=18[C];
	T_db_setpoint_high=25[C]
	RH_setpoint=0.45[-];		RH_setpoint_low=0.40[-];		RH_setpoint_high=0.55[-]
	T_DP_setpoint_high=15[C]; T_DP_setpoint_low=5.5[C]}

	"! RAH Setpoints"
	"T_SA=20[C]"
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"CRAH_Qty=10[-]"
"CRAH_AirFLow=14158[L/s]"
"CRAH_BkW=17.4[kW]"
"CRAH_TSP=771.4[Pa]"
CRAH_T_chws=T_chw[6]
CRAH_T_chwr=T_chw[7]
CRAH_m_dot_chwr=m_dot_chw[7]

"! MAH Setpoints"
"MAH_Qty=1[-]"
"MAH_AirFLow=4000[cfm]*convert(cfm,L/s)"
MAH_BkW_Design=7.5[hp]*convert(hp,kW) "Max BHP for Fan"
MAH_V_dot_design=4000[cfm]*convert(cfm,m^3/s) "Max CFM for Fan"
MAH_ME_design=0.93[-]"Fan Motor Efficiency, Assumed VFD"
"MAH_ESP=1[inH2O]*convert(inH2O,kPa)"
"MAH_Prefilter_dP=0.150[kPa]"
"MAH_Postfilter_dP=0.250[kPa]"
"MAH_HEPAfilter_dP=0.350[kPa]"
"MAH_SAT=18.3[C]"

"! Chiller Setpoints"
"Chiller_Flow_Min=25[L/s]"
"DELTAS_T=0.9449[kW/K]"
"R=0.0046[kW/K]"
"Q_leak_eqv=-1646[kW]"
T_evap_i=convertTemp(C,K,T_chw[20])
T_cond_i=convertTemp(C,K,T_cw[20])
T_cw[1]=convertTemp(K,C,T_cond_e)
T_chw[1]=convertTemp(K,C,T_evap_e)
m_dot_cw=m_dot_cw[20]
m_dot_chw=m_dot_chw[20]
Chiller_KW_Demand_Trigger=4501[kW]

"! Chilled Water Setpoints"
"T_chw[1]=T_chws_setpoint;      "      P_chw=517[kPa]
CH_PMP_DesignFlow=1920[gal/min]*convert(gal/min,m^3/s)
CH_PMP_ActualFlow=m_dot_chw[10]/CH_PMP_density
CH_PMP_density=Density(Water,T=T_chw[10],x=0)
CH_PMP_DesignDensity=Density(Water,T=convertTemp(F,C,55[F]),x=0)
CH_PMP_DesignHead=100[ft]*convert(ft,m)
CH_PMP_Eff=0.84[-]
CH_PMP_DesignBkW=(CH_PMP_DesignFlow*CH_PMP_DesignDensity*g#*CH_PMP_Desi
gnHead)/(CH_PMP_Eff)*convert(W,kW)
CH_PMP_BkW=CH_PMP_DesignBkW*(CH_PMP_ActualFlow/CH_PMP_DesignFlow)^3
CH_PMP_FkW=CH_PMP_Eff*CH_PMP_BkW
cp_chw[10]=Cp(Water,T=T_chw[10],x=0)
DELTAT_CH_PMP_FkW=(CH_PMP_BkW-CH_PMP_FkW)/(m_dot_cw[10]*cp_cw[10])


"!Condenser Water Pump Sizing"
CW_PMP_DesignFlow=3360[gal/min]*convert(gal/min,m^3/s)
CW_PMP_density=Density(Water,T=convertTemp(F,C,80[F]),x=0)
CW_PMP_DesignHead=100[ft]*convert(ft,m)
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CW_PMP_Eff=0.84[-]
CW_PMP_DesignBkW=(CW_PMP_DesignFlow*CW_PMP_density*g#*CW_PMP_DesignHead)/(CW_PMP_Eff)*convert(W,kW)
CW_PMP_BkW=CW_PMP_DesignBkW*(CW_PMP_DesignFlow/CW_PMP_DesignFlow)^3
CW_PMP_FkW=CH_PMP_Eff*CH_PMP_BkW
cp_cw[10]=Cp(Water,T=T_cw[10],x=0)
DELTAT_CW_PMP_FkW=(CW_PMP_BkW-CW_PMP_FkW)/(m_dot_cw[10]*cp_cw[10])

"!Cooling Tower Setpoint"
CT_Approach_Min=2.8[C]
CT_Approach=T_w_e-T_wb_i
CT_Range=T_w_i-T_w_e
T_db_i=T_db_OSA "Inlet dry-bulb temperature"
T_wb_i=T_wb_OSA      "Inlet wet-bulb temperature"
T_w_i=T_cw[6]      "Inlet water temperature"
T_w_e=T_cw[7]"T_cw_setpoint" "Exit water temperature"
LG=L/G

