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For the degree of Master of Science in Mechanical Engineering

Is approved by the final examining committee:

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Approved by Major Professor(s): Jie Chen

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EVALUATION OF PERFORMANCE OF AN AIR HANDLING UNIT USING WIRELESS MONITORING SYSTEM AND MODELING

A Thesis

Submitted to the Faculty

of

Purdue University

by

Akram Ghassan Khatib

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of

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I would like to dedicate this thesis to my family that has been supporting me in all my endeavor to reach my goals and become a successful person. Ghassan, Malake, Ola, Houssam, and Ghinwa without you and your love I would not have been able to be where I am now.

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LIST OF SYMBOLS

SYMBOL	UNIT	DESCRIPTION
V	V	Voltage
I	A	Current
R	Ω	Resistance
Q	W	Heat transfer rate
$\dot{m}$	kg/s	Mass flow rate of a fluid
C_p	$\mathrm{kJ/kg.°K}$	Specific heat
T	℃	Temperature
ρ	kg/m^3	Density
C	m^3/s	Flow rate
β	%	Fan % speed
α	m^3/s	Max coil capacity flow rate
$\dot{q}$	m^3/s	Theoretical flow rate through the coil
h	kJ/kg	Specific Enthalpy
ω	%	Relative Humidity
P	-	Final Parameter
T_x	-	Initial twelve bit number

LIST OF SUBSCRIPTS

a	air
w	water
S	Supply
O	outlet
I	Inlet

ABSTRACT

Khatib, Akram Ghassan. M.S.M.E., Purdue University, August 2014. Evaluation of Performance of an Air Handling Unit Using Wireless Monitoring System and Modeling. Major Professor: Jie Chen.

Heating, Ventilation, and Air Conditioning (HVAC) is the technology responsible to maintain temperature levels and air quality in buildings to certain standards. In a commercial setting, HVAC systems accounted for more than 50% of the total energy cost of the building in 2013[13]. New control methods are always being worked on to improve the effectiveness and efficiency of the system. These control systems include Model Predictive Control (MPC), Evolutionary algorithm (EA), Evolutionary Programming (EP), and Proportional-integral-derivative (PID) Controllers. Such control tools are used on new HVAC system to ensure the ultimate efficiency and ensure the comfort of occupants. However, there is a need for a system that can monitor the energy performance of the HAVC system and ensure that it is operating in its optimal operation and controlled as expected.

In this thesis, an air handling unit (AHU) of an HVAC system was modeled to analyze its performance using real data collected from an operating AHU using a wireless monitoring system. The purpose was to monitor the AHU's performance, analyze its key

parameters to identify flaws, and evaluate the energy waste. This system will provide the maintenance personnel to key information to them to act for increasing energy efficiency. The mechanical model was experimentally validated first. Them a baseline operating condition was established. Finally, the system under extreme weather conditions was evaluated. The AHU's subsystem performance, the energy consumption and the potential wastes were monitored and quantified.

The developed system was able to constantly monitor the system and report to the maintenance personnel the information they need. I can be used to identify energy savings opportunities due to controls malfunction. Implementation of this system will provide the system's key performance indicators, offer feedback for adjustment of control strategies, and identify the potential savings.

To further verify the capabilities of the model, a case study was performed on an air handling unit on campus for a three month monitoring period. According to the mechanical model, a total of 63,455 kWh can be potentially saved on the unit by adjusting controls. In addition the mechanical model was able to identify other energy savings opportunities due to set point changes that may result in a total of 77,141 kWh.

1. INTRODUCTION

1.1 Problem Statement

A large quantity of today's research is aimed to reduce the overall energy consumption on our planet and move closely to a more energy sensible lifestyle. The main reason for the increased interest in energy conservation is the increasing damage that is done to the environment due to the release of carbon dioxide and other greenhouse gases into the atmosphere. The building sector has the highest percentage of energy usage in comparison with transportation and industrial operations (Figure 1.1). In addition, the building sector dominates 74% of the total electricity used in the U.S. (Figure 1.2). This being said, and to reduce the amount of greenhouse gasses being dumped in the atmosphere every day, we need to focus on the building sector to effectively reduce the total energy consumption and reduce the usage baseline. Within the building sector, Heating, ventilation, and air conditioning (HVAC) systems make up the major energy users with up to 50%[2]. Therefore it is important to improve the performance of HVAC systems in order to achieve a reduction in the energy consumed in the building sector.

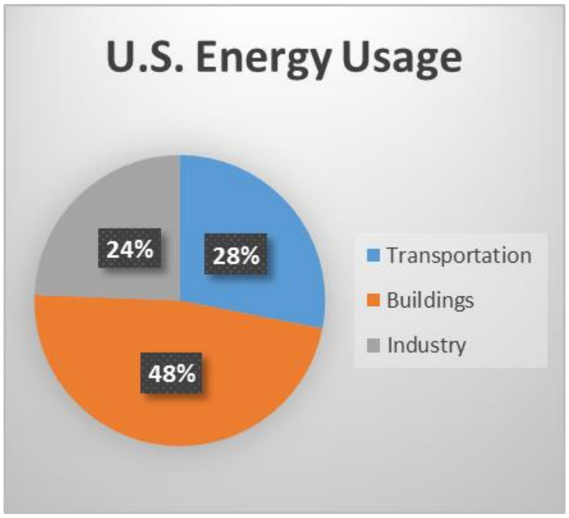


Figure 1.1 U.S. Energy Usage by sector [8]

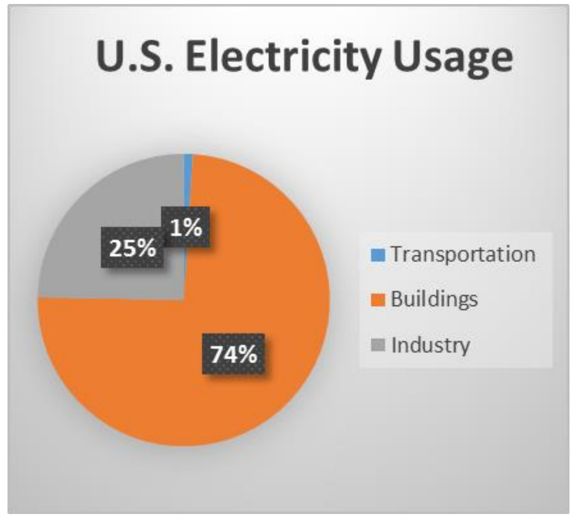


Figure 1.2 U.S. Electricity Usage by sector [8]

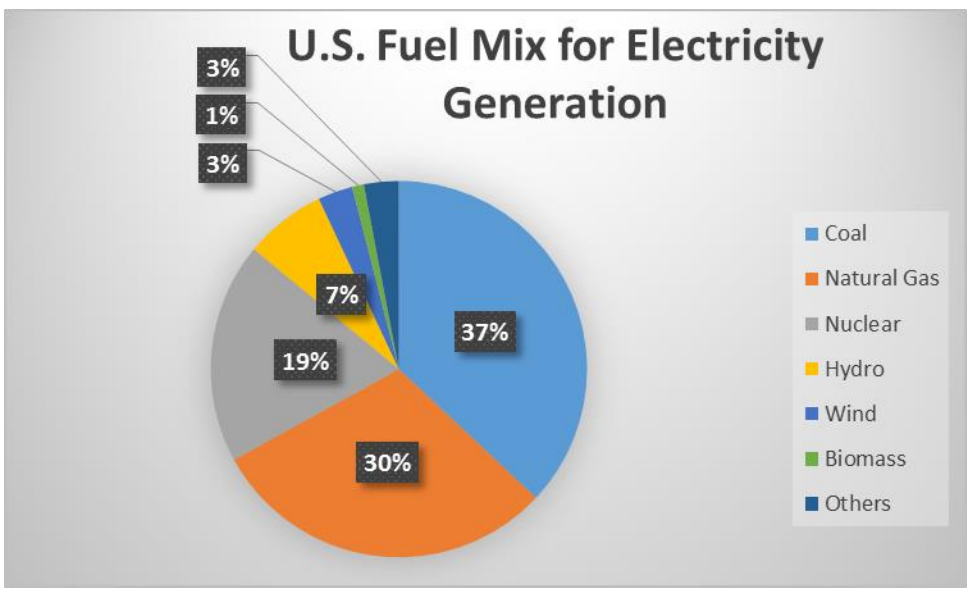


Figure 1.3 Electricity Production by Fuel Sources [9]

1.2 Previous Work

In the past, many research has been done on ways to improve the overall performance of HVAC systems using different control strategies. Different control strategies have been developed such as Model Predictive Control (MPC)[2], Evolutionary algorithm (EA)[6], Evolutionary Programming (EP)[4], and Proportional-integral-derivative (PID) [3]Controllers. However, all these control systems do not allow for a validation of the energy performance of an HVAC system, especially because those control system should be adjusted for climate conditions, internal loads, and building shape.

A study on data-driven (inverse) approach has been done in order to determine a mathematical description of the system and to estimate system performance in terms of system parameters [10]. This approach in contrast to forward approach, which is used in Model Predictive Control, requires real data to be measured for a certain period of time and buildings performance to be known ahead of time. The inverse data-driven approach is currently used for evaluation or fault detection. The data- driven approach is popular in many fields, such as physics, biology, engineering and economics. This approach is classified into two modeling approaches.

First approach is called the “Black-Box” model where a simple or multivariable regression model between measured energy use and the various influential parameters (Building occupancy, climatic variables, etc…) is identified. This is non-physical model approach (statistical/stochastic) without physically significant parameters. They reflect again the structure of the real behavior of the system using coefficient according to a regression analysis. At the end, a model is obtained that has the ability to have sufficient accurate relationships between the input and output variables. Due to its limitations, it is mainly used for error detection not for optimization. This approach is appropriate to evaluate the demand side management programs, identify energy conservation measures in an existing building and to develop baseline models in energy conservation measurements and verification projects.

Second approach is called the “Grey-Box” model. The gray box model gives a representation of the physical structure of a building, develops a physical model, and selects the most important parameters representing the aggregated physical model statistically. The grey-box approach has a great potential for fault detection and support diagnosis. This grey method has been presented by Coffey [11] as a software framework what could be characterized as a “grey-box” approach in predictive control since it combines physically-based models with a generic algorithm. The result from a case study demonstrated that the framework can be used to minimize cooling demand through optimal demand response using zone temperature ramping in an office space [12]. This approach still needs improvement to increase the effectiveness in solving complex problems like HVAC system control.

1.3 Objectives

The goal of this research is to reduce energy consumption of an air handling unit in a commercial setting. In particular, this thesis focuses on analyzing and predicting the performance of heating and cooling coils in an AHU. Different parameters specified in

Section 4.1 were used as inputs to a mechanical model that will then be used to predict and compare energy consumption. The mechanical model uses heat transfer equations and control logic to anticipate the operation of the coil at a certain set temperature. The detailed analysis of those subsystems allowed for clear all inclusive mathematical model that is capable of evaluating efficiencies and performance the unit. Using monitored parameters to evaluate performance allows for this objective has been accomplished by initially picking a wireless system to be used to monitor control parameter sensors that are already in use in the AHU. Through this system, performance information were collected on a server, outliers were filtered, and parameters were converted into more common and understandable units. A MatLab code was then used to analyze the performance of the AHU and establish a baseline of energy usage. This baseline is then compared to the actual operation of the unit and energy consumption difference was calculated

1.4 About This Thesis

Following the introduction in Chapter 1, Chapter 2 defines a HVAC system in addition to defining an air handling unit (AHU). Chapter 2 also explains the main energy users in an AHU and their mathematical representation. Finally Chapter 2 identifies the main parameters needed to be monitored in order to satisfy the mathematical equations. Chapter 3 identifies the unit that was picked to perform experiments on, discusses the control sequence of this unit and identifies the used parameters in the unit's control system. Chapter 4 goes in details about how the mechanical model was adjusted to be able to accept the already monitored parameters. In addition, Chapter 4 describes the wireless system that was used to collect the data needed, and also describes the coding and algorithms behind compiling all collected data. Chapter 5 describes the validation process of the mechanical model then shows results due to applying the mechanical model on current operating conditions. Moreover, Chapter 5 shows calculations of waste energy in the current operation of the unit in comparison with the mechanical model. Finally Chapter 5 shows results of a set point analyses that identifies the optimal supply air set point in different weather conditions. Results are discussed and future recommendations are identified in Chapter 6.

2 METHODS

2.1 HVAC Definition

HVAC systems are integral to a building's ability to maintain a comfortable space for a wide variety of usages, and thus are tailored to meet different needs for thermal control. Buildings with different purposed and in different climates have different cooling/heating loads, in addition to specific humidification requirements. Therefore, it is important to learn the operation of the building in addition to its internal loads before sizing or installing an HVAC unit. In addition, it isn't possible to create an efficient universal HVAC unit that can be installed in different buildings, therefore, different types of controls and systems are created for every specific building.

2.1.1 HVAC Components

A HVAC system as a whole is a result of different smaller components operating in sync and delivering conditioned air to building rooms. Figure 2.1 shows a simple diagram of the different components that make up an HVAC system.

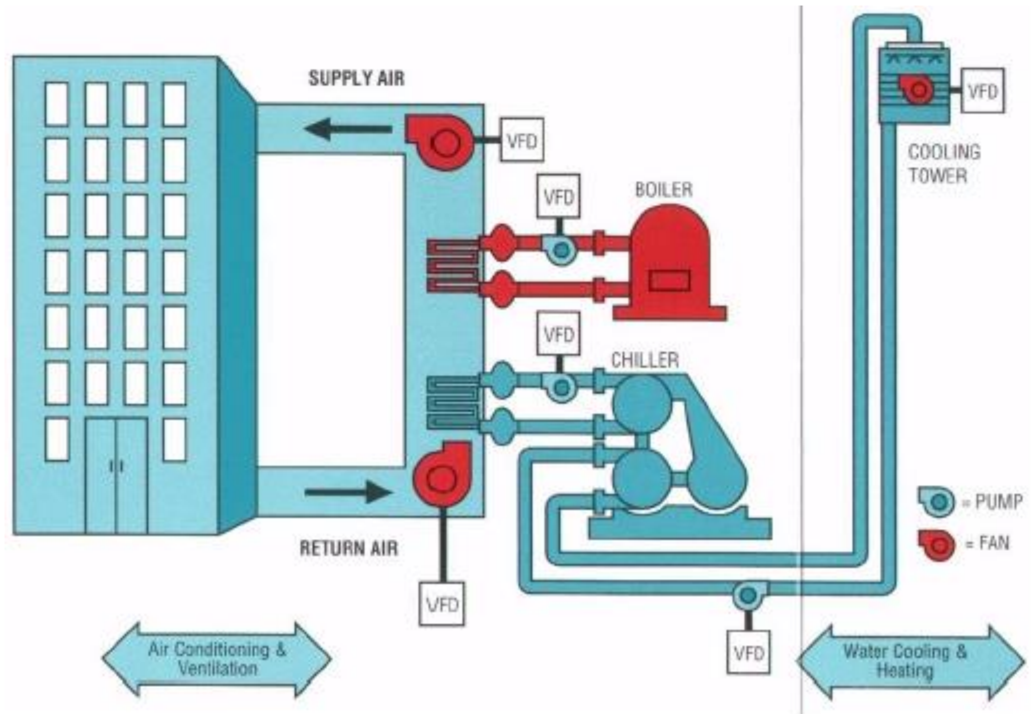


Figure 2.1 Heating, ventilation, and air conditioning system diagram [14]

Figure 2.1 shows a simple view of different components of an HVAC system. The chiller and cooling tower are there to remove energy from the return air and dispense it into the environment. The boiler is responsible for adding heat and energy to the supply air going into the building. The fans are essential for moving air through the ducts and into the office spaces and rooms. Inside the building different air handling systems can be used to deliver air as needed to each room in the building.

For the purposes of this research, we will be only studying variable air volume (VAV) system since this is what is being used in the monitored unit.

In a VAV system, one thermostat can be responsible for controlling the air flow in more than one room. Figure 2.2 shows a diagram of a VAV box delivering air to more than one area at a time.

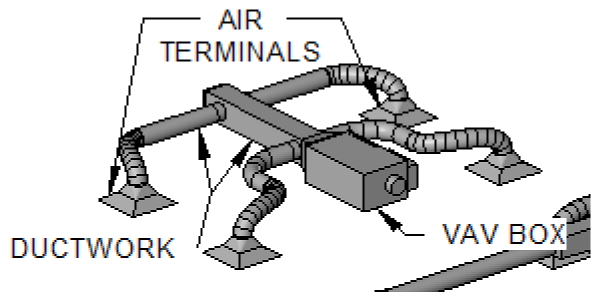


Figure 2.2 Variable Air Volume diagram [15]

As seen in Figure 2.2 one VAV box is connected to one main duct that is delivering conditioned air, this one air supply is then controlled by a damper and supplied to each air terminals.

In this thesis the main focus is on the operation of Air Handling Units. Therefore, in order to reduce energy usage of an AHU it is needed to study the energy usage of each component of an AHU and develop a theoretical model that will allow to study the energy consumption of the unit.

2.1.2 AHU Energy Usage

Energy users in an AHU are broken down into two categories. The heat exchangers and the electric users. The heat exchangers are the heating coils and the cooling coils, and the electric users are the fan and the control system.

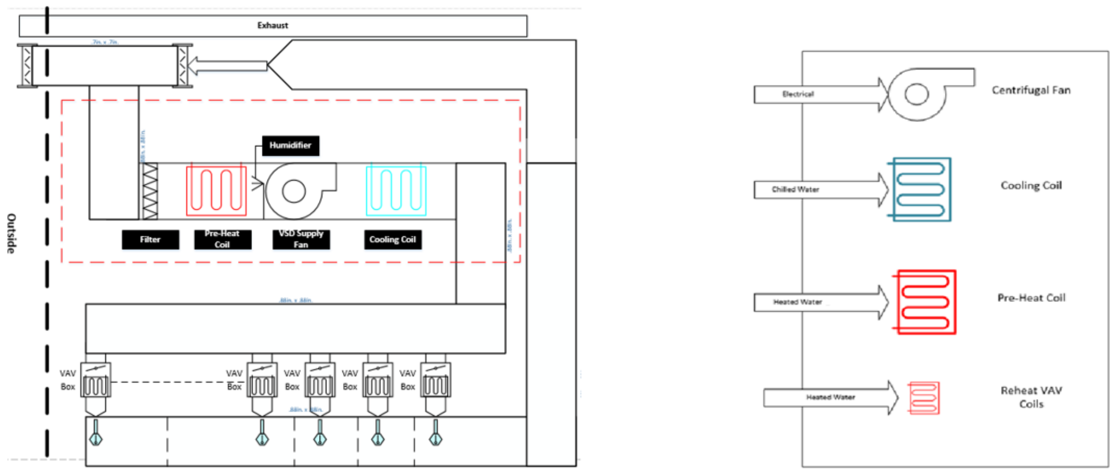


Figure 2.3 Air handling unit energy users

Figure 2.3 shows a diagram of the energy users of an AHU. Depending on the unit, there might be different components added to the system, however the components shown above are the most commonly used.

2.2 Theory and Mechanical Model

In order to evaluate the performance of each component of the AHU, a mechanical model had to be introduced in order to be able to identify the inputs needed to run the mechanical model. Therefore, electrical and heat exchanging equations were used as shown below.

Fan Energy usage

A fan in an AHU is controlled by a Variable Speed Drive (VSD) that controls the speed of the fan depending on certain monitored parameter.

