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UNIVERSITY OF CALIFORNIA, SAN DIEGO

Effects of Ambient Humidity on the Energy Use of Air Conditioning Equipment

A Thesis submitted in partial satisfaction of the requirements
for the degree of Master of Science

in

Engineering Sciences (Mechanical Engineering)

by

Justin George White

Committee in charge:

Professor Jan Kleissl, Chair
Professor Joe Goddard
Professor Rajesh Gupta

2010

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The Thesis of Justin George White is approved and it is acceptable in quality and form for publication on microfilm and electronically:

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2010

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LIST OF ABBREVIATIONS

ASHRAE American Society of Heating, Refrigeration, and Air-Conditioning Engineers.

CZ07 identifier for the California Energy Commission's weather file for the San Diego region which they call climate zone 7.

DDC direct digital control.

EBU identifier for Professor Jan Kleissl's weather station at EBU-2, UCSD, La Jolla, CA 92093.

HVAC heating, ventilation, and air conditioning.

KSAN identifier for the weather station at the San Diego International Airport, Lindbergh Field.

NOAA National Oceanic and Atmospheric Administration.

OTT Otterson Hall, Rady School of Management, UCSD, La Jolla, CA 92093.

SAT supply air temperature, or the temperature exiting an air handler.

UCSD University of California, San Diego.

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ABSTRACT OF THE THESIS

Effects of Ambient Humidity on the Energy Use of Air Conditioning Equipment

by

Justin George White

Master of Science in Engineering Sciences (Mechanical Engineering)

University of California, San Diego, 2010

Professor Jan Kleissl, Chair

This paper addresses the real-time use of ambient wet bulb temperature measurements in the optimization of building air conditioning system control as a means to save energy. This is not currently a common industry practice due to the maintenance requirements and perceived unreliability of humidity sensors. However, research shows that ambient wet bulb temperature is of central importance in determining optimal supervisory control of these systems. Since the UCSD campus operates its own weather stations, a case study was performed here as a first step in the implementation of their use in campus building control.

Introduction

The mechanical systems of most modern institutional and commercial buildings are controlled electronically. These buildings have a network of sensors and controllers which relay information back and forth between equipment and a central device connected to the internet. The sensors measure fluid properties such as temperature and pressure and the central device can then run programmed algorithms and output appropriate commands to the controllers. Since the central device is connected to the internet, a building operator can view all the measurements and change controller commands from any personal computer with a simple web browser. This so-called direct digital control (heretofore referred to as DDC) system therefore allows significant automation and quick operator troubleshooting, especially compared to the pneumatic control systems of the past. However, these systems are complex and are not always utilized to their full potential. The algorithms must be chosen appropriately and must be fine tuned on site in order to function reliably. And only then will a building operator leave them in place rather than manually overriding them.

In particular, there is a 'supervisory' algorithm, or customizable control sequence, that is recommended by the American Society of Heating, Refrigeration, and Air-Conditioning Engineers, heretofore referred to as ASHRAE. It is based on the mathematics of optimization and, in the simplified model, requires inputs of only building cooling/heating load and typical local ambient air, or outside air, wet bulb temperature. By way of a case study of a building on the University of California - San Diego campus, heretofore referred to as UCSD, this thesis focuses on the aforementioned wet bulb temperature dependence. This dependence may be strong on campus given the proximity to the coast. Furthermore, Professor Jan

Kleissl has multiple well-maintained weather stations on campus which log relative humidity, dry bulb temperature, and solar radiation measurements in 5-minute time steps and transmit this information in real-time to an internet-connected server. Given the fact that none of the buildings on campus measure humidity, which is a common-place industry practice, the campus has a unique opportunity.

First, background research and case study methods are described. The methods include interviews with campus facilities staff. This is then followed by case study results and conclusions. The conclusions section includes recommendations on how the campus can save energy by utilizing humidity measurements.

Chapter 1

Background

1.1 Schematic of a Building Cooling System

A schematic of a typical building cooling system is shown in Figure 1.1. This type of comfort cooling system is installed in the building used in the case study. The system can be divided into three subsystems: the chilled water loop, the condenser water loop, and the air distribution system.

The chiller uses electricity to run a refrigeration cycle. This refrigeration cycle cools the water in the chilled-water loop and rejects the heat to the condenser water loop. The tower then rejects the heat in the condenser water loop to the atmosphere.

The cold, or chilled, water in the chilled water loop is pumped through the cooling coil. This coil is a heat exchanger within an air handler, which is not shown on the diagram. It uses the chilled water to cool the recirculating air in the air distribution system. This cold air is then delivered to the building by the supply air fan which is also a component of the air handler. The building is labeled as 'zones', which are subdivided spaces within the building. Although not pictured, each zone has a terminal box which has a valve that modulates the air delivered to each zone. The economizer is not identified either but it has valves which modulate the amount of ventilation air, or outside air, being delivered to the building in order to maintain proper oxygen levels.

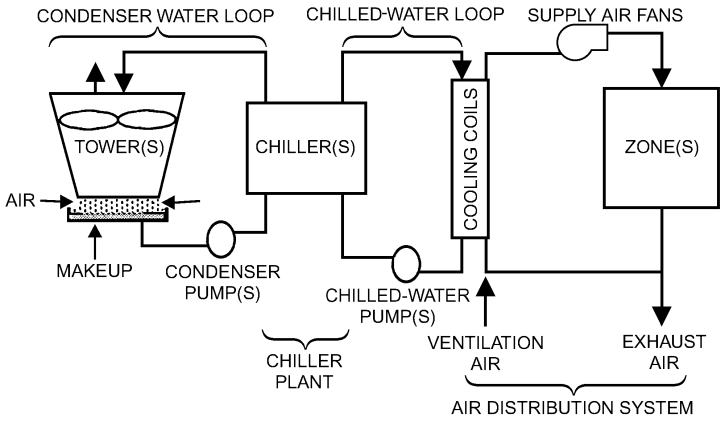

Figure 1.1: Schematic of a typical building cooling system. The building in the case study has this type of cooling system and it is typical for all buildings on campus. The heating system is different and is not shown here. *Source:* ASHRAE 07[1](41.1).