CT_X_dot_in=CT_X_dot_air_i+CT_X_dot_w_i+CT_Fan_Total_BkW+X_dot_IW
CT_X_dot_out=CT_X_dot_air_e+CT_X_dot_w_e
CT_eta_II=ETA1(CT_X_dot_out,CT_X_dot_in)
G_ACM=G/Density(AirH2O,T=T_db_i,B=T_wb_i,P=P_OSA)
G_CFM=G_ACM*convert(m^3/s,cfm)
V=Cell_Area*Fill_Depth*Cell_Qty
KaVL=Ka*V/L
m_dot_cw[20]=L "L is set in the Cooling Tower Dialog"
Ka=0.3528 "Cooling Tower Characteristic"
Fill_Depth=7.01[m]"2.4224"
Cell_Area=49.1[m^2]
Cell_Qty=2[-]
Cell_bkW=23[kW]
Cell_MaxACM=2.5[m/s]*Cell_Area;

"! Data Center Airflow Parameters"
"Airflow_bypass=0.30[-]" "Percentage of Air that is bypassed for every CFM of server air"

"! Calculate Heat Load to be Removed"
NRows=4 "Specify number or rows to use in calculating DC Load. Table must be complete"
CALL DATACENTERLOAD(NRows, T_db_setpoint,P_OSA, RH_setpoint: Airflow_Server, CoolingLoad_DC,Rack)

"! Performance Metrics"
WPSM=CoolingLoad_DC*convert(kW,W)/Area
WPSF=CoolingLoad_DC*convert(kW,W)/(Area*convert(m^2,ft^2))
Total_Airflow_bypass=Airflow_Server*convert(m^3/s,L/s)*Airflow_bypass
kWPerRack=CoolingLoad_DC/Rack

"! Air Side Flow"
P_RA=P_OSA
Airflow_RA=Airflow_Server+Airflow_bypass
DELTAT_coil=CoolingLoad_DC/(Airflow_RA*Cp(AirH2O,T=T_db_setpoint,w=MAH_omega[7],P=P_RA)*Density(AirH2O,T=T_db_setpoint,w=MAH_omega[7],P=P_RA))
T_RA=convertTemp(K,C,convertTemp(C,K,T_db_setpoint)+DELTAT_coil)
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T_SA=T_db_setpoint+T_SA_offset
AirFlowPerRAH_Nplus1=Airflow_RA*convert(m^3/s,L/s)/CRAH_Qty
AirFlowPerRAH_N=Airflow_RA*convert(m^3/s,L/s)/(CRAH_Qty-1)

"! Exhaust Leakage Assumed to be same as Leaving Exergy from MAH"
EXH_X_dot=MAH_X_dot_air[2]

"! Chilled Water Mass Balance"
m_dot_chw[1]=m_dot_chw[2]"CHILLER EVAP OUTLET"

"Chiller ByPass Valving"
CHW_CV[3]=CHTCV(T_chw[20],T_chws_setpoint) "% OPEN For CH bypass flow"
m_dot_chw[3]=m_dot_chw[2]+m_dot_chw[28]; m_dot_chw[28]=CHW_CV[3]*m_dot_chw[18]
    m_dot_chw[28]=m_dot_chw[27]; m_dot_chw[27]=m_dot_chw[26];
m_dot_chw[26]=m_dot_chw[25]

m_dot_chw[4]=m_dot_chw[3]

"Minimum Flow ByPass Valving"
CHW_CV[1]=0[-] "% OPEN For future minimum bypass flow for variable primary pumps"
m_dot_chw[5]=m_dot_chw[4]-m_dot_chw[29];  m_dot_chw[29]=CHW_CV[1]*m_dot_chw[4];
    m_dot_chw[30]=m_dot_chw[29]; m_dot_chw[31]=m_dot_chw[30];
m_dot_chw[32]=m_dot_chw[31]

m_dot_chw[6]=m_dot_chw[5]
"Demand Set by CRAH"
"m_dot_chw[7]=100;" "m_dot_chw[7]=m_dot_chw[6]""!TEMP"

m_dot_chw[8]=m_dot_chw[7]
m_dot_chw[9]=m_dot_chw[8]+m_dot_chw[32]
m_dot_chw[10]=m_dot_chw[9];   m_dot_chw[11]=m_dot_chw[10];
m_dot_chw[12]=m_dot_chw[11]