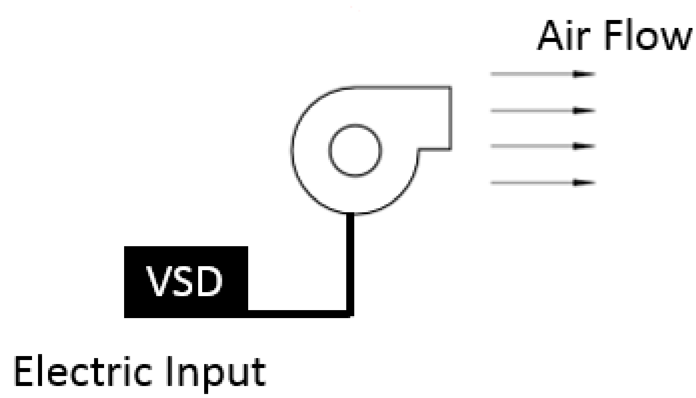


Figure 2.4 Variable speed drive controlled air handling unit fan.

Governing Equation

$$P = \frac{V \times I \times PF \times \sqrt{3}}{1000} \tag{1}$$

$P = Three - Phase\ Power\ in\ kW$

$V = RMS\ Voltage$

$I = RMS\ Current$

$PF = Power\ Factor$

Sensible Heating

Sensible heating and cooling occurs when the temperature of an air-water mixture is raised (lowered) but the absolute moisture content remains the same. Sensible heating or cooling occurs as air in spaces is warmed or cooled by building loads that do not change the moisture content of the air. Sensible heating or cooling is also performed by systems to compensate for loads. For instance, room air may be cooled by an outside wall during cold winter weather. To compensate, a heater at the base of the wall may warm the air.

In this research, the mathematical model used the law of energy transfer to calculate the amount of sensible heat needed to increase the temperature of the air to the set point by adjusting the valve on the coil. [8]

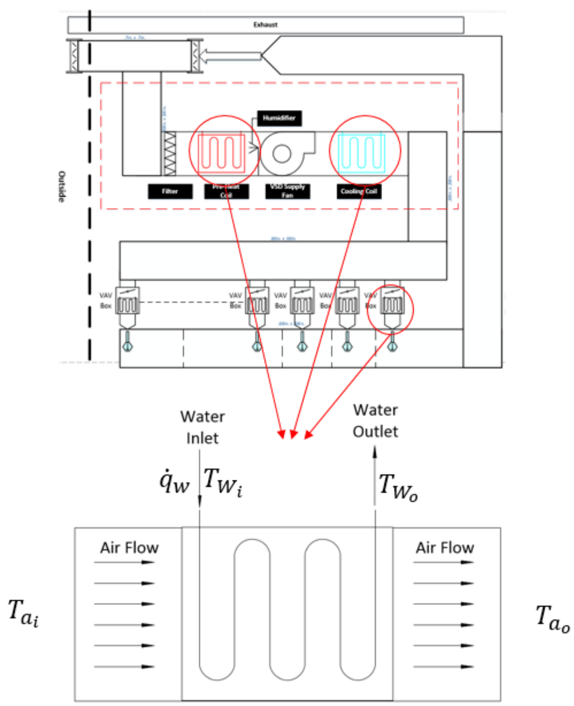


Figure 2.5 Heating and cooling coils in an AHU modeled as air to liquid heat exchanger

To calculate sensible energy added to or removed from the supplied air, the following equation can be used.

$$\dot{Q}_a = \dot{Q}_w$$

$$\dot{m}_a C_{p,a} \Delta T_a = \dot{m}_w C_{p,w} \Delta T_w$$

$$\dot{m}_w = \frac{\dot{m}_a C_{p,a} \Delta T_a}{C_{p,w} \Delta T_w} \quad (2)$$

Where $\dot{m}_w$ is the mass flow rate required to change the temperature of the air by ΔT_a.

By knowing the mass flow rate and the inlet/outlet temperatures of the coil we are then capable of calculating total energy input into the system.

Latent Cooling

Heating and cooling represent a change of sensible heat; humidification and dehumidification represents change of latent heat. The amount of moisture liberated or absorbed by air is measured by its initial and final absolute humidity. [8]

Latent energy flow

$$\dot{Q}_L = \frac{h_{fg} \rho_a \dot{q} \Delta W}{3600} \quad (3)$$

In case of a cooling coil, the total energy will be the result of the addition of the sensible heat and latent heating.

$$\dot{Q}_T = \dot{Q}_S + \dot{Q}_L \quad (4)$$

2.3 Parameter Selection

Generating a mechanical model that is capable of analyzing AHU performance required monitoring operation parameters. Therefore, based on the mechanical equations that were defined for each component, a list of inputs was generated to allow for an accurate representation of each component in the model. This list along with parameter units is represented in Table 2.1.

Table 2.1 Needed parameters and their units

Parameter	Unit
Current	A
Air Mass Flow rate	Kg/s
Water Mass Flow rate	Kg/s
Inlet coil Temperature	°F
Outlet Coil Temperature	°F
Inlet Air Temperature	°F
Outlet Air Temperature	°F
Pre-Heat Air relative humidity	%
Supply Air relative Humidity	%

3 EXPERIMENTAL SETUP

The science and engineering (SL) building at IUPUI consists of three floors and a basement. Each floor has two AHUs that are responsible for maintaining adequate temperatures and air quality in the floor. On the second floor there are two AHU one is responsible for the east side and the other is for the west side. This chapter discusses in operation details of the east side AHU and the tools utilized in order to collect data from the unit.

3.1 Current AHU

The AHU is responsible for maintaining required temperatures in east half of the second floor of the science building. The AHU was and continues to provide air to a total of thirty three offices and seven labs. Total office square footage is 7,657 ft^2, and total lab area 6,236 ft^2. Figure 3.1 shows the office and lab layout of the second floor that are being conditioned by the AHU.

Figure 3.1 East half side of the Science building with offices and labs specified

The presence of the labs meant that there had to be an outside exhaust at all times. This is to make sure that none of the chemicals used in the labs is mixed with the return air to the unit. Exhaust fans were also used to maintain a negative pressure in the lab area to limit the leak of lab fumes into the plenum and maintain a positive pressure in the plenum with respect to the lab space.

Variable Air volume (VAV) boxes were used to distribute and reheat air before it reached the offices/labs. Each VAV box was responsible for one or more rooms depending on the size and location. There was one thermostat in every set of offices that was sending a signal to the VAV box dampers and reheat coil. This signal is responsible for controlling the dampers to supply more air at a higher temperature to the room to maintain the set temperature, or it can be asking for more air at a lower temperature. In both cases the thermostat was responsible to maintain the set temperature in the room. Figure 3.2 shows a schematic of the AHU with all the variable components.

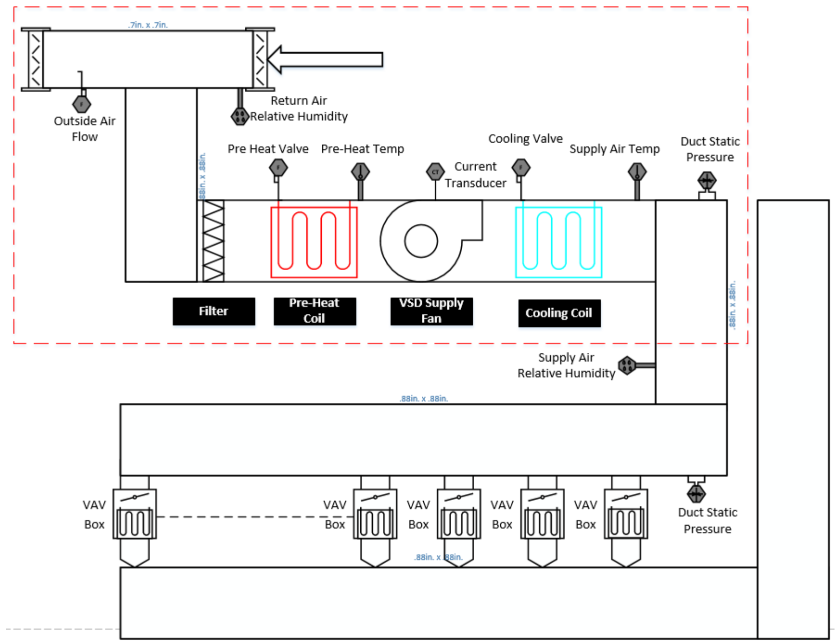


Figure 3.2 Current AHU Schematic with control sensors locations

3.2 Control Sequence

The AHU is controlled by a set of logic controllers that were programmed by campus engineers in each building. For this particular unit the controls were originally put together by Johnson Controls and additional adjustments were put in-place later by CFS. The main goal for the unit is to maintain air temperatures in the office areas.

One thing to note about this specific unit is that it does not monitor CO_2 levels in the building, and therefore is only maintaining a set temperature. The theoretical control sequence is as follows.

General Operation:

The AHU includes different components that are controlled through the control system those components are OA (Outside Air) & RA (Return Air) dampers, preheat coil, cooling coil, re-heat coil, humidifier and supply fan VFD. The AHU runs continuously and is commanded occupied/unoccupied thru the operator workstation. Occupied being daytime when students are attending students, and unoccupied being night time, holidays, and weekends.

Fan Control:

The supply fan shall run continuously. Supply duct static pressure shall be maintained by modulating the supply fan speed through the variable frequency drive(s) (VFD). The VFD is controlled to maintain a duct static set point of +2" WC (adjustable) at two locations in the duct system indicated on the drawings. The duct static pressure transmitters shall control the VFD through the AHU Controller. An electric static pressure high limit safety device shall stop the supply fan(s) whenever fan discharge static pressure rises above the 5" WC (Water Column) (adjustable) high limit.

Heating Control:

The preheat coil discharge air temperature shall be maintained by modulating control valve(s) in sequence with outdoor air dampers and return air dampers to maintain a preheat coil discharge air temperature set point of 53°F (adjustable).

Cooling Control:

The AHU discharge air temperature shall be maintained by modulating cooling coil control valve(s) in sequence with outdoor air dampers and return air dampers to maintain an AHU discharge air temperature set point of 55°F (adjustable). Based on space design conditions of 75db/50%RH db (Dry Bulb), when cooling is required and the outdoor air relative humidity is below the listed value (adjustable) at the corresponding outdoor air dry bulb temperature, the economizer operation shall provide free cooling by modulating the outdoor air dampers from minimum position to maximum position as the return air dampers are modulated from maximum position to minimum position.

Dehumidification Control:

The return air relative humidity sensor shall automatically initiate a dehumidification cycle if the relative humidity exceeds a high limit set point of 57% RH (adjustable). The AHU discharge air temperature shall be reset to 50°F. Outdoor air dampers shall be reset to minimum position. The space temperatures shall be maintained by the individual control of the VAV terminal box units with reheat coils. When the return air relative humidity falls below the set point, the AHU discharge temperature shall reset to normal control.

Humidifier Control:

The humidifier control valve shall be modulated to maintain a return air relative humidity set point of 35%RH (adjustable) as sensed by the return air relative humidity sensor. The supply air modulating high limit humidity sensor shall prevent the supply air duct humidity levels rising above 80% RH (adjustable). The air proving switch shall prevent the humidifier from operating under no air flow conditions. Controller to lock out humidifiers above 50°F outside air.

Shutdown:

When the unit is stopped, outdoor air dampers shall close, return air dampers shall open, chilled water valve shall close, humidifier control valve shall close, preheat coil valve shall remain in control, and the variable frequency drives shall ramp down.

3.3 Monitored AHU Parameters

The AHU already had different parameters monitored for control and troubleshooting purposes. A study of these parameters allows for a clear understanding of the function they can play in the mechanical model. Figure 3.3 shows the monitored points, set points, and status updates as seen by campus facility services (CFS).

Item	Value	Description
072-SL-AHU2E-OA-T	88.8 deg F	OUTDOOR AIR TEMPERATURE
072-SL-AHU2E-SF-S	On	SUPPLY FAN STATUS
072-SL-AHU2E-SF-C	On	SUPPLY FAN COMMAND
072-SL-AHU2E-RA-H	54.9 in wc	RETURN AIR HUMIDITY
072-SL-AHU2E-DRYBULB-SP	65.0 deg F	DRY BULB SETPOINT
072-SL-AHU2E-CLG-VLV	32.4 %	COOLING COIL VAVLE
072-SL-AHU2E-BLD-STAT	-0.043 in ...	BUILDING STATIC
072-SL-AHU2E-DA-T	55.0 deg F	DISCHARGE AIR TEMPERATUER
072-SL-AHU2E-OA-H	50.0 %RH	OUTDOOR AIR HUMIDITY
072-SL-AHU2E-DAT-SP	56.0 deg F	DISCHARGE AIR TEMPERATURE SETPOINT
072-SL-AHU2E-EE-FLOW	283 cfm	EXHAUST AIR FLOW (X 10)
072-SL-AHU2E-DEWPT-SP	55.0 deg F	DEWPOINT SETPOINT
072-SL-AHU2E-HUM-VLV	0.0 %	HUMIDIFER VALVE
072-SL-AHU2E-LOWLIMIT	Normal	LOWLIMIT STATUS
072-SL-AHU2E-MA-DMPR	25.0 %	MIXED AIR DAMPER
072-SL-AHU2E-OA-FLOW	853 cfm	OUTDOOR AIR FLOW (X 10)
072-SL-AHU2E-PH-T	63.5 deg F	PREHEAT AIR TEMPERATURE
072-SL-AHU2E-PHT-SP	55.0 deg F	PREHEAT AIR TEMPERATURE SETPOINT
072-SL-AHU2E-PHT-VLV	100.0 %	PREHEAT VAVLE
072-SL-AHU2E-RA-H	89.4 %RH	RETURN AIR HUMIDTY
072-SL-AHU2E-RAH-SP	0.0 %RH	RETURN AIR HUMIDITY SETPOINT
072-SL-AHU2E-RLF-C	On	RELIEF FAN COMMAND
072-SL-AHU2E-RLF-S	Off	RELIEF FAN STATUS
072-SL-AHU2E-RLF-VFD	0.0 %	RELIEF FAN VFD
072-SL-AHU2E-SA-DMPR	54.4 %	SUPPLY AIR DAMPER
072-SL-AHU2E-SF-VFD	100.0 %	SUPPLY FAN VFD
072-SL-AHU2E-DEW-PT	67.7 deg F	CALCULATED DEWPOINT
072-SL-AHU2E-SASTAT-SP	1.2 in wc	SUPPLY STATIC SETPOINT
072-SL-AHU2E-BLGDSP-SP	0.010 in wc	BUILDING STATIC SETPOINT
072-SL-AHU2E-SA-STAT	0.9 in wc	SUPPLY STATIC PRESSURE

Figure 3.3 Monitored AHU parameters, set points, and status updates as seen by CFS

Having the list of monitored parameters allowed for manipulating the original mechanical model in order for it to accommodate the inputs already monitored through the Metasys system. Input such as return air humidity was monitored as relative humidity, however in order to use it in the mathematical model, it had to be changed to humidity ratio. Other parameters, such as Pre-heat air temperature and outdoor air were converted into SI unit readings in order to fit the mechanical model.

4. ADJUSTING MECHANICAL MODEL AND COLLECTING DATA

4.1 Usage in the Mechanical Model

After identifying all parameters monitored in the AHU for control and troubleshooting purposes, a relationship between the needed parameters and monitored parameters was established. The purpose of this step is to utilize the presence of sensors on the unit and not having to install new sensors. Table 4.1 shows the relationship between the monitored parameters and the mechanical model.

Table 4.1 Relationship between monitored parameters and mechanical model

Parameter	Unit	Action	Equation
Current	A	Measured	-
Air Mass Flow rate	Kg/s	Can be calculated through the Fan Speed measurement	$\dot{m} = \rho\beta C_{Fan}$
Water Mass Flow rate	Kg/s	Can be estimated by monitoring the Valve position on the coils	$\dot{m} = \alpha\dot{M}$
Inlet coil Temperature	°F	Estimated from specification sheet	$T_c = 7.22℃$ $T_h = 93.33℃$
Outlet Coil Temperature	°F	Estimated from specification sheet	$T_c = 12.77℃$ $T_h = 77.22℃$
Inlet Air Temperature	°F	Measured	-
Outlet Air Temperature	°F	Measured	-
Pre-Heat Air relative humidity	%	Measured	-
Supply Air relative Humidity	%	Measured	-

A current transducer [17] was purchased in order to be able to monitor the energy usage of the supply fan. The current transducer provided a current reading that could be used in Equation (1).

To find the mass flow rate of water through the coils the valve position was monitored and the relationship between the valve position and flow rate was found

$$\dot{m} = \alpha \dot{M} \tag{5}$$

Where,

$\dot{m}$ is the current flow rate through the coil ($\frac{Kg}{s}$)

α is the valve position (%)

$\dot{M}$is the total flow rate capacity of the coil

Equation (5) assumes a linear relationship between the valve position and the percentage of max Flow rate. Figure 4.1 shows the different relationships between valve positions and percent flow rates. It was estimated that there is a linear relationship between the valve position and the flow rate

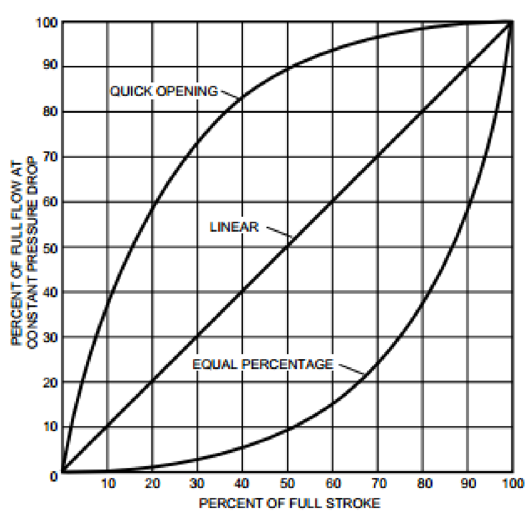


Figure 4.1 ASHRAE Valve Characteristics Graph[16]

In order to find the total mass flow rate of air through the coils, affinity laws were used. It states that fan speed is directly related to flow rate of air Equation 6.

$$\frac{q_1}{q_2} = \frac{n_1}{n_2} \tag{6}$$

Based on Equation 6 and knowing that at 100% speed the total CFM is 24000 the following relationship were obtained.

$$q_1 = 24000 \times n_1 \tag{7}$$

Inputs shown above in Table 4.1 represent the building blocks of the mechanical model. Now we will discuss how each parameter falls into place.

Heat transfer formulation

Sensible Heat

Sensible heat lost or gained by the air was calculated the following way. This allowed us to estimate the water flow rate required to reach a certain set point

$$Q_a = Q_W$$

$$\dot{m}_a C_{p,a} \Delta T_a = \dot{m}_w C_{p,w} \Delta T_w$$

$$\dot{m}_w = \frac{\dot{m}_a C_{p,a} \Delta T_a}{C_{p,w} \Delta T_w} \tag{8}$$

Some of this equation’s parameters can be manipulated the following way

Measured Air Mass Flow Rate ($\dot{m}_a$)

Because there was not a mass flow sensor installed on the fan, we could use the fan speed data to calculate the mass flow rate in the following way.

$$\dot{m}_{a,t} = \rho_a \beta C_{fan}$$

Total air mass flow rate

$$\dot{m}_a = \sum \rho_a \beta C_{fan}$$

Equation 8 becomes the following

$$\dot{m}_w = \frac{\rho_a \beta C_{Fan} C_{p,a} \Delta T_a}{C_{p,w} \Delta T_w} \tag{9}$$

Predicting Valve Position

Valve position can be predicted from the mechanical model and compared to the current operation through the following equations

$$\dot{m}_w = \rho \dot{q}$$

$$\dot{q} = \frac{\dot{m}_w}{\rho}$$

$\dot{q}$ is calculated based on the monitored air flow rate and temperature. In order to be able to compare this value with the monitored valve position value, am additional equation needs to be added in order to obtain a valve position estimate

$$\%\,Valve = \frac{\dot{q}}{\alpha}$$

Latent Heat Calculations

Based on Equation (3) (Latent Heat), energy due to removing latent heat from the supply air can be calculated. h_{fg} is the enthalpy of evaporation which was obtained from the saturated steam tables at 55 °F (12.78℃)and 1 kPa (absolute).

$$h_{fg} = 2465.56 \frac{kJ}{kg}$$

Engineering Equation Solver (EES) was used to convert the measured relative humidity into humidity ratio in order to be used in Equation (3). Finally, the density of air was estimated to be constant at $1.25 \frac{kg}{m^3}$.