1.2 Optimization

Chapter 41 entitled "Supervisory Control Strategies and Optimization" in the 2007 ASHRAE Handbook[1] provides an overview of the electronic control of heating, ventilation, and air conditioning (heretofore referred to as HVAC) equipment. It includes control strategy recommendations based on rigorous research. The ASHRAE handbooks are intended to be definitive design guides for people in the HVAC industry, especially mechanical engineers, and updated editions are provided to ASHRAE members every year. Of particular interest is the recommended "simplified system-based optimization approach"[1](41.33) and its dependence on typical local ambient air wet bulb temperature. While an optimal and more mathematically sophisticated sequence is also provided, the simplified sequence provides a practical approach which takes into account the real life limitations in the field. These limitations are severe in that buildings rarely have any humidity sensors, not even to measure the local ambient air wet bulb temperature. The optimal model depends on cooling coil wet bulb temperature but this was simplified to typical local ambient wet bulb temperature as could be found in a database of historical measurements from nearby weather stations.

According to ASHRAE 07, control can be divided into two categories: "local-loop control" and "supervisory control"[1](41.1). HVAC control can then be described as a series of small 'local' control loops which usually receive one sensor input (e.g. air handler supply air temperature) and then output one command to an equipment controller (e.g. cooling coil chilled water valve) whose purpose is to maintain a set point on the same sensor as provided by the 'supervisory' system. Furthermore, these local loops are all interrelated as information travels from the most remote part of the HVAC system to the most central parts, and vise versa. Information travels from room thermostats to terminal boxes, to supply fans, to cooling coils, to chilled water loop pumps, to chillers, to condenser loop pumps, and finally to the tower.

The focus of ASHRAE 07 is on supervisory control, or the selection of set points, with the assumption that the local loops are functioning properly and that *all of the zones are satisfied*. This distinction is important to building operators because the ASHRAE optimization methods in themselves will therefore never cause occupant comfort complaints[1]. Another assumption is that the "individual zone latent-to-sensible load ratios and ratios of individual sensible zone loads to total sensible loads for all zones" are relatively constant[1](41.34). This means the amount of water vapor produced in the building must not vary significantly. Humans are typically the biggest source of this water vapor so their occupancy levels in each room should be steady.

Iterative numerical analysis, or "deterministic methods", underlie the simplified system-based optimization approach[1](41.31). This approach includes solving an equation which relates the air conditioning equipment to total energy use. For example, fan speed is proportionally related to fan electricity demand. All of these equations are then empirically solved for the coefficients using measurements of actual energy use and linear regression techniques[1](41.34-35). The equation is

$$J(\mathbf{f}, \mathbf{M}, \mathbf{u}) = \mathbf{u}^T\mathbf{A}\mathbf{u} + \mathbf{b}^T\mathbf{u} + \mathbf{f}^T\mathbf{C}\mathbf{f} + \mathbf{d}^T\mathbf{f} + \mathbf{f}^T\mathbf{E}\mathbf{u} + g \tag{1.1}$$

[1]Zone temperatures are typically monitored and cooling is increased if any of the zones get hotter than a user-defined limit.

where J is is the total power, $\mathbf{f}$ is a vector of uncontrolled variables, $\mathbf{M}$ is a vector of discrete control variables, $\mathbf{u}$ is a vector of free control variables, $\mathbf{A}$, $\mathbf{C}$, and $\mathbf{E}$ are coefficient matrices, $\mathbf{b}$ and $\mathbf{d}$ are coefficient vectors, and g is a scalar. The uncontrolled variables are typical local ambient wet bulb temperature and chilled water load. The free control variables are air handler cooling coil exiting supply air temperature, heretofore referred to as SAT, chilled water supply temperature, tower airflow relative to design capacity, condenser water flow relative to design capacity, and number of operating chillers[2]. An example of a discrete control variable is the economizer mode which is either 100% outside air or minimum design outside air. It does not show up in Equation 1.1 but results in multiple similar equations for all values of the vector $\mathbf{M}$.

To determine the optimum values of the free control variables, defined as $\mathbf{u}^*$, the Jacobian of Equation 1.1 with respect to $\mathbf{u}$ is then solved to first order[1](41.34) giving

$$\mathbf{u}^* = \mathbf{k} + \mathbf{K}\mathbf{f} \tag{1.2}$$

where $\mathbf{f}$ is the same vector of uncontrolled variables as in 1.1 and $\mathbf{k}$ and $\mathbf{K}$ are a vector and matrix of coefficients based on Eq. 1.1

$$\mathbf{k} = -\mathbf{A}^{-\mathbf{1}}\mathbf{b}/2 \tag{1.3}$$

$$\mathbf{K} = -\mathbf{A}^{-\mathbf{1}}\mathbf{E}/2 \tag{1.4}$$

In regards to the inputs, typical local ambient wet bulb temperature is determined by way of a historical database from nearby weather stations and chilled water load by way of a programmed automatic calculation of cooling coil energy demand[1](41.31) as shown in Eq. 1.7. The reason for using typical local ambient air wet bulb temperature is that it was found through empirical research to be approximately linearly dependent on the wet bulb temperature of the air directly entering the cooling coil. Per ASHRAE 07[1](41.18), "energy transfer across

[2]If used, energy storage should also be included in the algorithm but this is outside of the scope of this paper.

each cooling coil is proportional to the difference between its entering air wet-bulb temperature and the entering water temperature (the chilled-water set point)".

