OPTIMIZED CONTROL OF STEAM HEATING COILS

A Thesis

by

MIR MUDDASSIR ALI

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

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Major Subject: Mechanical Engineering

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Approved by:

Committee Chair,	David E. Claridge
Committee Members,	Michael Pate
	Jeff Haberl
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ABSTRACT

Optimized Control of Steam Heating Coils.

(December 2011)

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Steam has been widely used as the source of heating in commercial buildings and industries throughout the twentieth century. Even though contemporary designers have moved to hot water as the primary choice for heating, a large number of facilities still use steam for heating. Medical campuses with onsite steam generation and extensive distribution systems often serve a number of buildings designed prior to the mid-1980s. The steam is typically used for preheat as its high thermal content helps in heating the air faster and prevents coils from freezing in locations with extreme weather conditions during winter.

The present work provides a comprehensive description of the various types of steam heating systems, steam coils, and valves to facilitate the engineer’s understanding of these steam systems.

A large percentage of the steam coils used in buildings are provided with medium pressure steam. Veterans Integrated Service Network and Army Medical Command Medical Facilities are examples which use medium pressure steam for heating. The current design manual for these medical facilities recommends steam at 30psig be provided to these coils. In certain cases although the steam heating coil is

designed for a 5psig steam pressure, it is observed that higher pressure steam is supplied at the coil. A higher steam pressure may lead to excessive heating, system inefficiency due to increased heat loss, simultaneous heating and cooling, and increased maintenance cost.

Field experiments were conducted to evaluate the effect of lowering steam pressure on the system performance. A 16% reduction in temperature rise across the coil was found when the steam pressure in the coil was reduced from 15psig to 5psig. The rise in temperature with lower pressure steam was sufficient to prevent coil freeze-up even in the most severe weather conditions. Additional benefits of reduced steam pressure are reduced flash steam losses (flash steam is vapor or secondary steam formed when hot condensate from the coil is discharged into a lower pressure area, i.e., the condensate return line) and radiation losses, increased flow of air through the coil thereby reducing air stratification and reduced energy losses in the event of actuator failure.

The work also involved evaluating the existing control strategies for the steam heating system. New control strategies were developed and tested to address the short comings of existing sequences. Improved temperature control and occupant comfort; elimination of valve hunting and reduced energy consumption were benefits realized by implementing these measures.

DEDICATION

TO MY FAMILY

ACKNOWLEDGEMENTS

I would like to thank Dr. David Claridge for his guidance and support during the course of this research and the time spent at the Energy Systems Laboratory. I would also like to thank Dr. Michael Pate and Dr. Jeff Haberl for agreeing to be on the thesis committee and providing their vital inputs. I am grateful to Song Deng, Chen Xu and Qiang Chen who provided me with the opportunity to be part of the Continuous Commissioning® team at Texas A&M University.

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NOMENCLATURE

A	Area of Heat Transfer Surface (ft^2)
AHU	Air Handling Unit
ASHRAE	The American Society of Heating, Refrigerating and Air-Conditioning Engineers
BAS	Building Automation System
Btu/lbm	British Thermal Units per pound mass
C_{air}	Heat Capacity Rate of Air $= m_a * Cp_a$
C_r	Heat Capacity Ratio $= C_{min}/C_{max}$
C_v	Valve Coefficient $(\frac{lb}{hr-psia})$
C_{v-d}	Design Valve Coefficient $(\frac{lb}{hr-psia})$
DAT	Discharge Air Temperature
F/B	Face/Bypass Damper
HDT	Hot Deck Temperature
h_{a_w}	Actual Enthalpy of Wet Steam at given Pressure $(\frac{Btu}{lb})$
h_f	Enthalpy of Saturated Water at given Pressure $(\frac{Btu}{lb})$
$h_{g_{p_i}}$	Specific Enthalpy of Saturated Steam at Pressure P_i $\left({}^{Btu}/_{lbm}\right)$
h_g	Enthalpy of Dry Saturated Steam at given Pressure $(\frac{Btu}{lb})$
m_a	Mass Flowrate of Air $\left(\frac{lbm}{hr}\right)$