Therefore, Equation (3) now becomes

$$\dot{Q}_L = 0.856\dot{q}\Delta W \tag{10}$$

Total energy removed due to passing through the cooling coil is then calculated by combining the sensible heat and latent heat equations

$$\dot{Q}_T = \dot{Q}_S + \dot{Q}_L$$

$$\dot{Q}_T = \rho_a \beta C_{Fan} C_{p,a} \Delta T_a + 0.856\dot{q}\Delta W \tag{11}$$

4.2 Main System Components

For this thesis a wireless system was used to evaluate the performance of the Air Handling Unit (AHU). A Fully Functioning Device (FFD) was installed to transmit data from the AHU to the server using ZigBee. ZigBee is a specification for a suite of high level communication protocols used to create Personal Area Networks (PAN) for small, low power digital radios. Two ECOMM WC21-1048-ENC4X FFD was used as a communication link between the sensors and the PAN coordinator. The ECOMM WC21-1048-ENC4X is capable of reading analog signals, in addition to having four analog outputs and eight digital outputs. Figures 4.2 and 4.3 show the ECOMM communication station in addition to a breakdown of the physical characteristics.

Figure 4.2 ECOMM WC21-1048-ENC4X in its shielding box including a 24VAC transformer

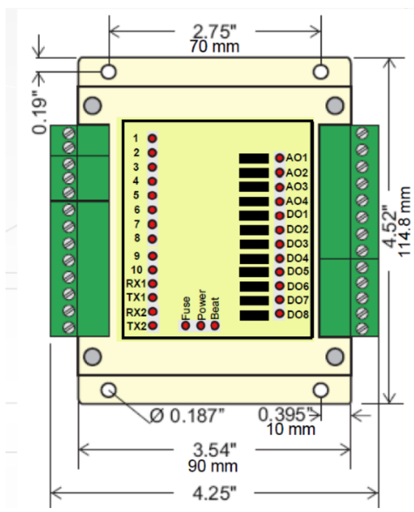


Figure 4.3 ECOMM WC21-1048-ENC4X's physical characteristics with the total input/output positioning.

The WC1-1048 is capable of receiving 10 inputs the support 0-3 VDC, 0-20 ma, 0-10V, Contact Closure and 10 K Ohms type three thermistor.

To adjust the input signal the jumpers in the circuit board has to be adjusted to the desired position as seen in the figure below.

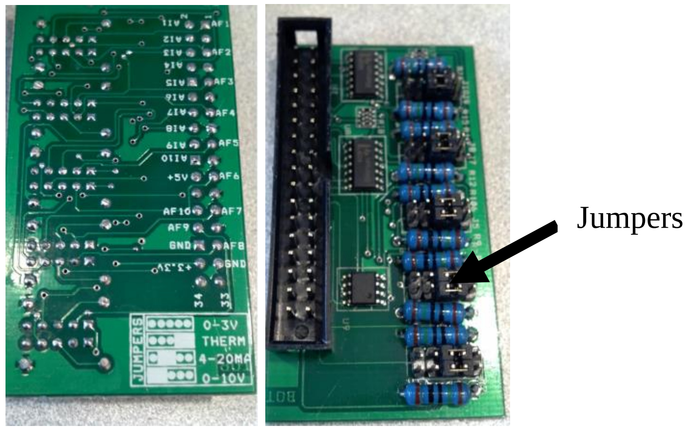


Figure 4.4 Circuit board that specifies the required position of the jumpers for each signal input at each channel.

After setting the type of input the system is receiving, the mesh connection was worked on to ensure the connection between the ECOMM and the PAN coordinator that was placed in an office are around 50ft from the main platform.

There were certain limitation on the distance and type of barriers that should separate the ECOMM from the PAN coordinator, those limitations had to be taken into consideration to make sure the connection was constant and reliable. Such limitations could differ depending on the type of walls in the building, different electrical equipment surrounding the PAN coordinator, and finally noise that could be a result of radio waves in the area.

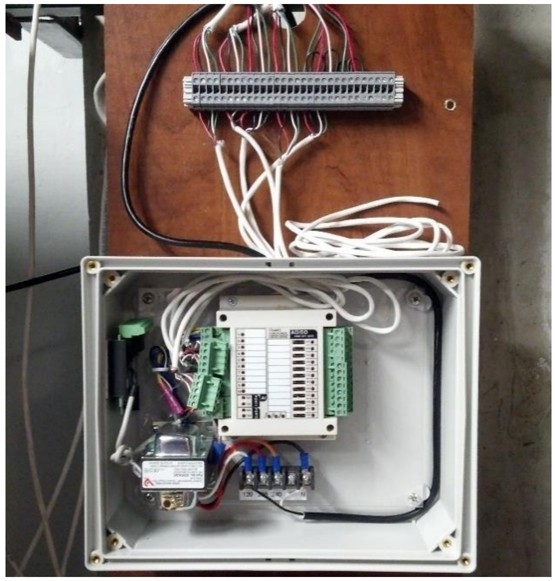


Figure 4.5 Initial Setup of the FFD with 10 channels connected

Figure 4.5 shows the initial FFD that is connected to AHU sensors and receiving 0-10V signals. A rail block was essential to make sure there was a termination point in case an error occurs. It is important to carefully not to short any of the connection, for that might result in a system failure. The best was to connect to the FFD is to connect to the green screw ins first while the FFD is powered off, then if the connections are verified and all essential checks are made you can plug in the FFD.

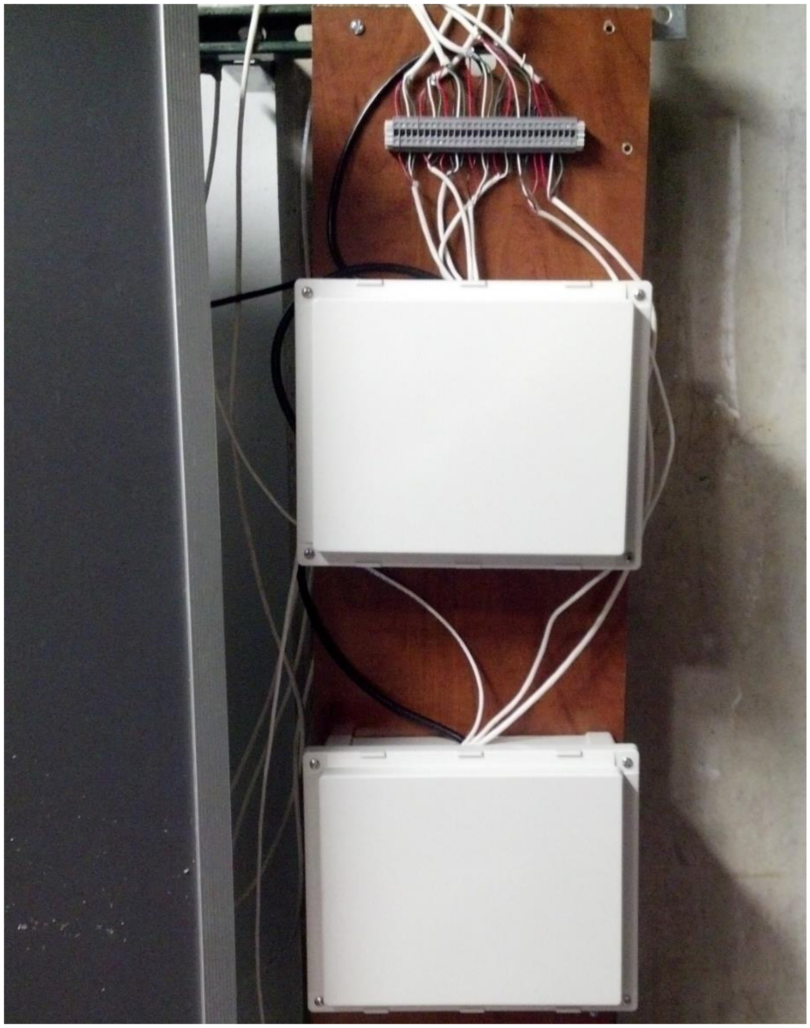


Figure 4.6 Final double setup of FFD

Figure 4.6 shows the final set up of the FFD and the signal wires connected to them. Power form the top FFD was used to power the lower FFD rather than running a new power cable.

4.3 Sensor Connection

The current AHU had its own sensors for controls purposes. For this thesis it was decided to tap into the incoming signal from the mentioned sensors. Input signals for the sensors varied between 4-20 ma, and 0-10V signals. To be able to monitor the 4-20 ma signal, it was decided to add a 150 ohm resistor to the circuit and measuring the voltage across the nodes by following ohms law

$$V = I \times R$$

Where V is the voltage drop across the nodes, I is the current flowing though the resistor and R is the resistance of the installed resistor.

This method proved to be a failure because the voltage drop across the resistor was causing the unit to receive wrong information that made it operated under false parameters. Therefore, it was decided to use a DX controller that would receive a gathering signal that included the inputs of all sensors, and break them into individual inputs of 0-10V to be connected to the ECOMM module.

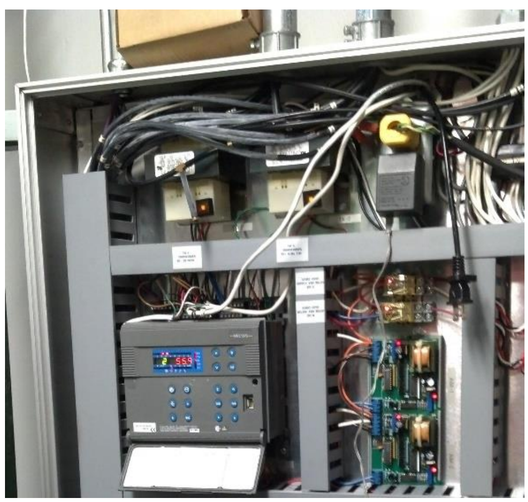


Figure 4.7 Original DX controller responsible for AHU controls

Figure 4.8 Additional DX controller responsible for sending signal to the FFD

Figure 4.7 shows that current DX controller that is being used to control the AHU with certain inputs established by the building engineers. Figure 4.8 shows the additional DX controller that was added to split the main signal output of the original DX into 0-10V signals that were picked up by the FFD.

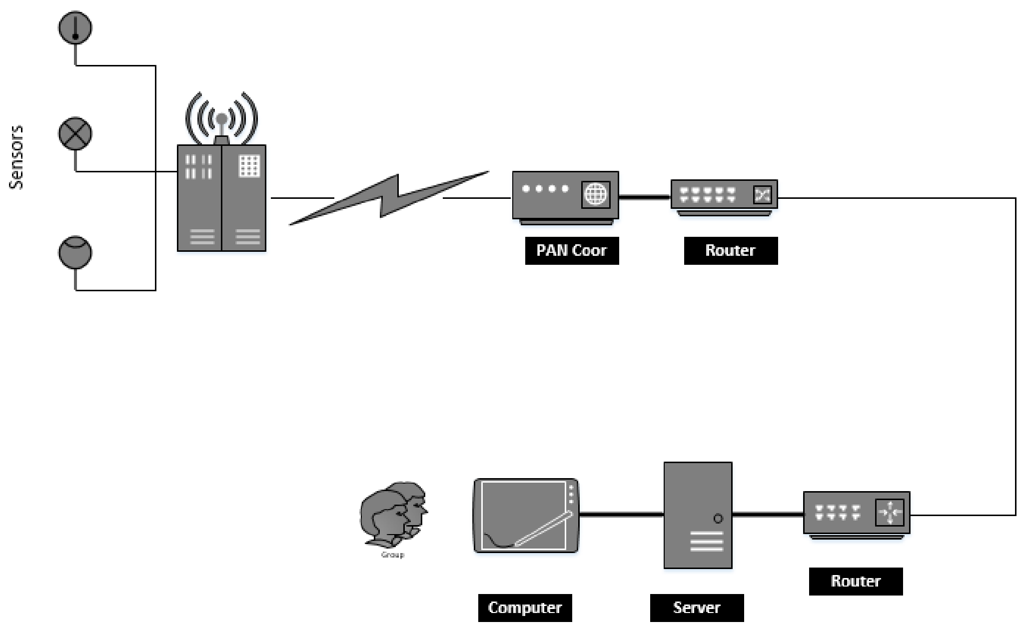


Figure 4.9 Wireless Architecture showing the signal received from the sensors to it being viewed by the user

Figure 4.9 is a visual illustration of the connection between the user and the unit sensors. The PAN Coordinator required a static IP address to make sure that it did not change over time and it could be reliable. A static IP address allowed the server to stay connected to the device at all times with no problem.

The FFDs and the PAN coordinator should be placed in a safe and secure place away from any electromagnetic waves that might interfere with the signal. Those electromagnetic waves could be due to a transformer, variable speed drive, and leaky microwaves and so on. Finally, it is important to take into account the distances and locations of the PAN coordinator and the FFDs.

4.4 Wireless Receiving Software

The PAN coordinator has a built in software package that allows the user to connect to through the static IP address listed above. This software is protected by a username and a password that is defined by the user. Each wireless system (FFD, or RFD) will have a tab specific for it in the software. This tab allows the user to adjust the settings of the wireless system as needed.

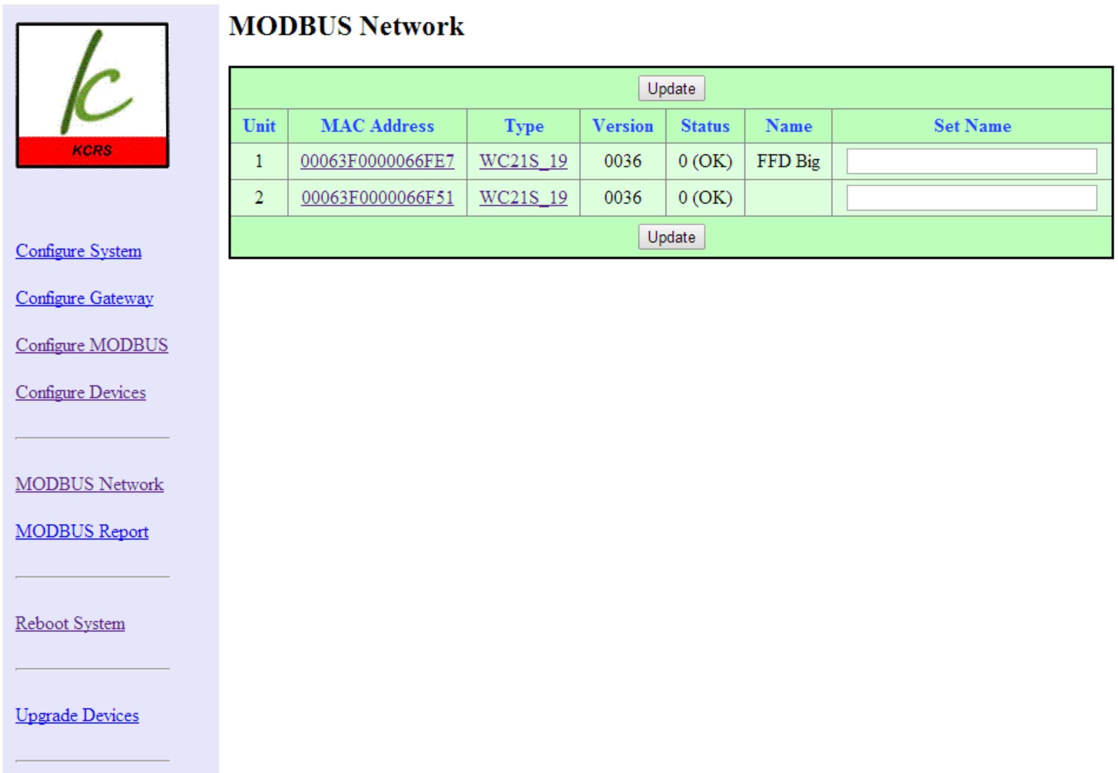
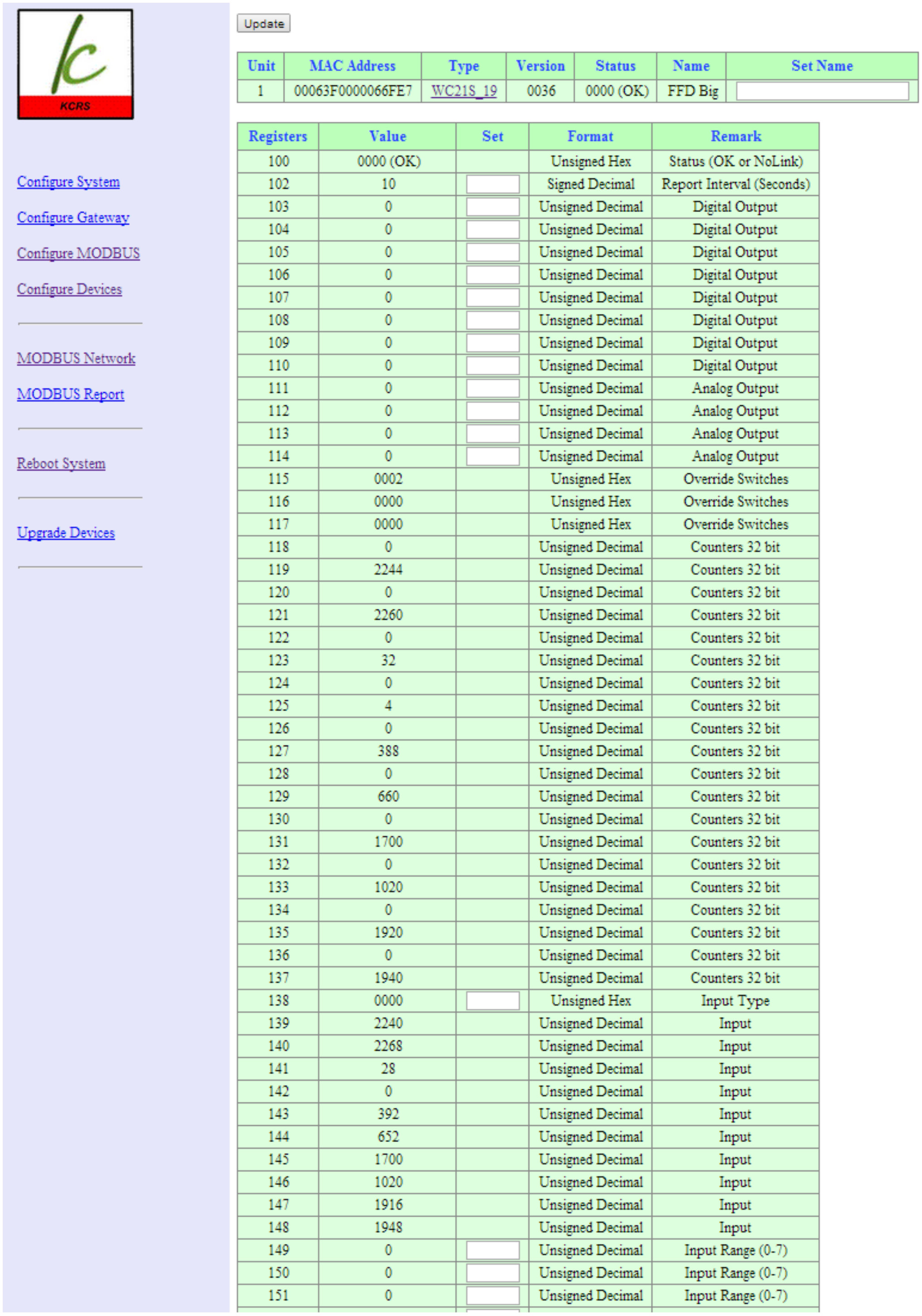


Figure 4.10 KCRS EC1 PAN Coordinator's main page, showing two units connected with different tabs for each

By clicking on the tab for the FFD, a new page will open up that allows the user to view the monitored points in real time, in addition to making adjustments to the FFD remotely. Some of those features are illustrated in Figure 4.11.