This is also shown by Cengel and Boles 98[2](740). The mass balance and energy balance for the air distribution side of the air handler, ignoring fan energy, and the energy balance for the chilled water side are

$$\dot{m}_w = \dot{m}_a(w_1 - w_2) \tag{1.5}$$

$$\dot{Q}_{out} = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w \tag{1.6}$$

$$\dot{Q}_{in} = \dot{m}_{chw} Sk(T_{wall} - T_{supply}) \tag{1.7}$$

where $\dot{m}_w$ and h_w are the condensate mass flow rate and saturated liquid enthalpy per unit mass, $\dot{m}_a$ is the air mass flow rate, w_1 and h_1 are are the entering air humidity ratio and dry air enthalpy per unit mass of dry air, w_2 and h_2 are are the exiting air humidity ratio and dry air enthalpy per unit mass of dry air, and $\dot{Q}_{out}$ is the heat rejection. For Eq. 1.7, $\dot{m}_{chw}$ is the mass flow rate of chilled water, S is the shape factor of the cooling coil pipes, k is the thermal conductivity of the pipe material, T_{supply} is the inner wall temperature which is approximated as the chilled water supply temperature, and T_{wall} is the temperature of the cooling coil outer wall which is approximated as the chilled water return temperature.

Enthalpy per unit mass of dry air at the location n is given by

$$h_n = C_p T_n + w_n h_{g,n} \tag{1.8}$$

and the humidity ratio entering the coil is given by

$$w_1 = (C_p(T_* - T_1) + w_* h_{fg*})/(h_{g1} - h_{f*}) \tag{1.9}$$

$$w_* = 0.662 P_{sat,*}/(P - P_{sat,*}) \tag{1.10}$$

where T_* is the adiabatic saturation temperature which can be approximated as the wet bulb temperature at the entering side of the cooling coil surface

since P is atmospheric pressure[2](731). Therefore, w_1 and h_1 are linearly dependent on this temperature. Also, T_{wall} is proportional to this temperature if condensation is occurring at the outer wall surface. The coil entering air is a mixture of return air and outside air. Assuming that the mass flow rates of these airstreams can be estimated, that the building latent heat load and return air temperature are fairly constant, and that dehumidification is occurring at the coil, the only remaining unknowns needed to calculate w_1, h_1, and T_{wall} are the local ambient dry bulb and wet bulb temperatures and the SAT set point. Furthermore, the dependence is again linear.

For instance, $\dot{Q}_{in}$ which equals $\dot{Q}_{out}$ could be calculated from Eq. 1.7 by measuring the mass flow rate of chilled water and the supply and return temperatures. This represents the total air handler cooling load. For a given load and constant chilled water supply temperature, if the SAT is increased then $\dot{m}_a$ and the fan power increase per Eq. 1.6. This is turn increases T_{wall} since it is proportional to the wet bulb temperature difference across the coil if condensation is occurring. Then, $\dot{m}_{chw}$ and the pump energy must decrease per Eq. 1.7. Since the room temperature must be satisfied, the SAT must be optimized by increasing the humidity ratio in the room to the maximum value allowed by the limits of human comfort as shown in Figure 1.2.

For a given sensible heat load and a higher local ambient wet bulb temperature, the entering air has a higher humidity ratio and the SAT should be increased to minimize condensation. For lower ambient wet bulb temperature, the entering air has lower humidity ratio so condensation is less of an issue and the heat exchanger is more effective. So, SAT set point should be decreased.

1.3 Economizers

Spitler 87[7] explores the use of economizers under the severe limitations of no room humidity sensing, no air handler supply air humidity sensing, and constant supply air temperature set point. Economizers are controlled to always maintain at least a specified minimum ratio of outside air to return air for health

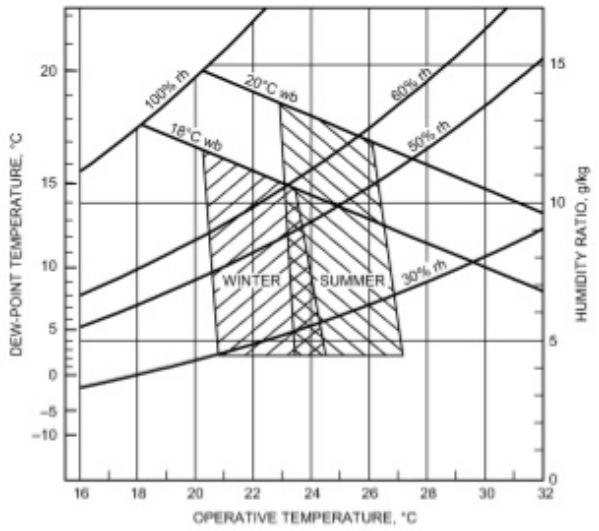

Figure 1.2: Thermal comfort zone, or the acceptable combinations of temperature and humidity within a building to insure that most occupants are satisfied with the climate. *Source:* ASHRAE 05[3](8.12).

reasons. However, the economizers can also be controlled to deliver 100% outside air if it is determined that this would save energy. With enthalpy-based control, this decision is made based on which airstream has the lower amount of enthalpy.

Spitler showed succinctly with a psychrometric chart how various set point and weather conditions affect this decision making process[7](22). This chart is shown in Figure 1.3. Vertical lines represent constant dry bulb temperature and horizontal lines represent constant mass of ambient water vapor. The diagonal straight lines represent constant enthalpy and the curved lines represent constant relative humidity. Return air enthalpy is assumed to be constant and is enumerated as 5. The SAT set point is also assumed to be constant and is shown as a vertical dashed line. Numbers 1, 2, 3, 4, and 6 mark areas of hypothetical outside air enthalpy.