"Heat Exchanger Valving"
CHW_CV[2]=HXTCV(T_cw[22],T_chw[22]) "% OPEN For HX bypass flow"
m_dot_chw[12]=m_dot_chw[21]+m_dot_chw[13];
m_dot_chw[13]=m_dot_chw[12]*CHW_CV[2]
m_dot_chw[22]=m_dot_chw[21]
m_dot_chw[23]=m_dot_chw[22]"! HX PLACE HOLDER"
m_dot_chw[24]=m_dot_chw[23]
m_dot_chw[14]=m_dot_chw[13]; m_dot_chw[15]=m_dot_chw[14];
m_dot_chw[16]=m_dot_chw[15]
m_dot_chw[17]=m_dot_chw[16]+m_dot_chw[24]
m_dot_chw[18]=m_dot_chw[17]
m_dot_chw[18]=m_dot_chw[19]+m_dot_chw[25]
m_dot_chw[20]=m_dot_chw[19] "CHILLER EVAP INLET"
m_dot_chw[20]=m_dot_chw[1]

"! Condensor Water Mass Balance"
CW_CV[3]=CHTCV(T_chw[20],T_chws_setpoint) "% OPEN For CHILLER bypass flow"
m_dot_cw[1]=m_dot_cw[2]"CHILLER CONDENSER OUTLET"
m_dot_cw[3]=m_dot_cw[2]+m_dot_cw[28];m_dot_cw[25]=CW_CV[3]*m_dot_cw[18]
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m_dot_cw[28]=m_dot_cw[27];m_dot_cw[27]=m_dot_cw[26];m_dot_cw[26]=m_dot_cw[25]

m_dot_cw[4]=m_dot_cw[3]

"CT Bypass Leg"
CW_CV[1]=0[-]     "% OPEN For CT bypass flow during winter conditions"
m_dot_cw[4]=m_dot_cw[5]+m_dot_cw[29];m_dot_cw[29]=CW_CV[1]*m_dot_cw[4]

m_dot_cw[32]=m_dot_cw[31];m_dot_cw[31]=m_dot_cw[30];m_dot_cw[30]=m_dot_cw[29]

m_dot_cw[6]=m_dot_cw[5];
m_dot_cw[7]=m_dot_cw[6]"CT"
m_dot_cw[8]=m_dot_cw[7];
m_dot_cw[9]=m_dot_cw[8]+m_dot_cw[32];
m_dot_cw[10]=m_dot_cw[9];      m_dot_cw[11]=m_dot_cw[10];m_dot_cw[12]=m_dot_cw[11]

"HX "
CW_CV[2]=HXTCV(T_cw[22],T_chw[22]) "% OPEN For HX bypass flow"
m_dot_cw[12]=m_dot_cw[13]+m_dot_cw[21];m_dot_cw[13]=m_dot_cw[12]*CW_CV[2];

m_dot_cw[22]=m_dot_cw[21];m_dot_cw[23]=m_dot_cw[22];m_dot_cw[24]=m_dot_cw[23]
"HX Bypass Leg"
m_dot_cw[14]=m_dot_cw[13];m_dot_cw[15]=m_dot_cw[14];m_dot_cw[16]=m_dot_cw[15]
m_dot_cw[17]=m_dot_cw[16]+m_dot_cw[24]; "HX Return"
m_dot_cw[18]=m_dot_cw[17]
m_dot_cw[18]=m_dot_cw[19]+m_dot_cw[25]; "Chiller Bypass"
m_dot_cw[20]=m_dot_cw[19]      "Chiller Condensor Inlet"
"m_dot_cw[20]=215";"m_dot_cw[20]=m_dot_cw[1]""!TEMP"

"!Energy Balance"
"Assume No Heat tranfer between pipe and environment"

"! Chilled Water Side"
"T_chw[7]=20.8[C]""!TEMP"
T_chw[2]=T_chw[1]; "CHILLER LWT"

m_dot_chw[3]*Enthalpy(Water,T=T_chw[3],x=0)=m_dot_chw[2]*Enthalpy(Water,T=T_chw[2]
,x=0)+m_dot_chw[28]*Enthalpy(Water,T=T_chw[28],x=0)
    T_chw[28]=T_chw[27]; T_chw[27]=T_chw[26]; T_chw[26]=T_chw[25];
T_chw[25]=T_chw[18];
T_chw[4]=T_chw[3];T_chw[5]=T_chw[4];T_chw[6]=T_chw[5];
    T_chw[29]=T_chw[4];
T_chw[30]=T_chw[29];T_chw[31]=T_chw[30];T_chw[32]=T_chw[31]
"RAH"
T_chw[8]=T_chw[7]"!T_chw[7] set by RAH Module"
"Bypass Loop"
m_dot_chw[9]*Enthalpy(Water,T=T_chw[9],x=0)=m_dot_chw[8]*Enthalpy(Water,T=T_chw[8]
,x=0)+m_dot_chw[32]*Enthalpy(Water,T=T_chw[32],x=0)
T_chw[10]=T_chw[9]
T_chw[11]=T_chw[10]+DELTAT_CH_PMP_FkW"!TEMP - INCLUDE PUMP RISE"
T_chw[12]=T_chw[11];T_chw[21]=T_chw[12];T_chw[22]=T_chw[21];
{T_chw[23]=T_chw[22]}"!ENTER HX HERE"
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	T_chw[24]=T_chw[23];