Unit	MAC Address	Type	Version	Status	Name	Set Name
1	00063F0000066FE7	WC21S_19	0036	0000 (OK)	FFD Big	

Registers	Value	Set	Format	Remark
100	0000 (OK)		Unsigned Hex	Status (OK or NoLink)
102	10		Signed Decimal	Report Interval (Seconds)
103	0		Unsigned Decimal	Digital Output
104	0		Unsigned Decimal	Digital Output
105	0		Unsigned Decimal	Digital Output
106	0		Unsigned Decimal	Digital Output
107	0		Unsigned Decimal	Digital Output
108	0		Unsigned Decimal	Digital Output
109	0		Unsigned Decimal	Digital Output
110	0		Unsigned Decimal	Digital Output
111	0		Unsigned Decimal	Analog Output
112	0		Unsigned Decimal	Analog Output
113	0		Unsigned Decimal	Analog Output
114	0		Unsigned Decimal	Analog Output
115	0002		Unsigned Hex	Override Switches
116	0000		Unsigned Hex	Override Switches
117	0000		Unsigned Hex	Override Switches
118	0		Unsigned Decimal	Counters 32 bit
119	2244		Unsigned Decimal	Counters 32 bit
120	0		Unsigned Decimal	Counters 32 bit
121	2260		Unsigned Decimal	Counters 32 bit
122	0		Unsigned Decimal	Counters 32 bit
123	32		Unsigned Decimal	Counters 32 bit
124	0		Unsigned Decimal	Counters 32 bit
125	4		Unsigned Decimal	Counters 32 bit
126	0		Unsigned Decimal	Counters 32 bit
127	388		Unsigned Decimal	Counters 32 bit
128	0		Unsigned Decimal	Counters 32 bit
129	660		Unsigned Decimal	Counters 32 bit
130	0		Unsigned Decimal	Counters 32 bit
131	1700		Unsigned Decimal	Counters 32 bit
132	0		Unsigned Decimal	Counters 32 bit
133	1020		Unsigned Decimal	Counters 32 bit
134	0		Unsigned Decimal	Counters 32 bit
135	1920		Unsigned Decimal	Counters 32 bit
136	0		Unsigned Decimal	Counters 32 bit
137	1940		Unsigned Decimal	Counters 32 bit
138	0000		Unsigned Hex	Input Type
139	2240		Unsigned Decimal	Input
140	2268		Unsigned Decimal	Input
141	28		Unsigned Decimal	Input
142	0		Unsigned Decimal	Input
143	392		Unsigned Decimal	Input
144	652		Unsigned Decimal	Input
145	1700		Unsigned Decimal	Input
146	1020		Unsigned Decimal	Input
147	1916		Unsigned Decimal	Input
148	1948		Unsigned Decimal	Input
149	0		Unsigned Decimal	Input Range (0-7)
150	0		Unsigned Decimal	Input Range (0-7)
151	0		Unsigned Decimal	Input Range (0-7)

Figure 4.11 Main FFD control page

Figure 4.11 shows the controls page of the main FFD box. On this page the user is able to view all the registers, control the input signal (149-158), control output signal (102-114), and view sensor inputs (139-148). The device has 8 digital output controls in addition to 4 Analog outputs. This option was not utilized in this research because of the sensitivity of the machine and the fact that a building engineer approval should be attained before any changes to the system set points.

The signal received from the FFD was in the form of a twelve bit analog number with a minimum of 0 and a maximum of 4096. Section 4.6 shows the detailed calculation to transform the twelve bit number into a useful measurement. For the purposes of this research registers 139 through 148 were closely monitored because they were the ones containing the sensor signals.

4.5 Configuring MODBUS

MODBUS is a messaging structure developed by Modicon in 1979. It is used to establish master-slave/client-server communication between intelligent devices. It is mostly used in industrial manufacturing environment [7]. By configuring the Modbus to FTP (File Transfer Protocol) the user is able to receive CSV files on a server with a specific IP address. Figure 4.12 shows the setup of the FTP page.

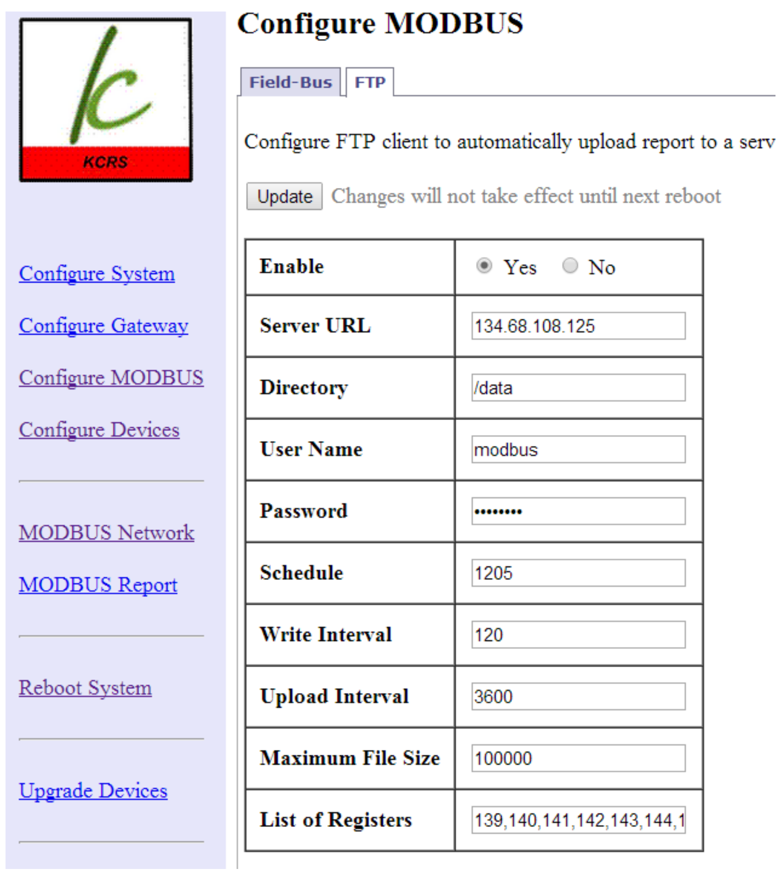


Figure 4.12 FTP Modbus setup

First step to enable the FTP option on the PAN coordinator is to set up a server with a static IP address to maintain a constant connection. Second is to install a FTP transfer software, for this research FileZilla was used to receive data. Third is to create a secure account with a secure password and a set directory on the server. When all of that is done the user is able to fill out the schedule, write interval, upload interval, and the list of registers that the user desires to monitor.

4.6 Signal Verification and Transformation

In order to verify the signal viewed on the server with the signal received form the sensors, a voltmeter was used to compare actual voltage readings at the sensor with voltage readings received on the server. To verify that this signal holds for the range of inputs (0 – 10 V), the set points for the system were manually adjusted to be able to reach the entire range, and the signal was measured for each set point. Results are shown below.

Table 4.2 Input channels with their type and range

Terminal	Channel	Type	Range	Units	Signal
1	AI 1	Pre-Heat Temperature	0-100	°F	0-10 V
2	AI 2	Discharge Temperature	0-100	°F	0-10V
3	AI 3	Supply Air Stat Pressure	0-5	In wc	0-10V
4	AI 4	Building Stat Pressure	(-0.5 +0.5)	In wc	0-10V
5	AI 5	Outside Air Flow	0-87,440	CFM	0-10V
6	AI 6	Exhaust Air Flow	0-17,500	CFM	0-10V
7	AI 7	Supply Air Humidity	0-100%	-	0-10V
8	AI 8	Return Air Humidity	0-100%	-	0-10V
9	AO 10	Mixed Air damper	0-100%	-	0-10V
10	AO 11	Pre-Heat Valve	0-100%	-	0-10V
11	AO 12	Cooling Valve	0-100%	-	0-10V
12	AO 13	Supply Fan Speed	0-100%	-	0-10V
13	AO 14	Relief Fan Speed	0-100%	-	0-10V
14	AI 9	Outside Temperature	0-100	°F	0-10V
15	AI 10	Fan Current	0-24	Amps	0-3V

The range for each sensor was attained from either the spec sheet or from the building engineers. However, a couple of sensors did not fall into the 0-10 V range, for those sensors a separate equation was calculated to be able to get an accurate reading.

Pre-Heat Temperature

$$P_1 = \left(\frac{T_1}{4096}\right) \times 100$$

Supply air temperature

$$P_2 = \left(\frac{T_2}{4096}\right) \times 100$$

Supply air static pressure

$$P_3 = \left(\frac{T_3}{4096}\right) \times 111$$

Since the static pressure sensor had a range of 0-111 IWC the equation for parameter three P_3 took into account this rage a measurement from the twelve bit number.

Building static pressure

$$P_4 = \left(\frac{T_4}{4096}\right) \times 100$$

Outside air Flow

$$P_5 = \left(\frac{T_5}{4096}\right) \times 87{,}440$$

Exhaust air flow

$$P_6 = \left(\frac{T_6}{4096}\right) \times 17{,}500$$

Exhaust air is a result of the lab exhausts and the fume hoods in labs. This exhaust had a maximum CFM of 17,500 CFM.

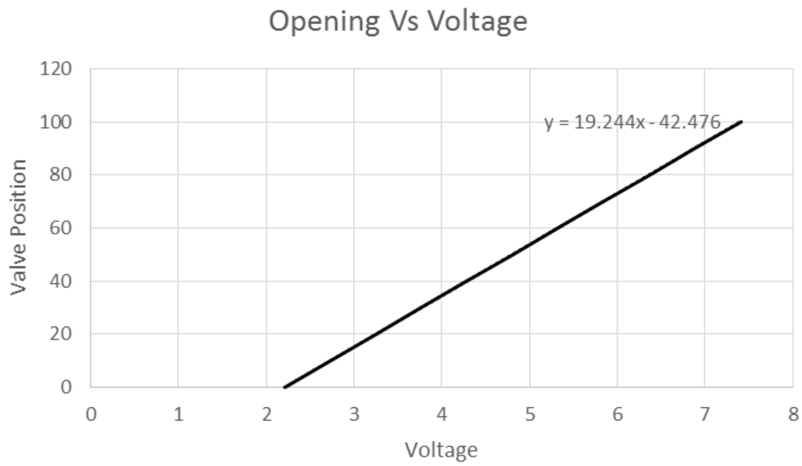
Supply air humidity

$$P_7 = \left(\frac{T_7}{4096}\right) \times 100$$

Return Air Humidity

$$P_8 = \left(\frac{T_8}{4096}\right) \times 100$$

Mixed air damper



Damper position %	Voltage
0	2.21
10	2.73
20	3.25
30	3.76
40	4.28
50	4.81
60	5.32
70	5.84
80	6.37
90	6.88
100	7.41

Figure 4.13 Mixed air damper position vs Voltage read by the wireless system

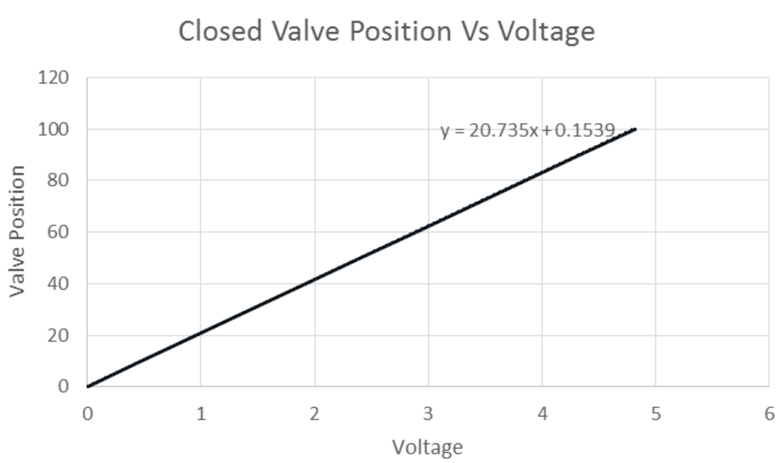
Figure 4.13 shows the mixed air damper position data, in addition to the equation of the trend line that is essential to be able to calculate the actual output signal of the damper.

$$P_9 = 19.244 \times \left(\frac{T_9}{4096}\right) - 42.476$$

The twelve bit number was first divided by $2^{12} = 4096$ to get a percentage of the total signal, then the equation from the graph is used to get the actual sensor measurement.

Mixed air damper signal is responsible for controlling the return air damper and the outside air damper. When the signal is 100%, it means that the unit is not using any return air (return air dampers are shut) and is using 100% outside air (economizing) outside dampers are fully open.

<u>Pre-Heat Valve</u>



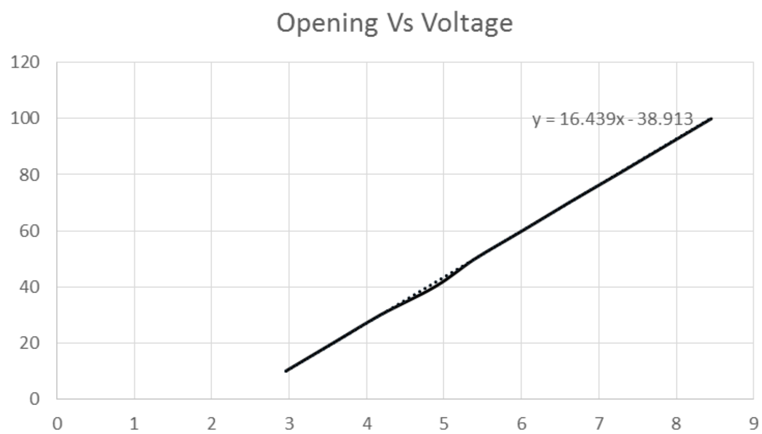
Valve Position (%)	Voltage
100	4.82
90	4.33
80	3.85
70	3.37
60	2.89
50	2.4
40	1.92
30	1.436
20	0.955
10	0.473
0	0

Figure 4.14 Pre-Heat valve position vs voltage as seen by the wireless system.

$$P_{10} = \left(\frac{T_{10}}{4096}\right) \times 100$$

The preheat valve is pneumatically controlled. For safety reasons the Pre-Heat valve fails open, which means that there is contact pressure on the valve to keep it closed. When the controller calls for heat, the pressure on the valve will decrease which allows the valve to open. For the reasons listed above, a signal of 100% means that the valve is fully closed and a 0% signal means that the valve is fully open.

Cooling Valve



Cooling Valve (%)	Voltage
100	8.45
90	7.84
80	7.23
70	6.61
60	6.01
50	5.4
40	4.88
30	4.18
20	3.57
10	2.96
0	2.35

Figure 4.15 Cooling valve position vs voltage as seen by the wireless system

Figure 4.14 and 4.15 show the equations used to calculate the percentage of valve open or closed in both the heating coil and the cooling coil.

$$P_{11} = 16.439 \times \left(\frac{T_{11}}{4096}\right) - 39.913$$

The cooling valve is also pneumatically controlled. For safety reasons the cooling valve fails closed, which means that the valve is always closed and will open only when air pressure is applied to it. A 100% signal means that the valve is fully open and a 0% signal means that the valve is fully closed.

Supply Fan Speed

$$P_{12} = \left(\frac{T_{12}}{4096}\right) \times 100$$

Supply fan speed for the fan is gotten from the variable speed drive responsible to maintaining a 1.2 IWC in the supply air duct.

Relief Fan Speed

$$P_{13} = \left(\frac{T_{13}}{4096}\right) \times 100$$

The purpose of the relief fan speed is to make sure that the pressure in the rooms do not exceed the plenum pressure. This is being done to make sure that no fumes from the labs get mixed with the return air to the unit.

Outside temperature

$$P_{14} = \left(\frac{T_{14}}{4096}\right) \times 100$$

Outside temperature is being measured in three places on-campus and being averaged and sent to all units on-campus.

Fan Current

$$P_{15} = \left(\frac{T_{15}}{4096}\right) \times 24$$

An Onset current transducer was clamped on one phase of the fan motor to record energy usage of the fan.

4.7 Matlab Code

The Matlab code can be broken into three parts. Part one transforms the twelve bit numbers into the desired units as discussed in Section 4.6. Part two applies a filter to eliminate any outliers and performs the necessary calculations to predict the performance. Part 3 compares theoretical results with measure parameters and plots the findings. Figure 4.16 shows a block diagram of the Matlab code and the calculation flow.

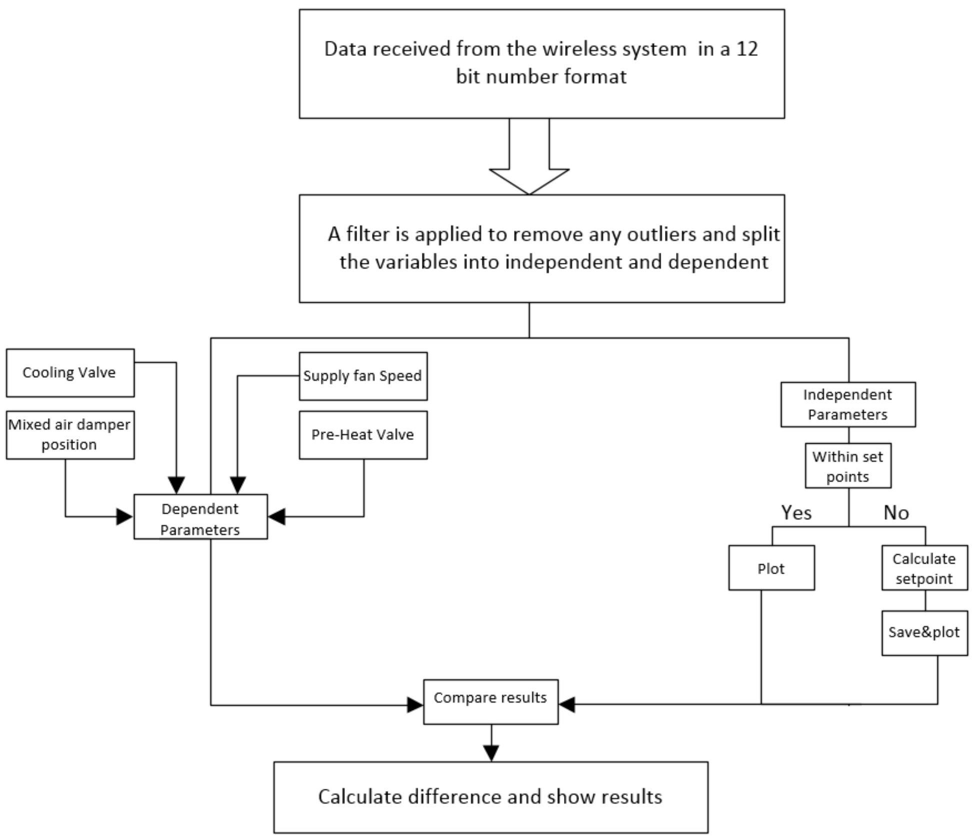


Figure 4.16 Matlab Code block diagram

The wireless system uploads information to the server every hour with recordings every two minutes. This information is saved in a comma separated variables file (CSV) and stored in a directory where Matlab is capable of accessing it. Matlab them scans the directory every hour to check for any new files, the new file is then opened, read, filtered, and converted into appropriate units. Tables 4.3 and 4.5 show the before code and after code format respectively of a six variable sample.