One counterintuitive result occurred when the outside air lies in area 3. Here, the outside air is hot and dry - such as occurs during so-called Santa Ana wind events in southern California - and the enthalpy is lower than the room air enthalpy. Only sensible cooling occurs at the cooling coil for either possibility of 100% outside air or minimum outside air going through the coil. This is shown for

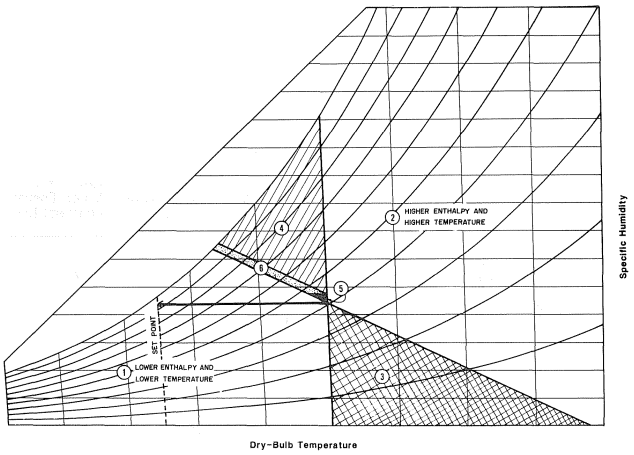

Figure 1.3: Psychrometric chart showing that enthalpy-based economizer control with constant supply air temperature set point is not always effective. In area 3, enthalpy-based controls would command the economizers to 100% outside air due to the lower enthalpy of the outside air than the return air. However, this results in more sensible cooling. *Source:* Spitler 87[7](22).

return air by the bold horizontal line of constant water vapor mass[3].

So, enthalpy-based economizers would command the air conditioning equipment to 100% outside air. However, this would result in more energy use than a minimum outside air command because the cooling coil would be doing more sensible cooling than required. Assuming that the room humidity would be acceptable for either economizer mode (i.e. room temperature and humidity would be within the comfort zone shown in Figure 1.2), it would have been better to use minimum outside air.

Spitler concludes that enthalpy-control under the constraint of constant supply air temperature set point is only worthwhile for certain climates. For locations where enthalpy control would not save significant energy, he recommends that humidity sensors not be installed. On the other hand, he states that "should reliable humidity sensors be developed at some time in the future" then they should be used in combination with supervisory algorithms to make "ideal" ventilation air ratio choices[7](17).

[3]A line is not shown for outside air but it would be longer and also horizontal.

Chapter 2

Methods

The above topics were explored through an analysis of recent San Diego weather measurements and a case study of a UCSD campus building called Otterson Hall, heretofore referred to as OTT. The data used in this case study includes and is limited to one year's worth of weather measurements from the National Oceanic and Atmospheric Administration's (heretofore referred to as NOAA) and Kleissl's closest weather stations, an energy model authored by the HVAC design engineer for OTT, and a year's worth of logged DDC sensor measurements for OTT. The time dependent data was from September 2008 to September 2009, based on data availability, and time interval steps were either 5-minute or 15-minute. This data was then analyzed in order to understand the current DDC operation and recommend energy saving measures. Experimental validation of the recommendations was not performed but is rather outlined for future study in the 'Discussion' section.

The occupancy-type of OTT is classroom and this building was chosen because of the availability of the HVAC engineer's energy model. The building has roughly 62,000 square feet of conditioned space. The roof insulation has an R-value of 25 and the wall insulation has an R-value of 11. The windows have a U-factor of 0.55 $BTU/hr\text{-}ft^2\text{-}^\circ F$ and a center of glass solar heat gain coefficient of 0.3. They have tinted dual-pane glass and the frame is non-metal. The main internal heat sources are lights, people, and personal computers. The cooling system consists of two air handlers and a few chilled water pumps. The chilled water is supplied to

the building from a remotely-located chiller at the central plant on campus. The building is equipped with a full modern DDC system which can be controlled, with proper access, from any web browser.

2.1 Interviews

Three employees of the campus facilities department were interviewed. The first interview was with John Dilliott, a senior manager of the facilities department. Within the aforementioned framework, the purpose of this encounter was to identify the area of focus of the case study. The manager explained that he was most interested in building-level controls - with all of it's sub-level architecture included. Furthermore, he suspected that energy-saving opportunities might exist during high ambient humidity conditions.

He claimed to have no interest in central plant control or equipment modifications due to implicit limitations. One important physical limitation was that the set point for chilled water supply temperature had to be held constant due to certain unmentioned critical building load demands. Also, he was aware of Kleissl's weather stations and hoped to take advantage of the real-time data.

The second interview as with Walter 'Richie' Richardson, a manager of facilities operations. He has direct communication with the control and troubleshooting staff of the central plant and all the campus buildings. He was firmly in support of continuous improvement of operations and cited that more automation would be desirable. However, it is imperative that this automation function properly. Otherwise, the troubleshooting staff has the authority to override it and would likely do so.

He went on to explain the current supervisory control sequence for the air handler SAT set point. It varies between 55 to 65 $^{\circ}F$ and an algorithm insures that the calculated average zone temperature per floor does not exceed the allowed dead band. However, it was not clear how the set point was determined if all zones were satisfied as would likely be the case during many part load conditions.

He concluded by authorizing access to the web-based building-level control

system and providing a suggestion for further study. This access included the ability to download all available logged data for a number of buildings, including OTT. His suggestion for further study was that a case study be performed on the 'Vaughn Hall' building since it has an on-site dedicated chiller whose chilled water supply temperature set point could be modified.