m_s	Mass Flowrate of Steam $\left(\frac{lbm}{hr}\right)$
MAT	Mixed Air Temperature
MEDCOM	United States Army Medical Command
NTU	Number of Transfer Units $= UA/C_{min}$
OAT	Outside Air Temperature
PRV	Pressure Reducing Valve
psig	Gauge Pressure
psia	Absolute Pressure
P_i	Inlet Pressure of Steam (psig)
P_o	Outlet Pressure of Steam (psig)
$\dot{Q}$	Heat Transfer Rate (Btu/hr)
T_{a_i}	Temperature of Inlet Air (°F)
T_{a_o}	Temperature of Outlet Air (°F)
T_s	Saturation Temperature of Steam (°F)
T_1	Inlet Air Temperature (°F)
T_2	Outlet Air Temperature (°F)
T_{s_p}	Temperature of Saturated Steam at Pressure P_i (°F)
U	Overall Heat Transfer Coefficient $\left({}^{Btu}/_{hr - ft^2\ °F}\right)$
VISN	Veterans Integrated Service Network
x	Dryness Fraction
ΔP	Differential Pressure (psia)

ΔT_{LMTD} Log Mean Temperature Difference (°F)

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1 INTRODUCTION

1.1 Background

To maintain acceptable indoor air quality it is vital that a certain percentage of the total air circulated in the building comes from a fresh air intake. In addition to maintaining occupant comfort, outside air intake is essential for maintaining a positive building pressure. This helps in reducing building energy consumption as it prevents intrusion of unconditioned outside air into the building through wall cracks, doors etc. This outside air needs to be preheated in locations where the outside temperature falls below freezing point in case of a fresh air unit or when mixed air temperature in case of re-circulation units falls below freezing. Failing to do will result in ruptured coils due to freezing. Air is preheated by preheat coils in the Air Handling Units (AHUs) using steam or heating hot water mixed with anti-freeze as the heat transfer medium.

An Air Handling Unit (AHU) is a device used to circulate conditioned air throughout all or part of a building. It typically consists of air filters, coils for heating and cooling classified as either preheat, cooling, heating or reheating coils. Supply fans help circulate the air directly into the space or through ducts.

Some units are equipped with humidifiers to maintain the humidity levels of the area they serve. Some AHUs also have return fans to help circulate air from the space back to the AHU.

This thesis follows the style of *ASHRAE Transactions.*

1.2 Motivation

During most of the twentieth century steam was widely used as a heating source in buildings. The steam was utilized either at localized steam radiators throughout the building or centrally at heating coils in AHUs.

Steam was a preferred heat source due to its inherent advantages such as:

I. Ease of transportation.

II. Increased reliability as it has less moving equipment.

III. Its high thermal content which helps in heating the building faster.

Despite the fact that steam has numerous advantages, it has been slowly replaced by heating hot water systems to resolve the control issues associated with steam heating such as:

I. High temperature of steam resulting in overheating of air, which may lead to unnecessary cooling.

II. Flooding of coils with condensate and its subsequent freezing when outside air temperature falls below 32°F.

III. Increased maintenance cost due to water hammer, corrosion of coils in the presence of non-condensable gases and leaking steam traps.

This transition has led to unfamiliarity with steam heating systems among present day engineers and building operators. Even though there has been a transition from steam to heating hot water in the design of modern heating systems, steam heating is still used in a large number of buildings located in regions with extreme freezing conditions during winter. Buildings designed prior to the 1980s, and medical facilities with centralized steam production and extensive distribution systems still make use of steam for heating to take advantage of the above mentioned benefits. The medical campuses of the South Central Veterans Affair Health Care Network (VISN 16) are a group of buildings which predominantly use steam for heating. According to the HVAC design manual for these hospitals the suggested operating steam pressure to the heating coils is 30psig (Office of Construction and Facilities Management, 2008).

Since temperature of the air exiting the coil increases monotonically as the steam pressure increases, a higher steam pressure may lead to overheating of the air and result in simultaneous heating and cooling. In addition to energy waste due to simultaneous heating and cooling, an improper operating strategy can result in hunting by the heating and chilled water valves and lead to control instability.