Table 4.3 Original data received from the wireless system

Register	139	140	141	142	143	144
41716.87639	2352	2208	32	0	308	384
41716.87778	2352	2208	32	0	308	384
41716.87917	2352	2208	32	0	308	384
41716.88056	2352	2252	28	8	308	452
41716.88194	2348	2236	32	0	304	468
41716.88333	2352	2244	32	4	304	472
41716.88472	2348	2256	36	4	308	468
41716.88611	2348	2256	36	4	308	468
41716.8875	2348	2260	32	0	312	384
41716.88889	2352	2272	32	0	308	404
41716.89028	2356	2272	32	4	312	456
41716.89167	2360	2284	32	4	316	452
41716.89306	2356	2284	32	4	308	452

Table 4.4 Filtered and converted data after going through the Matlab code

Time	Pre-Heat Temperature	Supply Air temperature	Supply Air Static Pressure	Building Pressure	Outdoor Air Flow	Exhaust Air Flow
3/18/2014 21:02	57.48	53.96	0.87	0.00	6581.51	1642.23
3/18/2014 21:04	57.48	53.96	0.87	0.00	6581.51	1642.23
3/18/2014 21:06	57.48	53.96	0.87	0.00	6581.51	1642.23
3/18/2014 21:08	57.48	55.03	0.76	0.20	6581.51	1933.04
3/18/2014 21:10	57.38	54.64	0.87	0.00	6496.03	2001.47
3/18/2014 21:12	57.48	54.84	0.87	0.10	6496.03	2018.57
3/18/2014 21:14	57.38	55.13	0.98	0.10	6581.51	2001.47
3/18/2014 21:16	57.38	55.13	0.98	0.10	6581.51	2001.47
3/18/2014 21:18	57.38	55.23	0.87	0.00	6666.98	1642.23
3/18/2014 21:20	57.48	55.52	0.87	0.00	6581.51	1727.76
3/18/2014 21:22	57.58	55.52	0.87	0.10	6666.98	1950.15
3/18/2014 21:24	57.67	55.82	0.87	0.10	6752.45	1933.04
3/18/2014 21:26	57.58	55.82	0.87	0.10	6581.51	1933.04

Matlab then plots all the parameters on the same figure with multiple graphs as shown in Figure 4.17

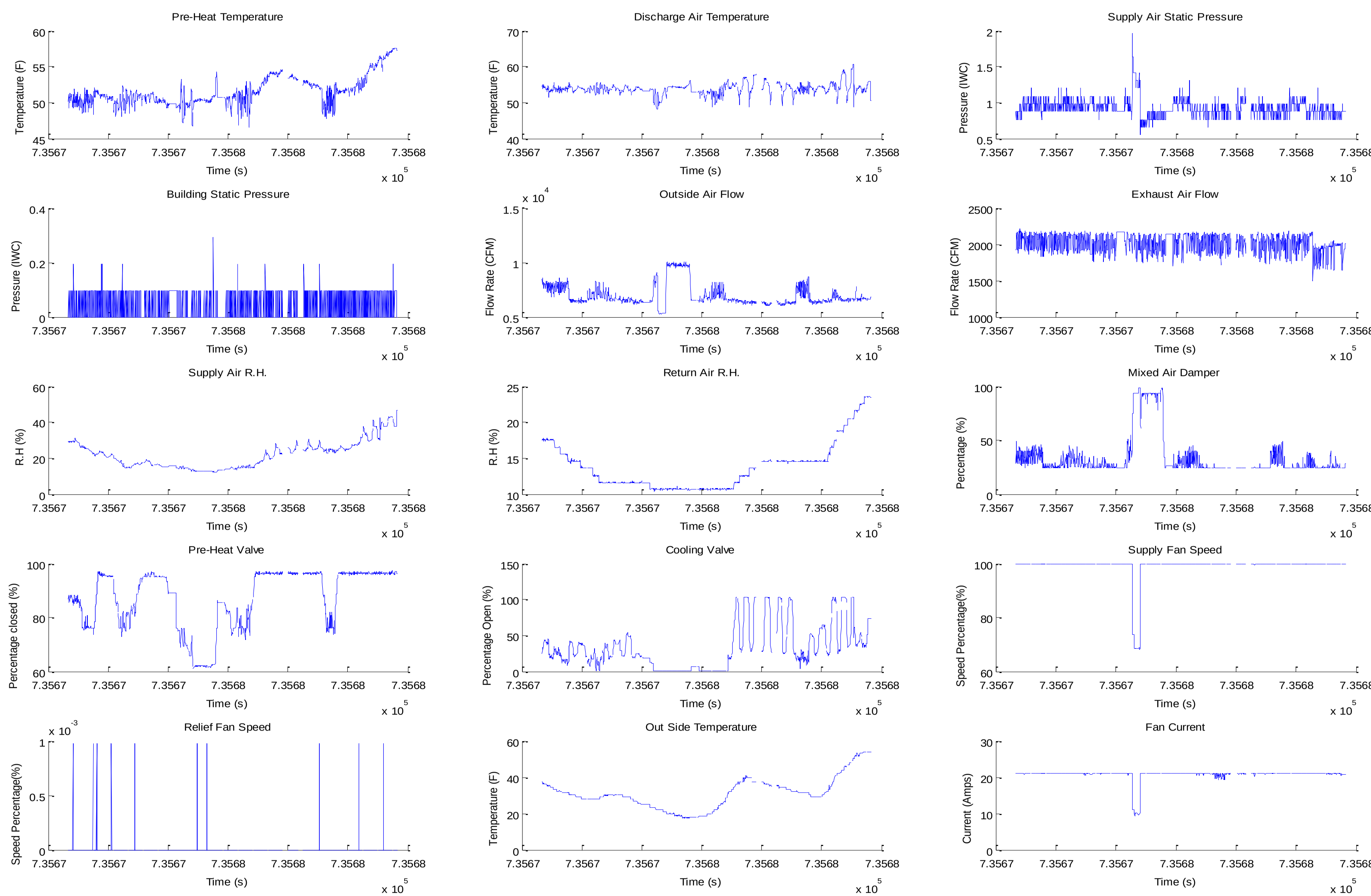


Figure 4.17 All monitored parameters displayed in separate graph

In addition to plotting the results in a graphs, Matlab is also saving all the data into an excel file. Each month the program will create a new excel file, each excel file will have 30 different sheets representing each day of the month. This is to allow the user to control monthly and daily monitoring of the unit.

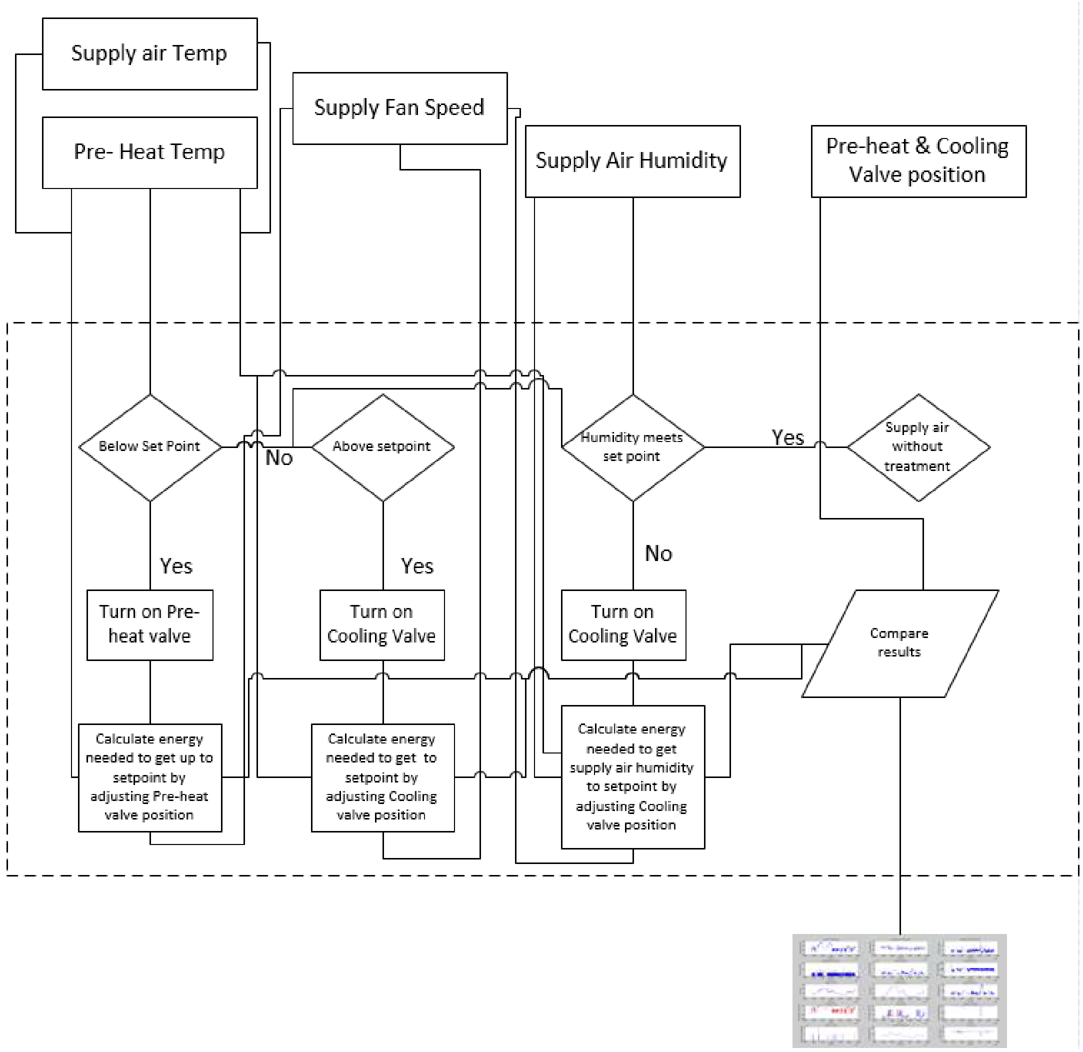


Figure 4.18 MatLab Code block diagram

5. MATHEMATICAL MODEL VALIDATION AND RESULTS

In this chapter it was shown how the mechanical model was validated by comparing its results to the actual operation of the unit. In addition, this chapter shows the results obtained from the mechanical model and discusses how they compared to these from the actual operation of the unit.

The mechanical model's aim was to predict the operation of the Pre-heat coil in addition to the cooling coil. By using the equations and control sequence presented in Chapter 3 the following results were attained. The mechanical model was applied for a monitoring period, during this monitoring period, outside temperatures fluctuated between 0°F and 80 °F.

5.1 Validation of the Mechanical Model

In order to make sure the results obtained from the mechanical model could accurately predict the performance of the unit, a validation system was put in place that allowed for comparing theoretical results with actual operating conditions. Based on that analysis, a baseline error report was established to indicate the accuracy of the mechanical model.

5.1.1 Pre-Heat Valve Validation

In order to validate the Pre-Heat valve operation, data during low outside temperature fluctuations was collected for six hours and then compared to the mechanical model. Results are presented below,

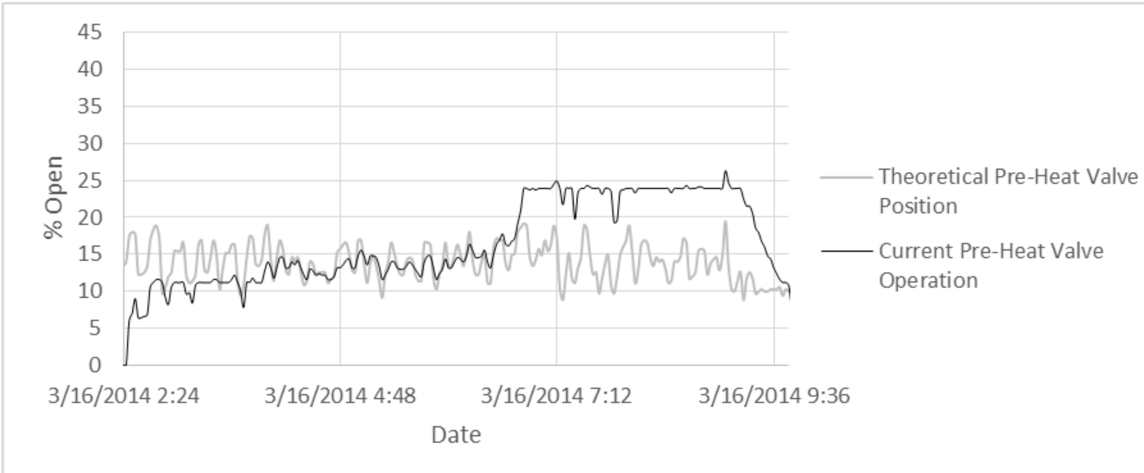


Figure 5.1 Actual Pre-Heat valve compared to theoretical results for validation purposes

Figure 5.1 shows a sampled result that compares the actual operation to the theoretical operation in order to validate the theoretical model. Table 5.1 shows the different error percentage associated with the validation process.

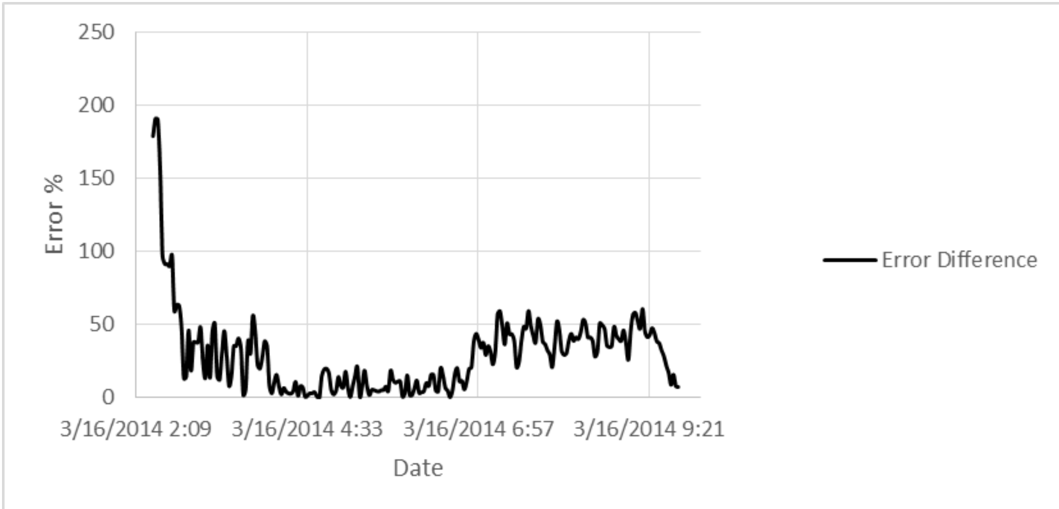


Figure 5.2 Difference graph between measured valve position and theoretical position

Figure 5.2 and Table 5.1 show the graph and values for the difference error respectively. With an average error of 29.85%, minimum error of 0% and a maximum error of 191.17%. This difference error is partly due to the estimations of the variables in the mechanical model that introduce an uncertainty to the total system calculation. However, the mechanical model was able of identifying potential energy savings of over 80% of the total energy used at some periods, therefore even with applying the error calculation there will still be an average of 50% energy savings.

Table 5.1 Error data results for comparing theoretical Pre-Heat valve position with actual valve position

% Difference	
Average	29.85
Minimum	0.07
Maximum	191.17

5.1.2 Cooling Valve Validation

The next step was to verify the results from the cooling valve operation. In order to do that, the actual operation from the AHU was collected and compared to the theoretical results obtained from the mechanical model.

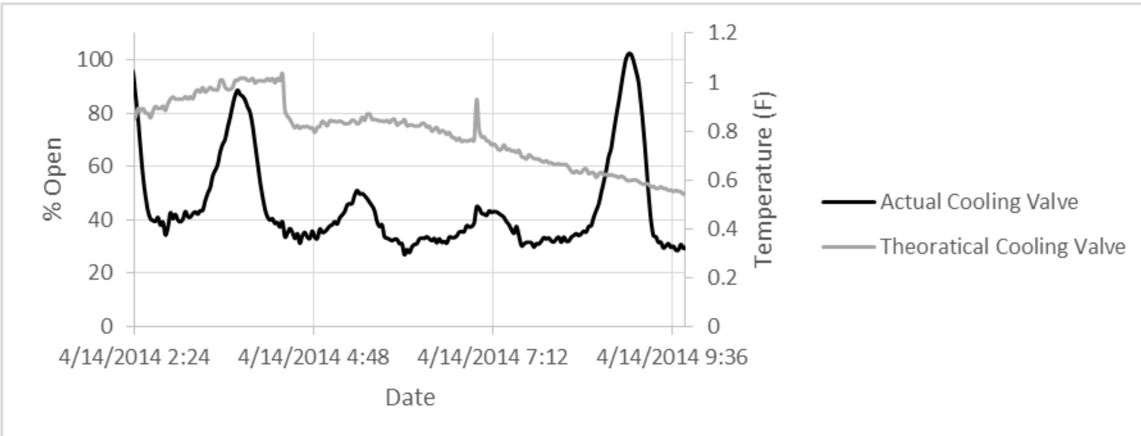


Figure 5.3 Actual Cooling valve compared to theoretical results for validation purposes

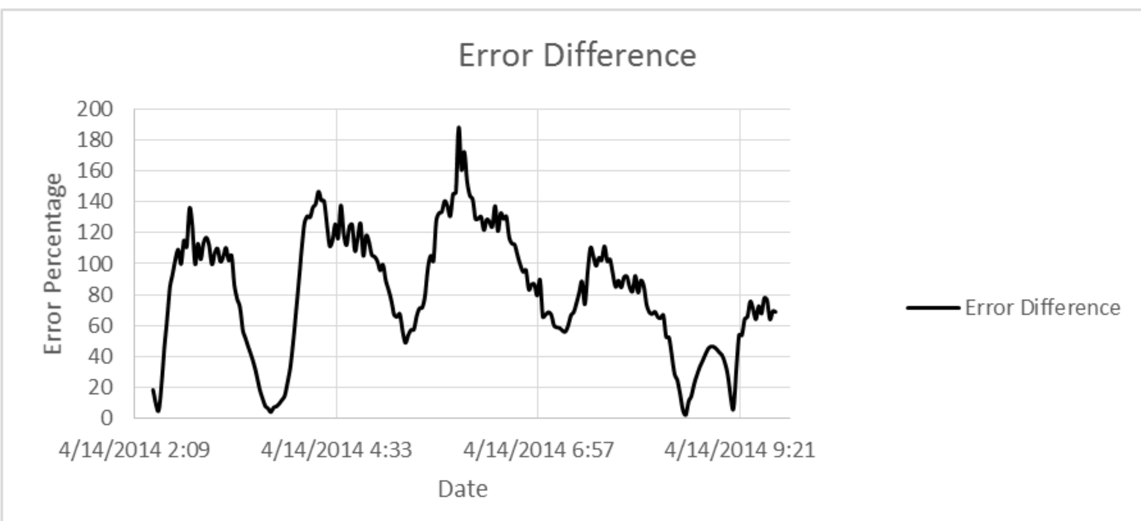


Figure 5.4 Difference graph between measured valve position and theoretical position

Table 5.2 Error data results for comparing theoretical cooling valve position with actual position

% Difference	
Average	82.71
Minimum	2.24
Maximum	188.05

Based on Figure 5.4 and Table 5.2 a general observation was made. The actual cooling valve was less sensitive to fluctuations in outside temperatures than the mechanical model. With an average fluctuation of 82%, a minimum of 2% and a maximum of 188%, a difference curve could be expected in the presented results of the cooling valve.