The last interview was with Tim Shultz, the manager of central plant operations. He stated that the chilled water supply temperature could not be changed due to the critical cooling load of a building housing supercomputers. The constant set point of 42 $^{\circ}F$ is maintained by the plant's chillers and 3,900 ton thermal energy storage system. The staff claims to have no difficulty satisfying the overall campus cooling demand. Their nine chillers range in size from 1,300 tons to 4,500 tons so they are also capable of running at part load. The plant has a DDC system as well but it is on a different web address and no access was granted for this study.

2.2 Weather

The NOAA weather station used is located at the San Diego International Airport (heretofore referred to as KSAN), also known as Lindbergh Field, which is about 10.5 miles away from the UCSD campus where OTT is located. The data was downloaded from the weather station's public website. Dry bulb temperature and relative humidity measurements[1] at 1-hour time steps from September 4, 2008 to September 4, 2009 were used. Kleissl's weather station (heretofore referred to as EBU) is located at the EBU-2 building on campus which is located less then 3/4 of a mile from OTT. Dry bulb, relative humidity, and global horizontal incident solar radiation measurements were used and wet bulb temperature was subsequently calculated using Stull 2000[6](101). The time step of the raw data is 1-minute.

These two sets of measurements were compared to prove that a so-called 'micro-climate' exists near OTT which has a radius much smaller than 10 miles. MATLAB was used for plotting and a few necessary calculations while a text editor and spreadsheet program were used for formatting imported data.

[1]Relative humidity is tabulated but it is calculated based on measured dry bulb and dew point temperatures.

A Department of Energy weather file converter program called DOEWth was used separately to produce properly formatted hourly weather files of the EBU data for use in energy modeling. This free program was obtained from the www.doe2.com website.

2.3 Energy Modeling

Energy models such as the one produced by the engineer of OTT are used primarily in the engineering field to produce state energy compliance documentation. Here they are used primarily to understand the dependence of building energy use on real - or statistically unsteady - weather. A proprietary software program called Energy Pro was used. It is a graphical user interface type program which calculates building energy use based on the a user's inputs and a hidden calculation algorithm. It outputs compliance forms which summarize the building's annual energy use. The program can also be used for non-compliance, performance only calculations using a Department of Energy algorithm. The program interface remains the same but it allows for hourly outputs of a few categorized energy uses within the building.

The weather file used in the engineer's model is the California Energy Commission's mandated weather file (heretofore referred to as CZ07) for the San Diego region. This weather file represents a typical year in the San Diego coastal region and does not attempt to capture extreme weather events in any given year. Two additional energy models were created which only differed from the engineer's model by the weather file that was used. These two weather files were created with the DOEWth program using EBU measurements instead of the CZ07 data. One weather file was based on EBU temperature and wet bulb measurements but with solar data from the CZ07 file. The other file was based on EBU temperature, wet bulb, and solar measurements.

The electricity demand outputs of all models were then compared. It would be preferable to also simulate the energy saving recommendations of this study. However, this type of analysis is cumbersome in the chosen proprietary software.

The choices in any given input field are limited, the calculation algorithms cannot be customized, and hourly outputs are only provided for a few parameters.

2.4 Trending

The DDC trends were obtained from two sources: the university's facilities department and the administrators of a public website that tracks the electricity use of numerous buildings on campus[2].

The facility's controls website logs all of the sensors within the OTT building at 15-minute time steps. However, only three days worth of data are stored permanently for most sensors. The saved data from the remaining sensors go back much further and at least until September 2008. The public website on the other hand has only overall building electricity demand measurements. The data obtained from the administrators has 15-minute time steps.

A year's worth of data was obtained from both sources. In regards to the facilities website, all available stored sensor data was obtained during the year-long period of choice. After reviewing plots of the EBU dry bulb and wet bulb temperatures, two three day periods were chosen to represent a low humidity and a high humidity period. These time ranges were then used to create subplots of a few important sensor trends along with the hourly energy use outputs calculated in the energy models.

The building sensors determined to be of importance where building electricity demand, chilled water supply and return temperature, and SAT. It would have been preferable to have chilled water flow measurements or functional chilled water load meters but this was not available. Also, only a few days of SAT set point commands were available. After a brief review of other buildings on the facilities website, these limitations appeared to be consistent across campus.

[2]The website is http://power.ucsd.edu and the data was provided in text format by Thomas Weng.

Chapter 3

Results

3.1 Weather

Figure 3.1 shows scatter subplots of KSAN versus EBU weather. The correlation coefficients for dry bulb temperature and relative humidity are 0.53 and 0.62. This indicates that while the measurements are similar there is a noticeable difference in weather due to location.

These data show that the weather stations are least correlated for maximum and minimum temperature and humidity. At the high end, KSAN values are lower than EBU values. At the low end, KSAN values are higher than EBU values. This appears to indicate the weather conditions at the airport are milder. In any case, accurately measuring maximum and minimum humidity is most important because energy savings based on humidity sensing is expected to be at a maximum or minimum during these periods. Therefore, the EBU measurements are significantly more useful.

The hourly solar radiation measurements from EBU were then compared to the hourly solar data from the CZ07 weather file. The correlation coefficient over the full year is 0.88 and the annual means are both roughly 210 W/m^2. This makes sense since the purpose of CZ07 file is to capture average seasonal weather not maximum or minimum weather events or highly variable cloud fractions as would be reflected in the EBU data.