The objectives of this research include:

I. Evaluating the feasibility of reducing supply steam pressure to the coil and its benefits

II. Evaluating existing control strategies and developing improved control sequences for various steam heating systems encountered by engineers

2 LITERATURE REVIEW

This literature review includes an overview of steam heating system, its benefits and various developments carried out to improve its performance. It also looks into different temperature control methods and provides various options which aid in preventing coil freeze.

2.1 Steam Heating Systems: An Overview

The earliest usage of steam can be traced back to the late eighteenth century in Europe (Wulfinghoff, 2011). Steam was used as a source of heat in factories and mills. From the days of the development of low pressure steam heating through radiators made by the Tudor company in the late nineteenth century (1875-1885) (Roberts, 1995), steam heating has been widely researched by engineers.

The improvements in the design of steam heating systems, introduction of variable vacuum heating (Anonymous, 1927 and Thinn, 1930) and problems associated with steam radiators and their control strategies have been discussed by various authors including Otto (1909), Bromley (1913), Thinn (1927), Shears (1934), Redd (1966), and Gifford (2003).

Steam was replaced by hot water as the primary heating source by designers during the late 1970s to take advantage of its modulating capability, ease in maintaining temperature and elimination of coil freezing which resulted due to improper draining of condensate in case of steam heating coils (Coad, 1995 & Lee, 1995). This transition of heating system selection has led to unfamiliarity among design engineers and building operators of the attributes of steam systems and has reduced research interest/progress in the advancement of steam heating systems.

Coad (1995) further discusses the physics associated with steam heating. The dependence of steam heating capacity on its pressure, different control strategies and guidelines to operate low pressure steam heating systems have been addressed by him. Since the steam pressure directly affects its temperature; during low load conditions it is suggested that the steam pressure be reduced for improved control.

Coad and Lee conclude that steam heating systems have inherent benefits such as the heat transfer mostly due to the condensation process which is a constant temperature heat transfer phenomena, source independence and minimum shaft energy requirement as they do not require pumps for distribution unlike heating hot water systems. The common problems associated with a steam heating system such as steam hammer, coil freezing and excessive corrosion can be avoided by installing the appropriate steam traps and vacuum breakers.

Even though there has been a transition in preferred building heating systems, there are a large number of buildings which still use steam as the primary source of heating, and it is essential to understand their operating principles, system characteristics, benefits and the challenges associated with them. The above mentioned work provides a comprehensive background on these issues.

2.2 Control Strategies for Steam Heating Systems

Radden (1940) in his article on Automatic Heating Controls addressed the control strategy for a steam heating system. The three different modes of operation for steam heating systems suggested were:

I. Intermittent control wherein the steam is turned on/off to maintain the room temperature.

II. Modulating the steam system, which varies the differential pressure across the valve through an orifice to control the steam flow into the coil?

III. A sub-atmospheric system where the steam is supplied at varying pressures below atmospheric pressure while the condensate is drained from the coil through a condensate return pump.

Hilmer (1978) also emphasizes modulating the steam pressure based on the outside air temperature. Smith (1958) and Boed (1998) have addressed the control strategy for steam heating systems using similar modes of operation as Radden and Hilmer. In addition, Smith compared the steam heating system with a hot water system. Durkin (1999) compared face/bypass damper control with modulating steam valve control. Based on his experience in using these two control strategies, he concludes a face/bypass (F/B) damper provides better control. The benefits of an integral face/bypass damper over built-up type face/bypass dampers are its reduced space utilization and a constant pressure drop regardless of damper position (Owen, 2005).

These authors provide us with general guidelines to operate various steam heating systems. These guidelines serve as a foundation on which standardized control strategies can be developed to optimize the energy consumption and resolve any control issues that may arise.

2.3 Coil Freezing

Coil freezing is one of the biggest challenges faced by building operators in locations where the outside air temperature falls below freezing. Freezing is common in preheat coils due to faulty condensate removal systems or clogged steam coils with debris. As an engineer or a building operator, it is essential to have an understanding of causes for coil freeze and ways to avoid them.