5.2 Mechanical Model Results and Waste Calculation

5.2.1 Performance of AHU During 1/8/2014-2/2/2014

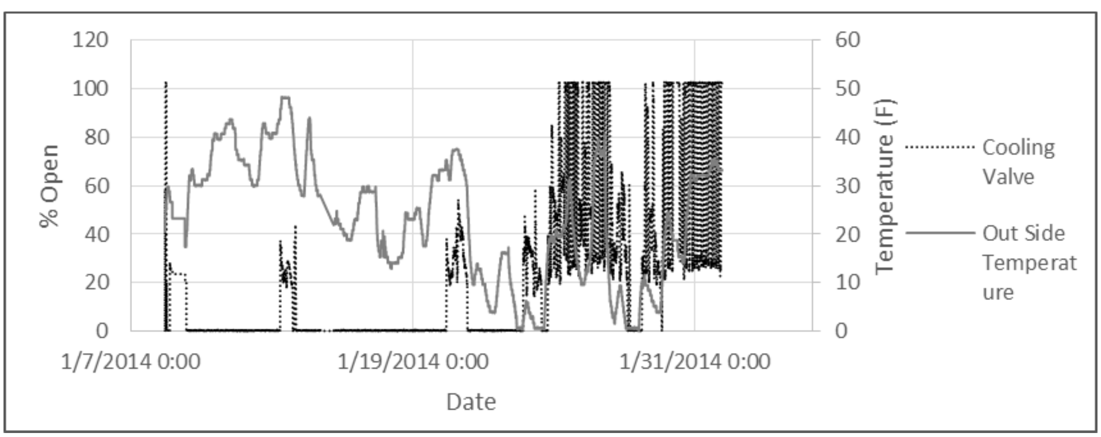


Figure 5.5 Cooling valve position in comparison with outside temperatures

By simply looking at Figure 5.5, a question remained. Why the cooling valve was operating while the outside temperature was below 50°F? In such cases, if the unit called for cooling, the unit should utilize the concept of free cooling or economizing, where the unit used outside colder air to cool the return air and bring it down to set point. In addition the AHU could be dehumidifying the supply air by removing humidity using the cooling coil. Figure 5.5 shows monitored data between January 8th 2014 and February 1st 2014. During that period outside temperatures went below 0°F, however the wireless system was not capable of recording negative voltages therefore the reading on the graph were shown as 0°F. Figure 5.5 represents the performance of the unit in the coldest time of the year.

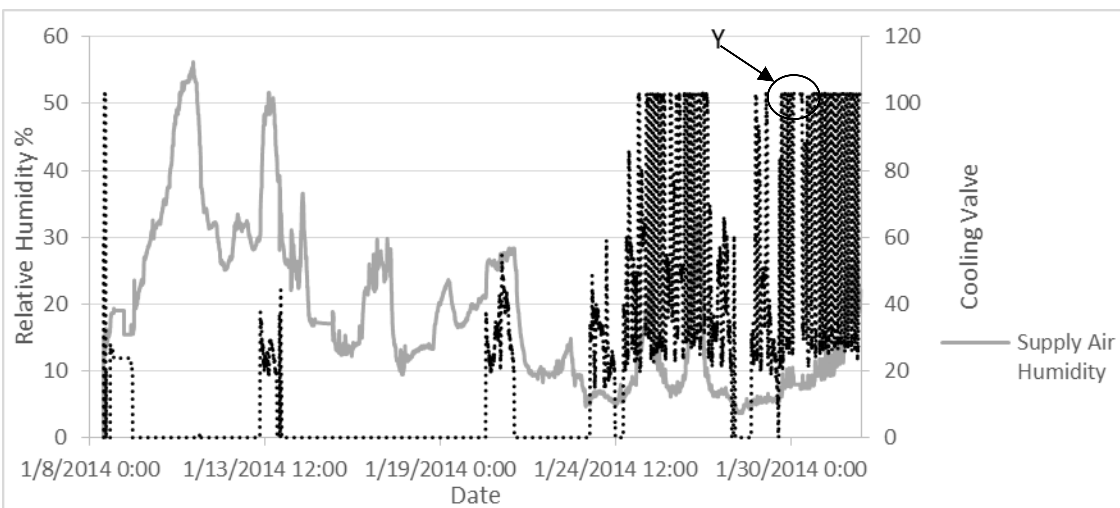


Figure 5.6 Cooling valve current operation in comparison with supply air humidity between (1/28/2014-2/2/2014)

Figure 5.6 shows the valve position in comparison to supply air relative humidity. Relative supply air humidity at some of the times did not exceed 25%, yet the cooling valve was operating at full capacity at different stages. According to R. Janis[8] the lower comfort limit in cold weather is 68 °F at about 30% relative humidity (RH). Therefore, based on this given, there was no logical explanation of the operation of the cooling valve under those conditions. Figure 5.7 shows a zoomed in section of Figure 5.6, each spike in the cooling valve operation is about one to two hours at that operating condition.

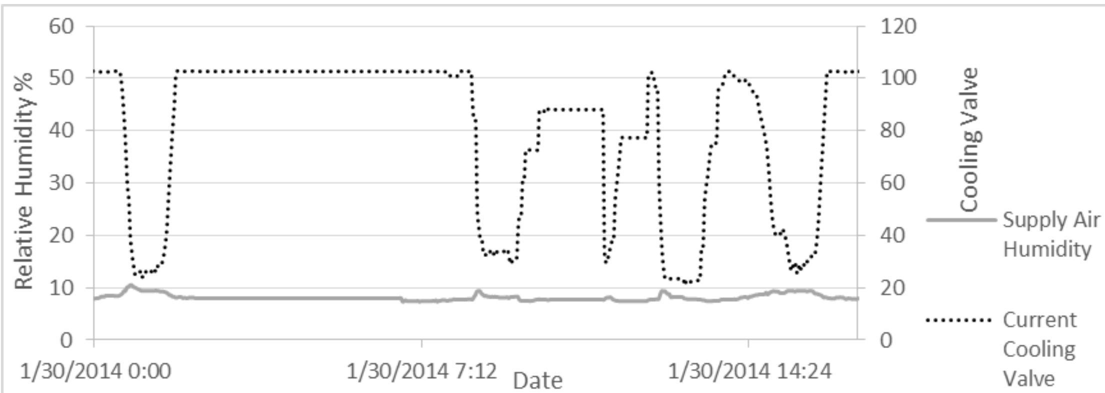


Figure 5.7 Zoomed in Section (Y) of the cooling valve showing the valve being open for around an hour at a time.

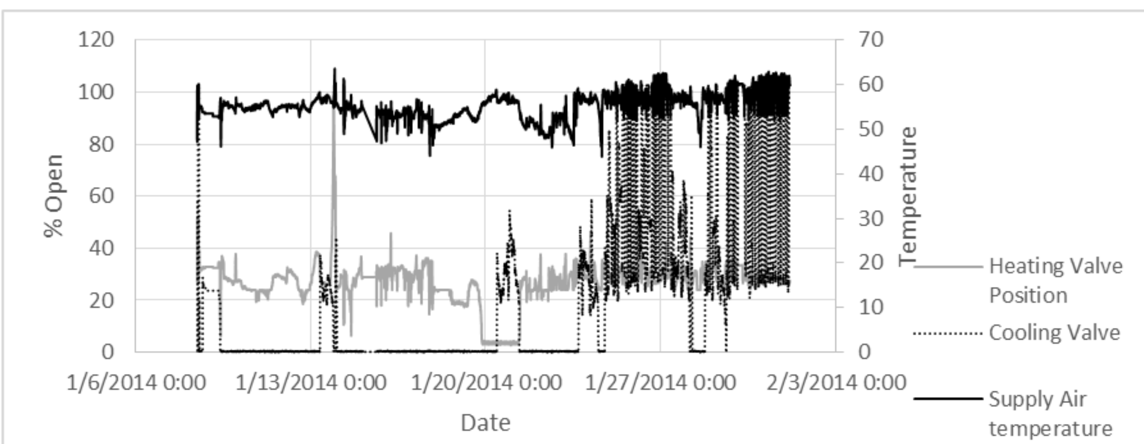


Figure 5.8 Actual cooling valve and heating valve position in comparison with supply air temperature

By comparing actual (Cooling/Heating) valve position with the Pre-Heat temperature in Figure 5.8 it was clear that the cooling valve trying to reduce the supply air temperature to a set point below 60°F. However it was important to note that the supply air temperature was above the recommended set point of 55°F , in fact this was a decision made by the building engineers to increase the supply air set point at lower outside temperatures to be able to maintain comfortable air conditions in the offices and labs. That being said, this decision did not explain the operation of the cooling valve. The only explanation to this operation was that the cooling valve was calling for cool to maintain the 55°F set point, however the heating valve was negating this call and applying heat to the supply air. Raising it to the new set point at 60 °F.

5.2.2 Mathematical Model Results

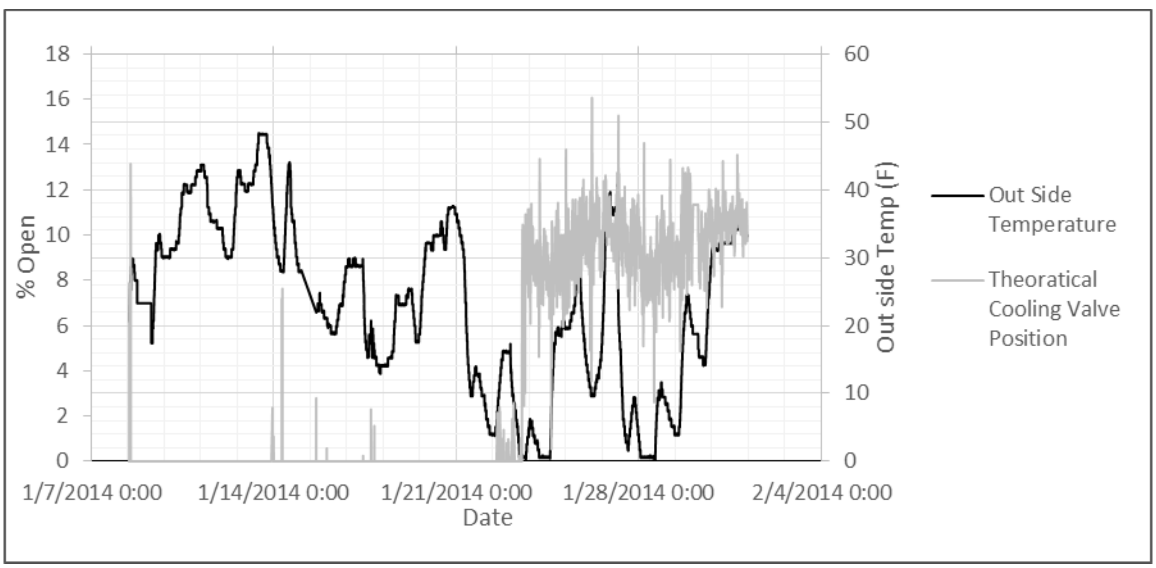


Figure 5.9 Cooling valve theoretical position compared to low outside temperatures

Figure 5.9 shows the theoretical position of the cooling valve in comparison with outside temperature. Even though outside temperature was lower than 55°F it was predicted that the valve would be open after the 24th of January. To be able to understand those results the pre-heat temperature inside the unit should be analyzed and understood since this was where the control system received its command.

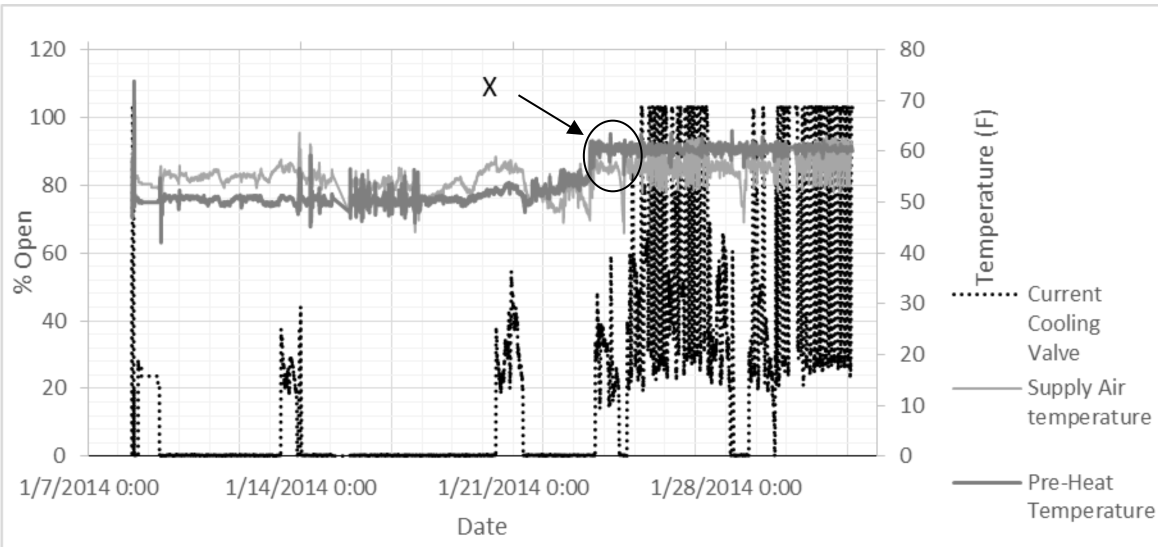


Figure 5.10 Theoretical cooling valve position in comparison with supply air temp and pre-heat temp

With a new set point of 60 °F , seen at X in Figure 5.10, it was clear that the subsystems in the unit was contradicting each other and the unit was confused about the desired supply air temperature. This was noticed because of the difference between the preheat temperature and the supply air temperature. A spike in Pre-heat Temperature happened at X, this was due to the increase in the supply air set point and the VAV air set point. However it seemed that this increase was not conveyed to the cooling coil controller and therefore the cooling coil was still struggling to bring supply air temperature to 55°F. This operation resulted in wasted resources that were calculated in the theoretical model.

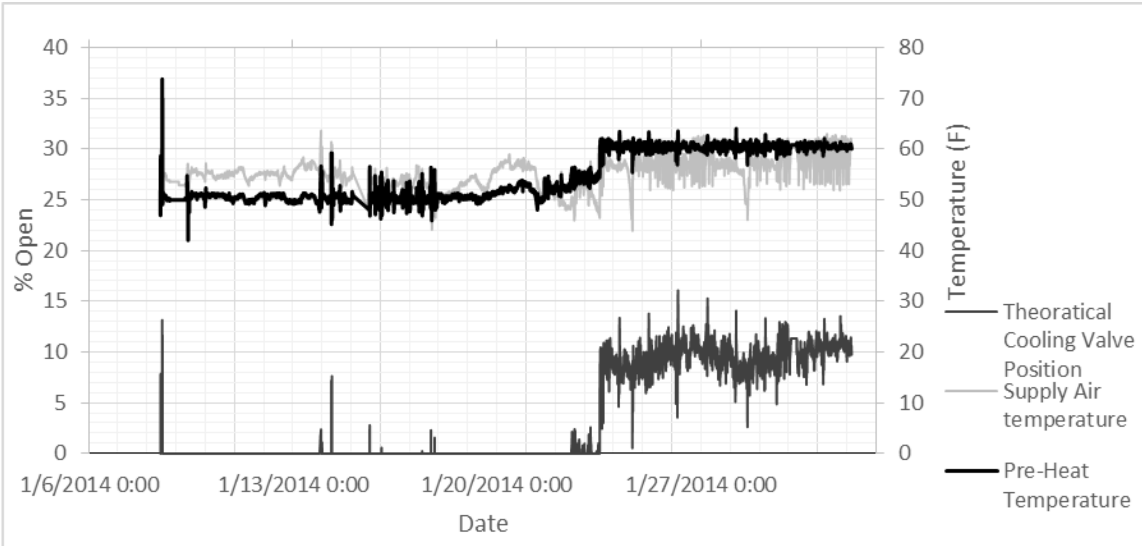


Figure 5.11 Theoretical valve position in comparison with supply air temperature and pre-heat temperature

Figure 5.11 shows the theoretical valve operation in comparison to the supply air temperature. As supply air temperature increases more cooling is demanded and the cooling valve is commanded to open more. Moreover, the actual operation of the cooling valve suggests that the actual valve was operating at 100% open position rather than what is suggested in the theoretical model. This data was calculated considering a supply air temperature set point of 65°F . Analyzing this data resulted in the calculation of the waste of chilled water.

5.2.3 Comparison of Theoretical with Actual Results

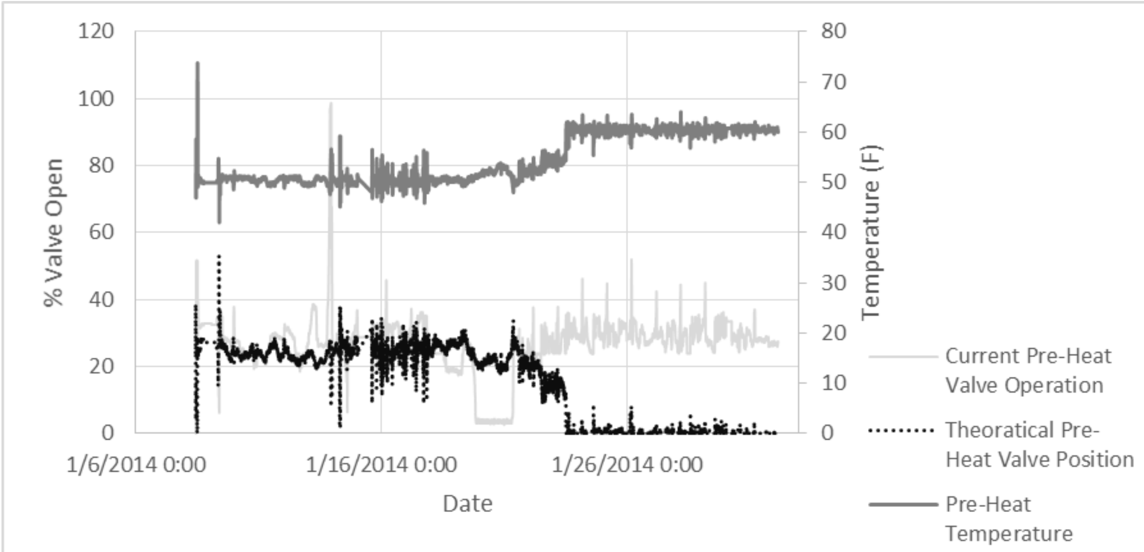


Figure 5.12 Theoretical and actual Pre-heat valve position in comparison with Pre-heat Temperature

Figure 5.12 shows the expected performance of the Pre-heat coil in comparison with Pre-Heat temperature. As seen in the theoretical model, the heating valve was not in operation whenever pre-heat temperature was above 55°F.