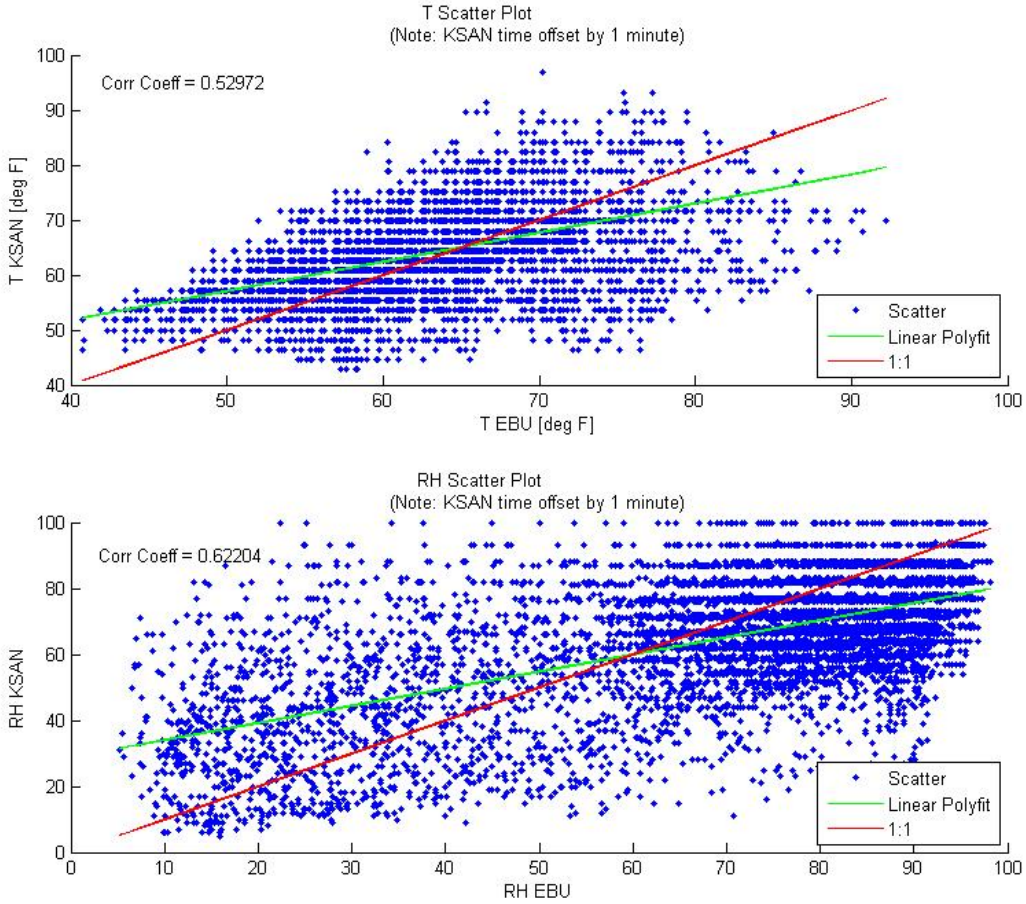

Figure 3.1: Comparison of dry bulb and relative humidity measurements from EBU and KSAN. All measurements shown are for the same one year range from Fall 2008 to Fall 2009. The linear regression equations used are $T_{ksan} = m_t T_{ebu} + b_t$ and $RH_{ksan} = m_r RH_{ebu} + b_r$, where m_t, b_t, m_r, and b_r are scalar constants which were calculated in MATLAB to fit the data.

3.2 Energy modeling

The bottom subplot of Figure 3.2 shows the calculated hourly building electricity demand for all three energy models for a three day period of high wet bulb temperature in early September. Actual building electricity demand measurements from the DDC electricity meter are also shown on this plot. A three day period of low wet bulb temperature in early January is then shown in Figure 3.3. The data from the electricity meter is labeled as $Meter$. The engineer's energy model which uses the CZ07 weather file is labeled as *CZ07*, the model based on EBU temperatures only is labeled *EBU*, and the model based on EBU temperatures and solar measurements is labeled EBU_{sol}. Both periods occur during the academic school year and are from 6AM on a Tuesday to 6AM on a Friday.

The high wet bulb period had a mean wet bulb of 60 $^{\circ}F$, a mean dry bulb of 64 $^{\circ}F$, and a mean solar radiation of 213 W/m^2 for the CZ07 weather. The EBU station measurements over the same period had a mean wet bulb of 58 $^{\circ}F$, a mean dry bulb of 62 $^{\circ}F$, and a mean solar radiation of 257 W/m^2.

The low wet bulb period had a mean wet bulb of 46 $^{\circ}F$, a mean dry bulb of 54 $^{\circ}F$, and a mean solar radiation of 130 W/m^2 for the CZ07 weather. The EBU station measurements over the same period had a mean wet bulb of 50 $^{\circ}F$, a mean dry bulb of 54 $^{\circ}F$, and a mean solar radiation of 137 W/m^2.

It is important to point out that the power measurements in the bottom subplots are for *building* electricity demand which includes fans and on-site chilled water pumps. They do not include power demands for chillers and additional pumps which in this case are located at the central plant and were not modeled.

The building electricity demand for the *CZ07* and *EBU* models are highly correlated for both time periods. The mean demand of each is about 130 kW for the high wet bulb period and 160 kW for the low wet bulb period. The EBU_{sol} model is much less correlated to these two, especially for the low wet bulb period. The mean of these data during the high wet bulb period is 170 kW and during the low wet bulb period is 450 kW.