Sheringer and Govan (1985) discussed various methods of controlling dampers and steam valves to prevent coil freezing. According to these authors, proper mixing of return air and cold outside air can be achieved when the outside air and return air dampers are placed on opposite sides of the mixing chambers rather than in a parallel arrangement. They also suggest that rather than using a single coil with a modulating valve to serve various outside air conditions, multiple coils with different heating capacities be installed in a series. One of the coils is equipped with a proportioning valve and is controlled from the discharge air temperature while the rest are controlled based on outside air temperature. Since the coils operate at moderate weather conditions most of the time, this will prevent sub-atmospheric conditions in the coil and coil freezing, as well as help reduce energy waste. Coad (1984) also recommends that sub-atmospheric conditions be avoided for coil safety purposes.

If sub-atmospheric steam heating is utilized White (1991) recommends installing vacuum breakers to prevent the condensate from flooding the coils in case of gravity return system. A vacuum breaker is a check valve which allows air to enter the coil as soon as vacuum is formed. The hydraulic head of the condensate in the coil now serves as the driving force to drain the condensate.

Solutions to prevent coil freezing such as installing vacuum breakers, proper selection of steam trap, pitching the coils towards the return main and using multiple control valves and steam coils have also been discussed by Alyea (1958) and Hilmer (1978).

While developing improved control sequences, the engineer must be aware of the potential causes of coil freeze and ways to mitigate them. The above mentioned papers provide generic solutions that can be considered.

2.4 Summary of Literature Review

This section summarizes the work carried out in the field of steam heating systems for buildings. Since the transition from steam to heating hot water as the preferred source of heating in buildings, there has been a reduced interest in steam heating systems among the research community. Consequently, less than half the literature reviewed here is from the last decade. The improvements in the design and control of these systems such as variable vacuum heating, two-pipe systems, etc. have been widely discussed by researchers and implemented over time with considerable success until today.

Though it is recommended by researchers that steam be used at low pressure for improved temperature control, no effort has been made to evaluate the effect of reducing the steam pressure based on the discharge air temperature. The authors mentioned have also discussed various methods of controlling discharge air temperature by employing face/bypass damper arrangement or modulating steam valves. They provide us with guidelines on operating these systems, but these sequences have a scope of improvement from energy consumption and control stability perspective.

The present work aims to address the above two shortcomings, i.e., evaluate the effect of reduced steam pressure on the temperature of the air leaving the preheat coil and its benefits, and to evaluate the existing control strategies while developing energy efficient control sequences for various steam heating systems encountered by commissioning engineers in the field.

2.5 Significance of the Study

Since steam has largely been replaced by heating hot water systems, current engineers and building operators are unfamiliar with steam heating system characteristics and ways to operate these systems efficiently. The current study focuses on educating engineers and building operators on steam heating systems by providing them with the following:

I. Overview of different types of steam heating system configurations, steam coils and valve types. This will help engineers have a better understanding of steam heating systems and aid them in selecting the most energy efficient control strategy.

II. Procedure to evaluate the feasibility of reducing steam pressure for a coil and its benefits.

III. Standard operating control sequences for various steam heating systems.

IV. Procedure to evaluate the steam pressure in the coil for various steam valve position and inlet air conditions.

2.6 Limitations

The steam pressure reduction feasibility study and the optimized standard operating sequences for the various steam heating systems developed as part of this thesis work assumes the following:

I. The pressure reducing valve on the steam line which supplies steam to the coil maintains a constant steam pressure at the inlet of steam valve serving the coil.

II. The steam coils do not have any ruptures on them and there is no steam leakage.

3 BASIC STEAM HEATING SYSTEM THEORY

This section provides a theoretical discussion on the effect of reduced steam pressure on the discharged air temperature for a given heating coil.

3.1 Theoretical Analysis

Let us assume that a steam heating coil with a given inlet steam pressure and steam flow rate is used to heat air. Figure 1 shows the control volume of the steam heating coil under consideration.