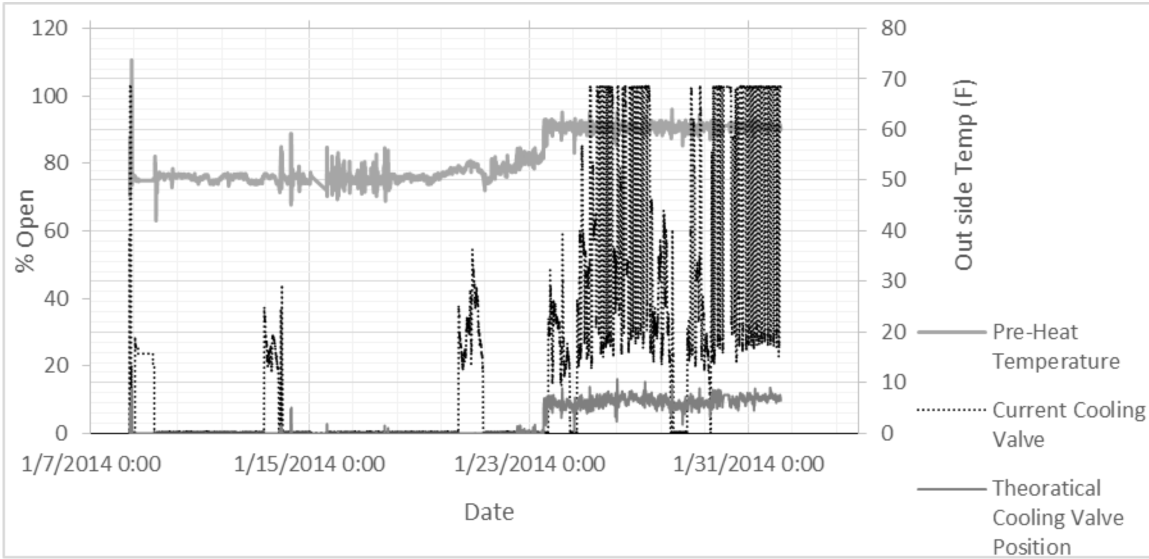


Figure 5.13 Theoretical and actual Cooling valve position in comparison with Pre-Heat Temperature

Figure 5.13 shows the theoretical and actual position of the cooling valve in comparison with the Pre-Heat temperature. As noted before the cooling valve was fully operational where theoretically it should not exceed 10% open position.

From Figures 5.12 and 5.13 the wasted energy was calculated through a comparison between theoretical and actual valve position.

5.2.4 Chilled Water Waste

By comparing the current operation to the theoretical operation of the valve it was possible to calculate the wasted flow in gallons of chilled water and convert that to a kWh measurement. This measurement would only consider the amount of energy needed to heat/cool water to reach a ΔT that was lost through the coils. Energy Cost was estimated at \$0.09/kWh based on CFS statistics.

Table 5.3 Current, theoretical and waste gallons used in the cooling coil in low temperatures

Cooling Coil				
Gal used			Energy waste (kWh)	Cost Savings (\$)
Current	Theoretical	Waste		
1,887,683	30,337	1,857,346	45,411	\$3,954

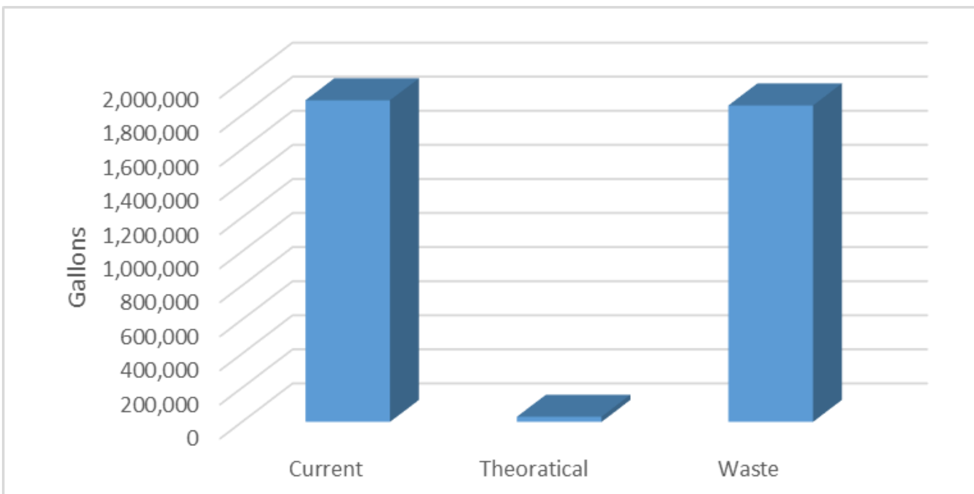


Figure 5.14 Gallons of water used between 1/8/2014-2/1/2014 and difference between theoretical and current

Finally for the low temperature analysis it was important to compare the results of the theoretical usage of the heating coil to actual usage. Keeping in mind that the supply air temperature should not exceed 55°F at anytime. Therefore theoretical set point was always set at 55 °F rather than 60°F.

5.2.5 Heated Water Waste

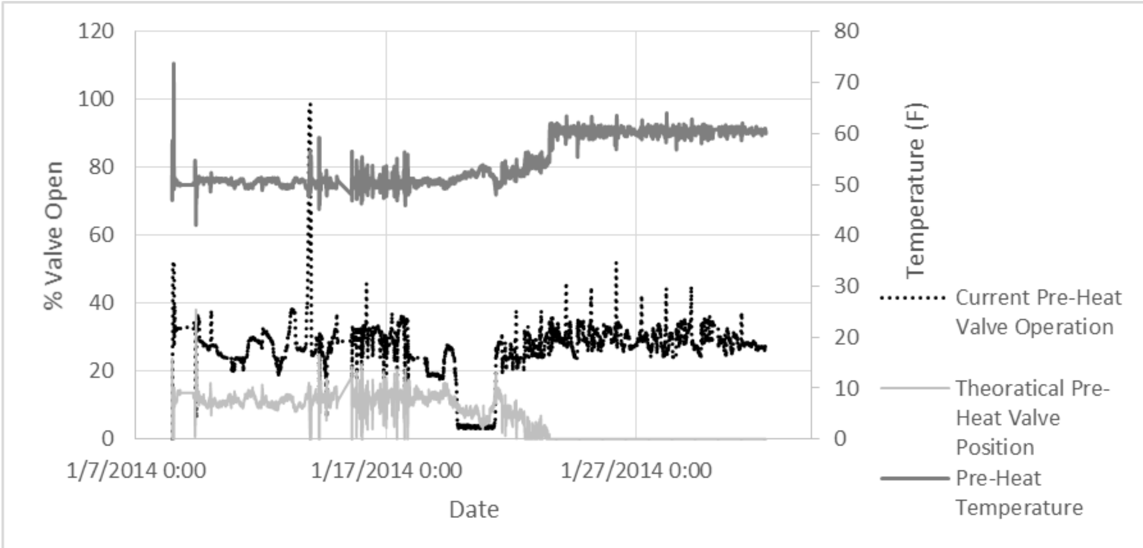


Figure 5.15 Heating valve actual performance in comparison to theoretical performance with a Pre-Heat Temperature reference

As seen in Figure 5.15 the heating valve kicked on even though the pre-heat temperature was above 55°F. Table 6 and Figure 5.15 shows the waste of water during that period in gallons of water at 200°F.

Table 5.4 Current, theoretical and waste gallons used in the Pre-Heat coil 1/8/2014-2/1/2014

Heating Coil				
Gal used			Energy wasted (kWh)	Cost Savings ($)
Current	Theoretical	Waste		
560,461	306,214	254,247	18,044	$1,571

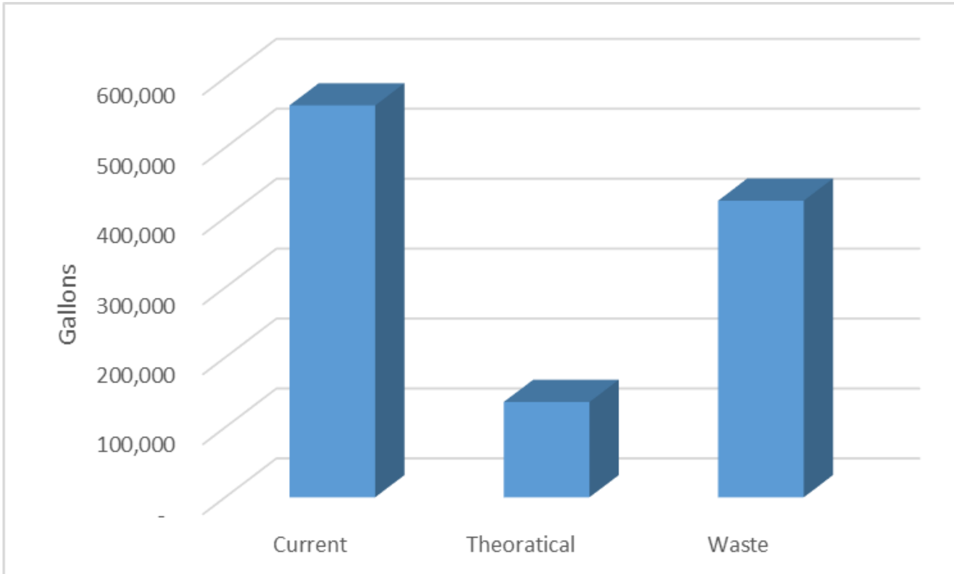


Figure 5.16 Gallons of water used between 1/8/2014-2/1/2014 and difference between theoretical and current

5.3 Mechanical Model Results Analysis Between 4/1/2014-4/30/2014

5.3.1 Performance of AHU in April of 2014

Data was collected between 4/1/2014 and 4/30/2014. This data was especially useful because of the high outside temperatures reached during that period. Therefore, a comprehensive analysis of the performance of the AHU allowed for a validation of the mathematical model and also allowed for an additional analysis of the control system.

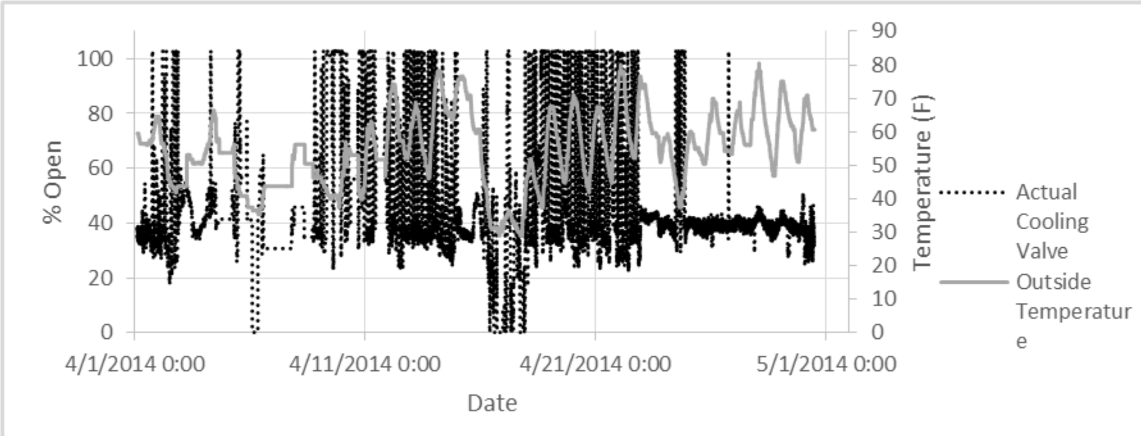


Figure 5.17 Cooling valve position compared to outside temperature

Figure 5.17 shows the current performance of the cooling valve compared in mild outside temperatures. As seen in the graph the cooling valve followed the trend of the outside temperature curve.

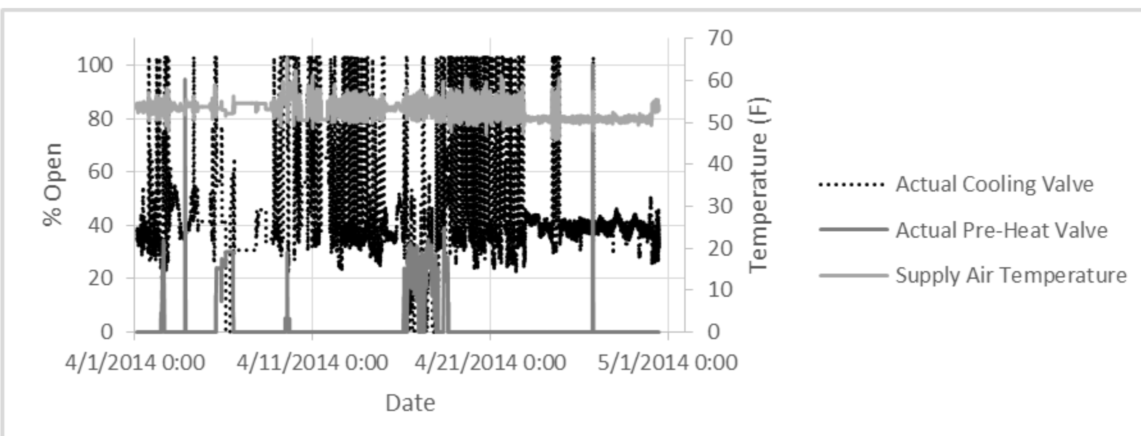


Figure 5.18 Actual cooling valve and Pre-Heating valve position in comparison with supply air temperature

Figure 5.18 shows a proper performance of the cooling valve and pre-heat valve. When the cooling valve was on, the heating valve was off and vice versa.

It was important in this case to study the supply air relative humidity shown in Figure 5.19. The study of relative humidity was able to give us a better understanding of the operation of the cooling coil.

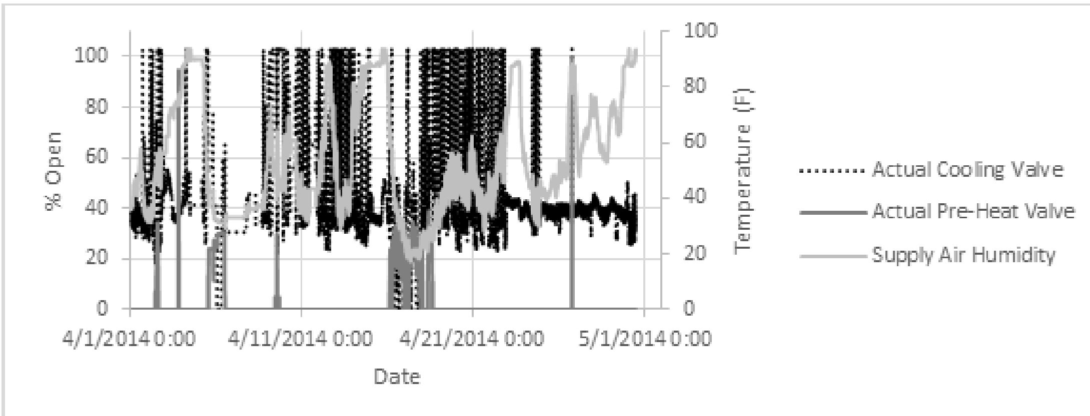


Figure 5.19 Relationship between Cooling valve, Pre-heat valve, and supply air relative humidity

Supply air relative humidity (RH) presented in Figure 5.19 shows that RH exceeded the desired set point, therefore the cooling coil was commanded to reduce this value to the desired set point. Therefore in addition to temperature control the cooling coil was engaged for humidity control as well.

5.3.2 Mathematical Model Results

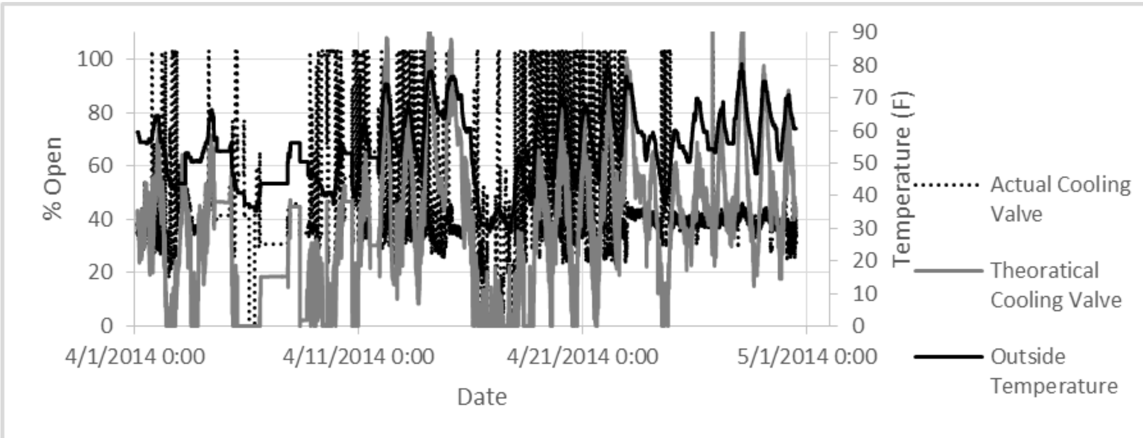


Figure 5.20 Cooling Valve theoretical and actual position compared to outside temperature

Figure 5.20 shows the theoretical position of the cooling valve with respect to outside temperature with a supply air set point of 55°F. As expected, as outside air temperature increased more chilled water was needed to reduce the supply air temperature to its set point. The mechanical model results almost always followed outside temperature fluctuations, however the actual cooling coil was operating in a different way. In contrast, Figure 5.21 shows the operation of the Pre-Heat valve, which was operating when outside temperatures were lower than 45°F.

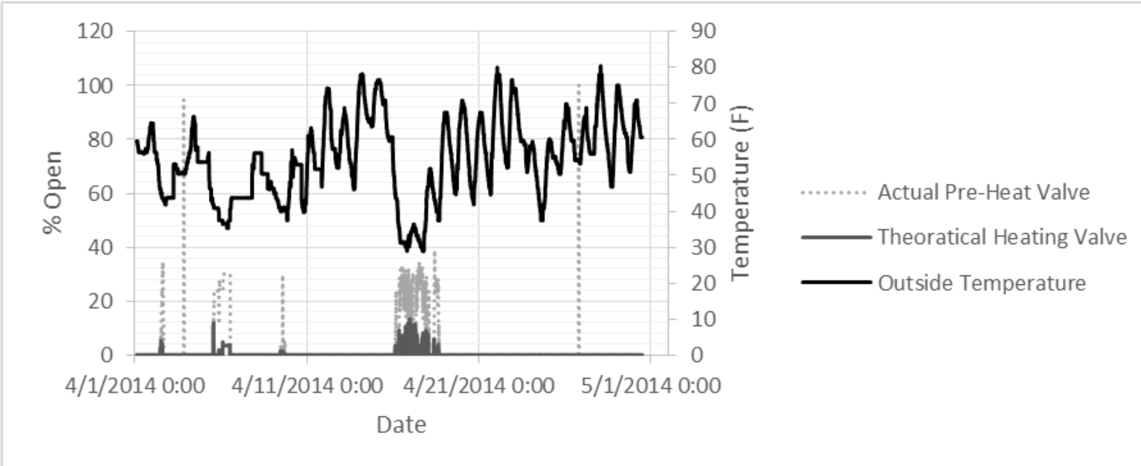


Figure 5.21 Pre-Heat valve theoretical and actual position compared to mild outside temperatures

5.3.3 Chilled Water Waste

By comparing the current operation to the theoretical operation of the valve it was possible to calculate the wasted flow in gallons of chilled water and convert that to a kWh measurement. This measurement only considered the amount of energy needed to heat/cool water to reach a ΔT that was lost through the coils.