The middle subplots in both figures compares the solar data used in the *CZ07* and *EBU* models to the EBU station solar measurements used in the EBU_{sol}

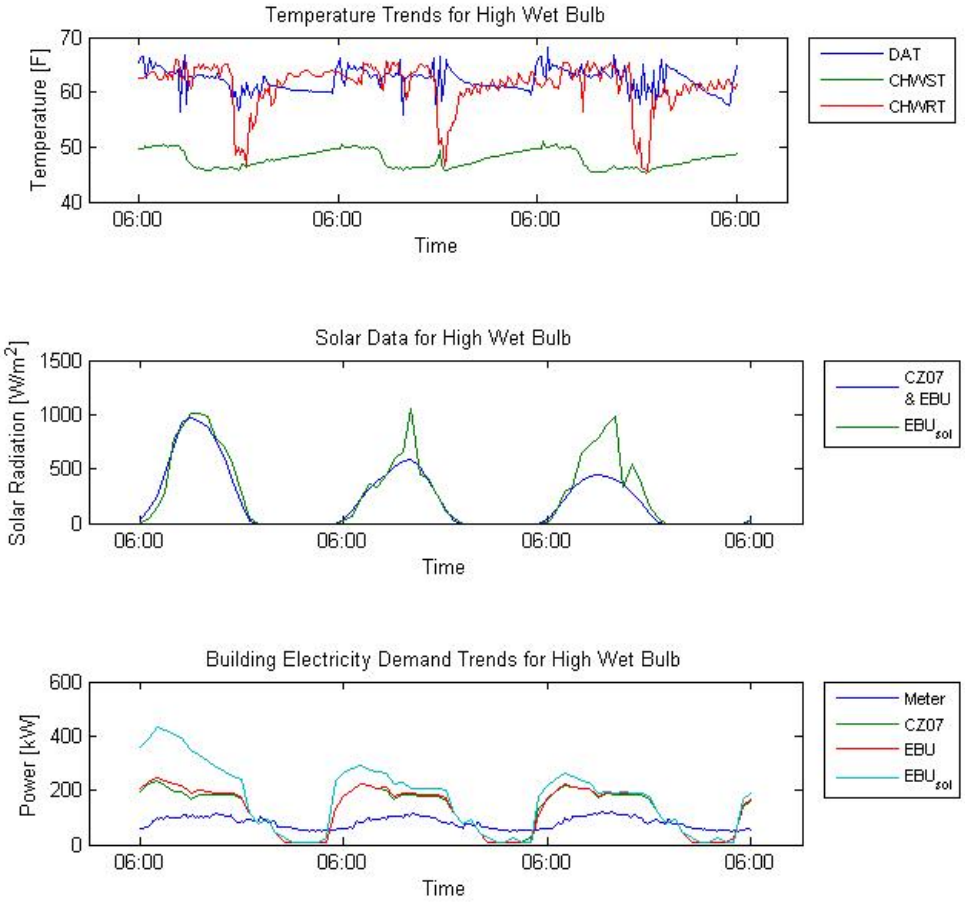

Figure 3.2: Top - Trends of discharge air, chilled water supply, and chilled water return temperatures for high ambient wet bulb period of 6AM Sep 01, 2009 to 6AM Sep 04, 2009. Middle - Comparison of the solar radiation used in the *CZ07* and *EBU* models to the solar measurements from the EBU weather station which were used in the EBU_{sol} model. Bottom - Building electricity demand outputs from the the building electricity meter (*Meter*), engineer's energy model using CZ07 weather (*CZ07*), EBU temperature-based model (*EBU*), and EBU temperature and solar-based model (EBU_{sol}) for the same time period.

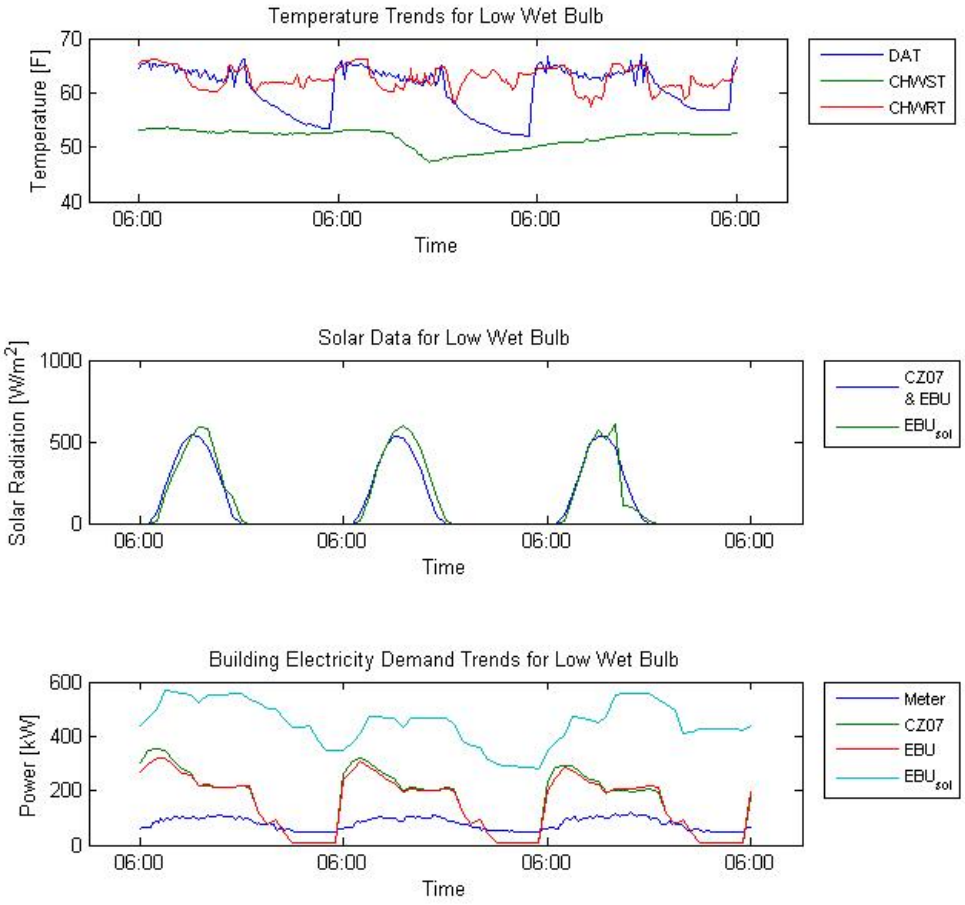

Figure 3.3: Top - Trends of discharge air, chilled water supply, and chilled water return temperatures for low ambient wet bulb period of 6AM Jan 06, 2009 to 6AM Jan 09, 2009. Middle - Comparison of the solar radiation used in the *CZ07* and *EBU* models to the solar measurements from the EBU weather station which were used in the EBU_{sol} model. Bottom - Building electricity demand outputs from the the building electricity meter (*Meter*), engineer's energy model using CZ07 weather (*CZ07*), EBU temperature-based model (*EBU*), and EBU temperature and solar-based model (EBU_{sol}) for the same time period.

model. The correlation coefficient for the high wet bulb period is 0.93 and the correlation coefficient for the low wet bulb period is 0.97. The lower correlation in the high wet bulb period is due to significant differences in maximum solar radiation in the last two days.