	T_chw[13]=T_chw[12];T_chw[14]=T_chw[13];T_chw[15]=T_chw[14];T_chw[16]=T_chw[15];
	m_dot_chw[17]*Enthalpy(Water,T=T_chw[17],x=0)=m_dot_chw[16]*Enthalpy(Water,T=T_ch
w[16],x=0)+m_dot_chw[24]*Enthalpy(Water,T=T_chw[24],x=0)
	T_chw[18]=T_chw[17];T_chw[19]=T_chw[18];T_chw[20]=T_chw[19]

	"! Condenser Water Side"
	T_cw[2]=T_cw[1]
	m_dot_cw[3]*Enthalpy(Water,T=T_cw[3],x=0)=m_dot_cw[2]*Enthalpy(Water,T=T_cw[2],x=0)
+m_dot_cw[28]*Enthalpy(Water,T=T_cw[28],x=0)
		T_cw[28]=T_cw[27];	T_cw[27]=T_cw[26];	T_cw[26]=T_cw[25];
	T_cw[25]=T_cw[18];
	T_cw[4]=T_cw[3];  T_cw[5]=T_cw[4];T_cw[6]=T_cw[5]
		T_cw[29]=T_cw[4];T_cw[30]=T_cw[29];T_cw[31]=T_cw[30];T_cw[32]=T_cw[31];
	"CT"
	T_cw[8]=T_cw[7]
	m_dot_cw[9]*Enthalpy(Water,T=T_cw[9],x=0)=m_dot_cw[8]*Enthalpy(Water,T=T_cw[8],x=0)
+m_dot_cw[32]*Enthalpy(Water,T=T_cw[32],x=0)
	T_cw[10]=T_cw[9]
	T_cw[11]=T_cw[10]+DELTAT_CW_PMP_FkW"!TEMP - INCLUDE PUMP RISE"
	T_cw[12]=T_cw[11]
	T_cw[13]=T_cw[12];	T_cw[14]=T_cw[13];	T_cw[15]=T_cw[14]; T_cw[16]=T_cw[15];
	T_cw[21]=T_cw[12];	T_cw[22]=T_cw[21]
	{T_cw[23]=T_cw[22]}	"!ENTER HX HERE"
	T_cw[24]=T_cw[23]
	m_dot_cw[17]*Enthalpy(Water,T=T_cw[17],x=0)=m_dot_cw[16]*Enthalpy(Water,T=T_cw[16]
,x=0)+m_dot_cw[24]*Enthalpy(Water,T=T_cw[24],x=0)
	T_cw[18]=T_cw[17];T_cw[19]=T_cw[18];
	T_cw[20]=T_cw[19]
	"T_cw[7]=30;""T_cw[1]=35""!TEMP"

	"! Exergy Calculations"
	Input=CH_PMP_BkW+CW_PMP_BkW+Power_chiller+CT_Fan_Total_BkW+CRAH_Total_B
kW+CT_X_dot_air_i+X_dot_IW+CRAH_X_dot_air[1]+MAH_X_dot_air[1]+MAH_X_dot_hw[1]+MA
H_X_dot_huw[1]+MAH_X_dot_chw[1]
	Output=CT_X_dot_air_e+EXH_X_dot+MAH_X_dot_hw[2]+MAH_X_dot_chw[2]+MAH_X_dot
_cd[1]+CRAH_X_dot_air[2]
	eta_II=ETA1(Output,Input)
	eta_II_CH_Plant=ETA1(CRAH_X_dot_chw[1]+CT_X_dot_air_e,CH_PMP_BkW+CW_PMP_
BkW+Power_chiller+CT_Fan_Total_BkW+X_dot_IW+CRAH_X_dot_chw[2]+CT_X_dot_air_i)
{
"!SAVE RESULTS TO A TABLE PER SPECIFIED PATH"
"City$='SFO'"
"FileLocation$='X:\SJSU\ME299\Models\'"
}
FileType$='.LKT '
FileNameString$=concat$(City$,FileType$)

FileString$=concat$(FileLocation$,FileNameString$)
$SaveTable 'Parametric:AllHours'  FileString$ /F
```