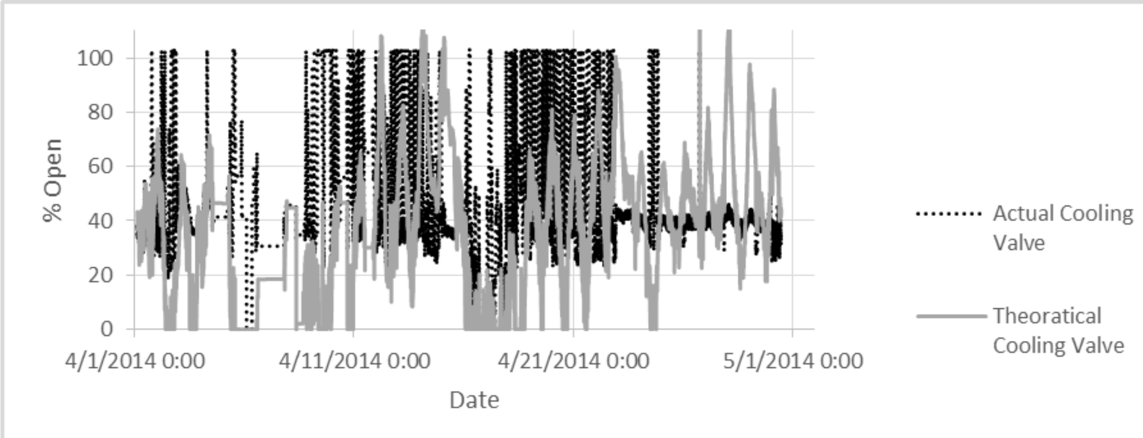


Figure 5.22 Theoretical and actual cooling valve position

Table 5.5 Current, theoretical and waste gallons used in the cooling coil in mild temperatures

Cooling Coil				
Gal used			Energy (kWh)	Cost Savings ($)
Current	Theoretical	Waste		
5,090,005	1,934,872	3,155,132	77,141	$6,942.73

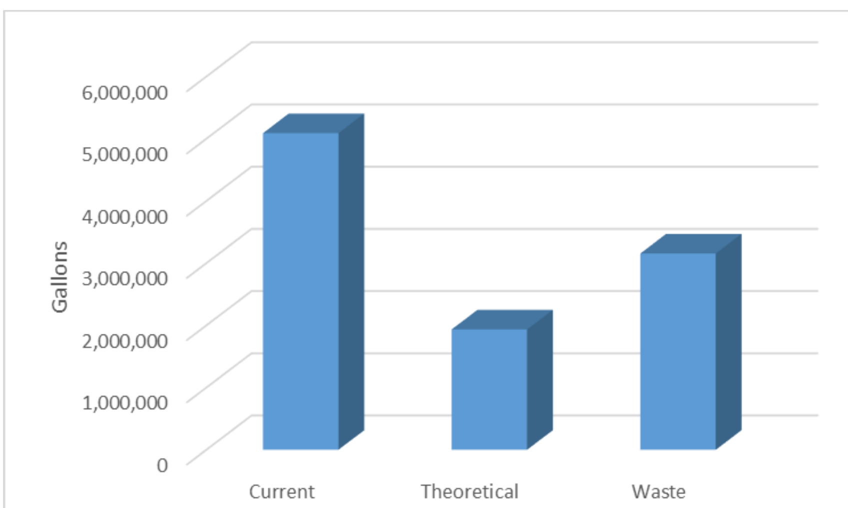


Figure 5.23 Gallons of water used between 4/1/2014 and 4/30/2014 and difference between theoretical and current

A total of 3,155,132 gallons of water was the difference between the actual usage and the theoretical operation of the cooling coil.

5.3.4 Heated Water Waste

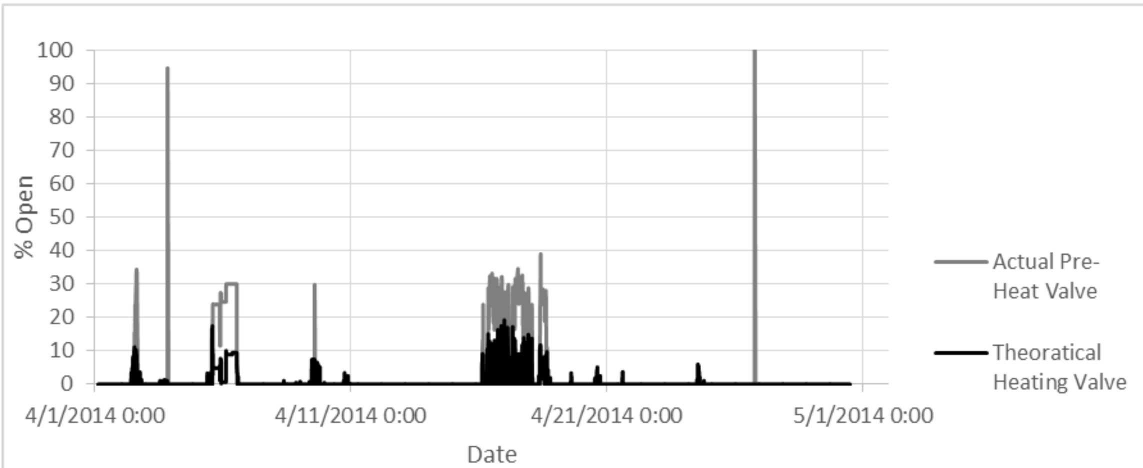


Figure 5.24 Theoretical and actual Pre-Heat valve position

Table 5.6 Current, theoretical and waste gallons used in the Pre-Heat coil

Heating Coil				
Gal used			Energy	Cost Savings ($)
Current	Theoretical	Waste	kWh	
27,549	2,917	24,632	602	$54

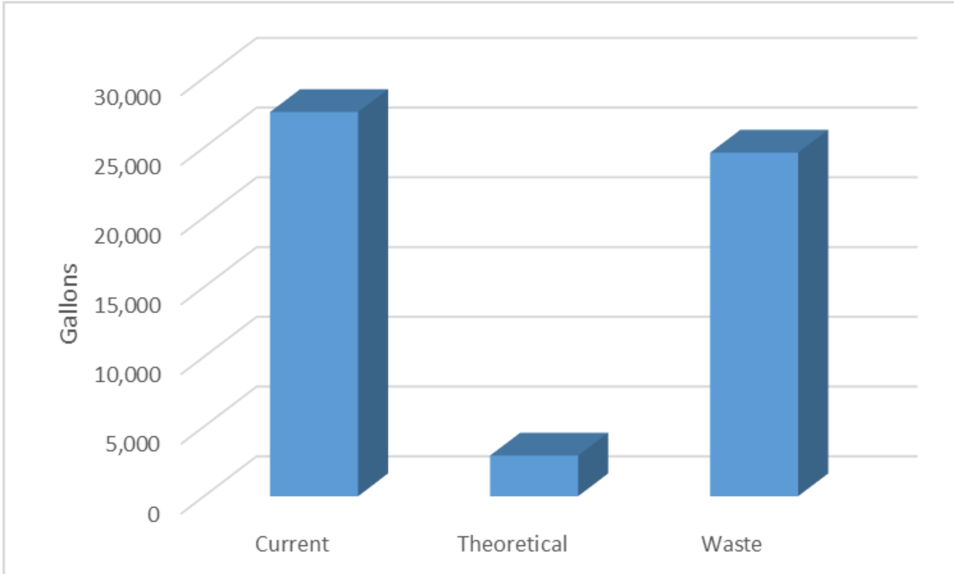


Figure 5.25 Gallons of water used between 4/1/2014 and 4/30/2014 and difference between theoretical and current

5.4 Set Point Analysis

After examining the operation of the AHU, it was decided to perform a supply air temperature set point analysis. The purpose of this analysis was to study the effects of a set point change on the energy consumption of the unit. Performing this analysis, it was clear that a single set point cannot be generalized for all operating condition, but it was a function of outside temperatures and building occupants. Building engineers manually changed supply air set points to maintain occupant comfort in the classrooms, offices, and labs.

This analysis was split into three monitored months, January, March, and April. The analysis used the mechanical model to simulate a temperature set point change with increments of 1°F starting at 45°F and ending at 65°F. It is important to note that the following analysis only applies to the operating conditions of SL 2E unit and cannot be applied to any other units with different operating conditions.

January 2014

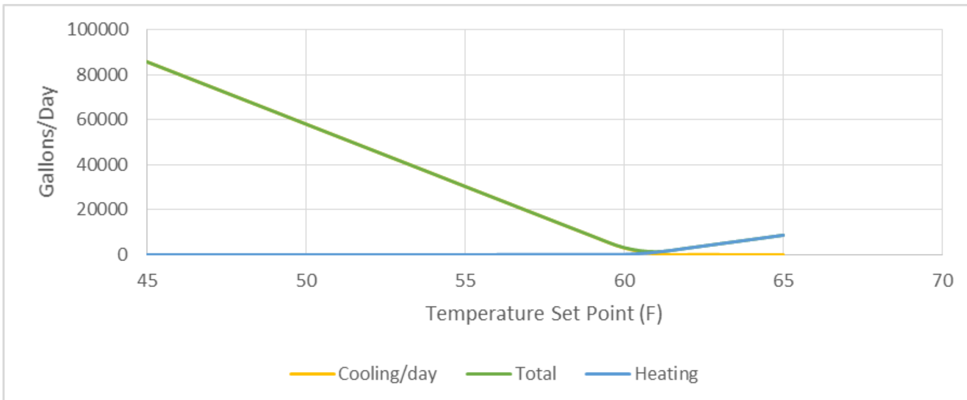


Figure 5.26 Effect of set point change on the gallons of water used in both the cooling and Pre-Heat coil during January 2014

Based on the Set Point analysis for January 2014, a supply air set point of 61°F would result in the minimum energy usage per day compared to other set points.

March 2014

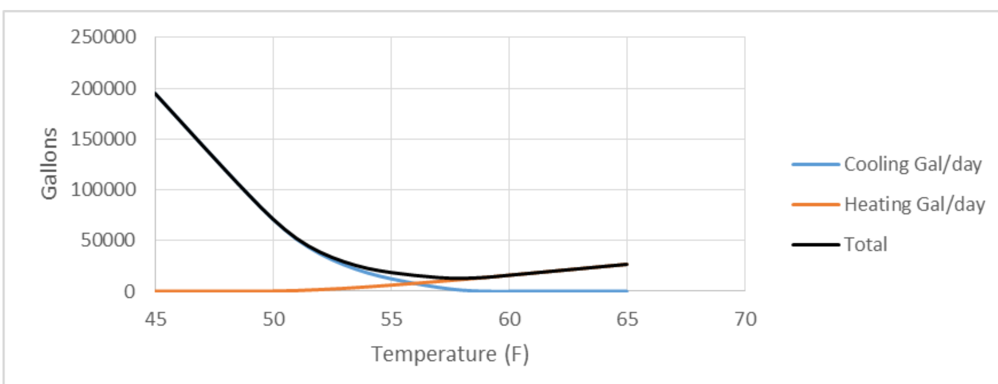


Figure 5.27 Effect of set point change on the gallons of water used in both the cooling and Pre-Heat coil during March 2014

The set point analysis done for March 2014 resulted in identifying a set point where the minimum energy could be used to maintain comfort levels. According to the analysis a set point of 58°F in March would result in the minimum energy usage per day for that specific month.

April 2014

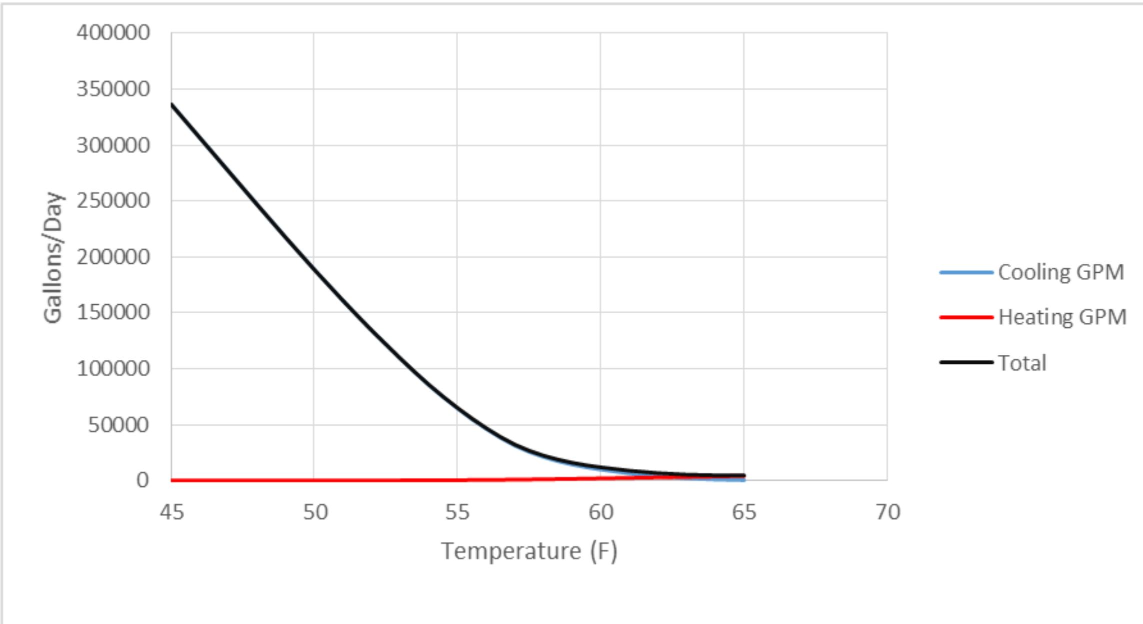


Figure 5.28 Effect of set point change on the gallons of water used in both the cooling and Pre-Heat coil during April 2014

As seen in Figure 5.28 because of the high outside temperature a clear understanding of the set point effects on the energy usage couldn't be established without a clear relationship between the set point and comfort levels in the rooms. For the reasons listed above, more data need to be collected in order to provide an accurate value for the set point analysis.

5.5 Discussions and Interpretation

Based on the analysis done in Section 5.1 and 5.2, it was clear that monitoring unit performance could give very important insight on the operation of the unit and its energy usage. It was clear by the analysis done on January 2014 that there was energy waste due to a contradiction in operation of the cooling coil and Pre-Heat coil. This contradiction was due to a change in the supply air set temperature that resulted in the system calling for both cooling and heating.

If a mechanical model was present at that time that could alert building engineers about such operation error, the waste would have been avoided and normal operation could have been established. It was clear conducting the study that for reducing energy consumption of the AHU a set point analysis should be done to determine the tradeoff between energy consumption and creature comfort.

The set point analysis lead to a collusion that states that for optimal energy usage, a set point should be selected depending on the weather conditions in addition to occupancy of the building. Because of the limited data for this research, the set point analysis was only able to identify the optimal set points for January 2014 and March 2014.

Finally it is important to emphasize the importance of maintaining historical data that logs the performance of the unit under different circumstances. This historical data could be used for control optimization, energy usage reduction, and also could be used for future trouble shooting.

6. CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

The AHU mechanical model used in this research, gave an insight of the operation of an AHU and helped identify different factors that affect the energy usage in the unit. The mechanical model gave the ability to log and analyze historical data that represented the main energy consumers in the unit.

A mathematical model was manipulated in order to accept already measured data for controls and troubleshooting purposes. This meant that using a mechanical model like one used in this research could reduce capital cost needed to monitor all necessary parameters.

A mechanical model was established for both the Pre-Heat coil and the cooling coil separately, but connected them through a controls algorithm based on the desired set points. This gave enough flexibility that allowed to analyze each coil separately of combine their operation and compare it to the current operation.

The model developed in this paper will allow for a clear representation of each component in the AHU, which neither the Black box method nor the grey box method is capable of doing.

Finally, this model along with the historical data will allow for energy calculation, controls verification, and overall performance of an AHU unit.

6.2 Recommendations for Future

Some work can be done in the future on the current mathematical model to improve its accuracy.

- Measure all the estimated values on the coils to ensure no error is resulting from this estimate. Those variables are, the flow rate in the coils using a flow meter, CFM seen by the coils rather than the calculating it through the fan speed, inlet and outlet temperatures on the coils.

- The operation of all VAV boxes with the operation of the reheat coils to show the relation between them and the main coils. This data should be already be controlled through the current control system, therefore no additional sensors are needed.

- To be able to accurately prove the results from the theoretical model, it is needed to use the full potentials of the FFD and use its writing ability to be able to control set points through the mathematical model. Moreover, this addition will allow for an analysis of the effects of a change of a set point on the energy usage of the unit.

LIST OF REFERENCES

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[1] B. Tashtoush, M. Molhim, and M. Al-Rousan, "Dynamic Model of an HVAC System for Control Analysis," *Energy,* Vol. 30, pp. 1729-745, 2004.

[2] A. Afram and F. Janabi-Sharifi, "Theory and Applications of HVAC Control Systems – A Review of Model Predictive Control (MPC)," *Building and Environment,* Vol. 72, pp. 343-55, 2014.

[3] P. Bhaskoro, S. H. Gilani, and M. S. Aris. "Simulation of Energy Saving Potential of a Centralized HVAC System in an Academic Building Using Adaptive Cooling Technique," *Energy Conservation Management,* Vol. 75, pp. 617-28, 2013.

[4] K.F. Fong, V.I. Hanby, and T.T. Chow, "HVAC System Optimization for Energy Management by Evolutionary Programming," *Energy and Buildings,* Vol. 38, pp. 220-231, 2006

[5] L. Perez-Lombard, J. Ortiz, and I. R. Maestre, "The Map of Energy Flow in HVAC Systems," *Applied Energy,* Vol. 88, pp. 5020-5031, 2011.

[6] K.F. Fong, V.I. Hanby, and T.T. Chow, "System Optimization for HVAC Energy Management Using the Robust Evolutionary Algorithm," *Applied Thermal Engineering,* Vol. 29, pp. 2327-334, 2009.

[7] "Modbus Organization." *Modbus FAQ*. N.p., n.d. Web. 10 Mar. 2014.

[8] "Building Sector, Energy, CO2 Emissions - Current Situation - Architecture 2030." Climate Change, Global Warming, and the Built Environment - Architecture 2030. Web. 1 Feb. 2011. www.architecture2030.org/current_situation/building_sector.html

[9] Clean Energy. Web. 2012. http://www.epa.gov/cleanenergy/energy-andyou/index.html

[10] "Fundamentals of HVAC," *2009 ASHREA Handbook Fundamentals*. Sec. 5, January 2009

[11] F. Haghighat, E. Morofsky, E. Kutrowski, and B. Coffey, "A Software framework for model predictive control with GenOpt," *Energy and Buildings*, Vol. 42, No. 7, pp. 1084-1092, 2010.

[12] S. Svendsen and S. Petersen, "Method for simulating predictive control of building systems operation in early stages of building design," *Applied Energy*, Vol. 88, No. 12, pp. 4597-4606, 2011.

[13] "Clean Energy," http://www.epa.gov/cleanenergy/energy-and-you/, January 2014.

[14] "Renewable Sources and Efficient Electrical Energy Systems," http://ecee.colorado.edu/ecen2060/efficiency.html, Jun. 2013 [Nov, 2013]

[15] "VAV diagram," http://en.wikipedia.org/wiki/Variable_air_volume, Mar. 2013 [Dec. 2013

[16] "Fundamentals of Control," *2013 ASHREA Handbook Fundamentals*. Sec. 7.5, Jan 2013

[17] "Onset Current Transducer," http://www.onsetcomp.com/products/sensors/ctv-a, [Aug. 25, 2013]