The high wet bulb period also has high dry bulb so we would expect the building energy use to be greater during this period. However, this is not the case for any of the models. This could be due to the fact that energy transfer across a cooling coil is proportional to the difference between the entering air wet bulb temperature and the entering chilled water temperature[1](41.18). The model includes an assumption that chilled water temperature is constant so the higher wet bulb temperature creates better cooling coil heat transfer efficiency. This results in less pumping power for a given air handler cooling load.

At night, the *CZ07* and *EBU* models indicate approximately no energy use. This is due to the fact that all the models include schedules which indicate that all equipment and lights are turned off at night.

However, the EBU_{sol} model indicates very high energy use at night in Figure 3.3. The middle subplot shows that the solar measurements used in this model are correct. So, it is likely that the weather file was not converted accurately and further study is needed.

3.3 Trending

The electricity meter trends are shown on the bottom subplots of Figures 3.2 and 3.3. The average demand during the high wet bulb period was 81 kW and the average demand during the low wet bulb period was 79 kW. For both periods, the building has a significant non-zero base load during the night and the maximum energy use during the day is much lower than the energy use calculated by the energy models.

The top subplots in Figures 3.2 and 3.3 show the temperature trends for the building. For the high wet bulb period, the mean chilled water supply temperature is 48 $^{\circ}F$, the mean return temperature is 61 $^{\circ}F$, and the mean discharge

air temperature is 62 $^{\circ}F$. For the low wet bulb period, the mean chilled water supply temperature is 52 $^{\circ}F$, the mean return temperature is 63 $^{\circ}F$, and the mean discharge air temperature is 61 $^{\circ}F$.

In regards to the base load, the assumption in the models that all equipment and lights are turned off at night is incorrect. Not much can be said beyond this because the meter only reads overall electricity demand. The lower peak daytime energy use indicated by the meter shows that peak loads were overestimated in the model.

The actual energy use is fairly constant between the two time periods whereas the calculations showed greater differences. This may indicate that weather has less of an effect on energy use than predicted. In could be that internal loads such as lights, people, and equipment are more important. However, it could also mean that the favorable wet bulb temperature is not being utilized.

The magnitudes of both average energy measurements are significantly lower than all the calculated averages from the models. There could be many reasons for this. For instance, the meter may not be monitoring all the equipment in the engineer's model due to its location. Another possibility is that the engineer's model simply overestimated the demand.

The chilled water temperature trends are of limited value given that chilled water flow rate trends were not logged during the periods in question. Therefore, building cooling load cannot be determined. While the building does have chilled water energy meters, the trends were not saved. However, it is worth noting that the chilled water supply temperatures are much higher than the 42 $^{\circ}F$ set point at the central plant. Room temperatures would indicate whether the cooling demand is satisfied but this information is not available for these time periods.

The discharge supply air temperature trends show that the set point is being varied between about 55 $^{\circ}F$ and 65 $^{\circ}F$. However, without set point and room air temperatures trends it is hard to determine, let alone evaluate, the set point algorithm.

Chapter 4

Discussion and Conclusion

The case study indicates that the UCSD facilities can likely save energy by implementing the use of Kleissl's humidity measurements in real-time as though they were located at nearby campus buildings. However, the exact amount of energy savings is not clear since no physical experiments were performed.

It is recommended that the simplified optimization sequence given by ASHRAE 07[1] and Braun 89a[4](657) be implemented on certain campus buildings including OTT. These buildings should have relatively stable return air sensible heat ratios and no tight zone humidity requirements. Many classroom and office buildings would likely meet this requirement. The rooms in such a building usually have consistently light plug loads with personal computers being the most energy intensive equipment. The occupancy load is usually the only source of latent heat and the occupancy levels are fairly constant during the day.

It would be ideal to measure humidity in all of the rooms and the return air to confirm with certainty the statements above. However, this data is not available for the buildings on campus. Therefore, a simple walk through of the building could be performed to assess the usage and load of each room.

In the case of buildings served by the central plant, only air handler SAT should be optimized. The optimization sequence should be initially tested on one building so that energy savings could be roughly determined and so that operators can vet the performance. If Kleissl's data became unavailable for some reason, ambient wet bulb data from the CZ07 file could be used.

Furthermore, a second building with an on-site chiller plant - such as Vaughn Hall - should also be tested. Here, optimization of chilled water set point could be tested in addition to SAT optimization. It is expected that energy savings would be greater here than in a building with only SAT optimization[1]. However, a building such as this would also require a walk through to determine room types.

Also, economizer experiments could be performed on any building with economizers and no 100% outside air requirement. Kleissl's data would be used along with SAT measurements to create an "ideal" economizer as described in Spitler 87[7].

On the modeling side, a more flexible design-based software could be used to simulate the energy savings of these recommendations. These recommendations are based on well established research. Therefore, the required algorithms are very likely already within the code of some energy software programs. For the most flexibility, custom code could be written in which one would have full control of the calculations, inputs, and outputs.

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