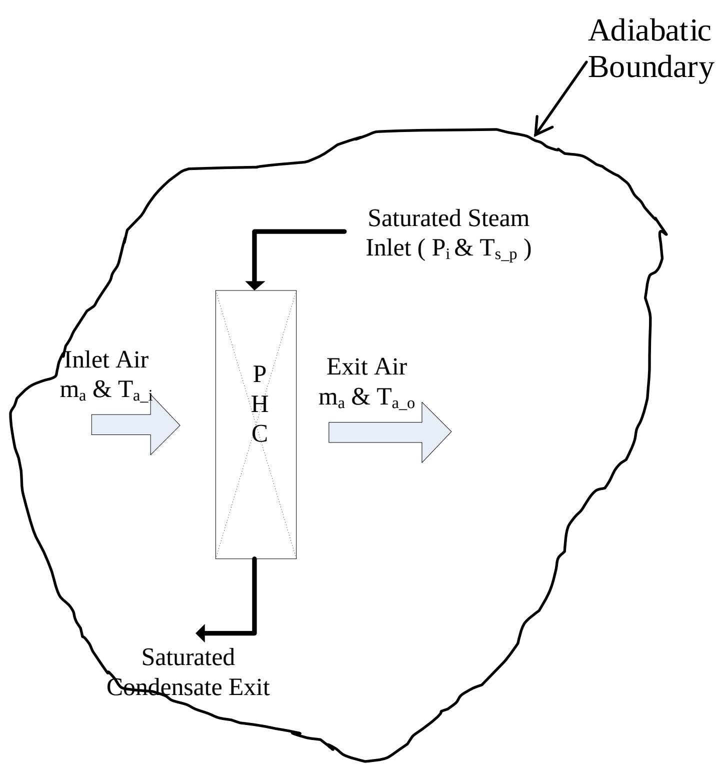

Figure 1: Control Volume of a Steam Heating Coil

The effectiveness of a heat exchanger (e) is defined as the ratio of actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate. It is given by:

$$e = [1 - \exp\{-NTU\,(1 + C_r)\}]/(1 + C_r) \qquad \textit{Eq. 3.1}$$

In a steam heating coil system the heat transfer occurs due to condensation of steam in the coil and can be approximated as a condenser type heat exchanger. Let us assume that heating occurs due to this condensation process and no sub-cooling of condensate occurs.

For a condenser type heat exchanger, $C_{max} = \infty$,

Hence the heat capacity ratio, $C_r = C_{min}/C_{max} = C_{min}/\infty = 0$

Therefore,

$$e = 1 - \exp(-NTU) \qquad \textit{Eq. 3.2}$$

The effective heat transfer is given by

$$q = e * C_{min} * (T_{s\_p} - T_{a\_i}) \qquad \textit{Eq. 3.3}$$

Hence, for a given heating coil, the effective heat transfer depends on the temperature differential between the steam saturation temperature and the inlet air temperature.

i.e. $$q \propto (T_{s\_p} - T_{a\_i}) \qquad \textit{Eq. 3.4}$$

According to the law of conservation of energy, the effective heat transfer by the heat exchanger is equal to the heat gained by the cold fluid.

Therefore,

$$q = e * C_{min} * \left(T_{s\_p\_} - T_{a\_i}\right) = m_a * Cp_a * (T_{a\_o} - T_{a\_i}) \qquad \textit{Eq. 3.5}$$

Based on the above equations, the temperature of the outlet air is linearly dependent on the saturation temperature of the steam. According to the properties of saturated steam, the saturation temperature increases as the operating pressure of steam increases. The saturation temperature increases from 212°F to 286°F as the steam pressure is increased from 0psig to 40psig.

In case of steam coils used in AHUs for heating, the condensate exits the coils in a sub-cooled condition. Assume that the coil is designed for 15psig inlet steam condition and the steam pressure is then reduced to 5psig. The exit condition of condensate is assumed to remain the same for both the conditions at 180°F and atmospheric pressure (0 psig).

Enthalpy of Saturated Steam @ 15 psig = 1,163.9 Btu/lb_m

Enthalpy of Saturated Steam @ 5 psig = 1,155.9 Btu/lb_m

Enthalpy of Condensate = 148 Btu/lb_m

The enthalpy difference between the two states (15 psig and 5 psig) is equal to 8 Btu/lbm which accounts for less than 1% of the initial value.

If heating equipment is sized for the most extreme weather it will not only result in high initial cost but high operating cost as the equipment will run at lower part load efficiency most of the time. A compromise on actual design conditions is made to reduce the equipment cost and eliminate lower system inefficiencies while accepting the risk of occupant discomfort during extreme weather condition (Curtiss et al., 2001).

ASHRAE has published design conditions for various locations based on the weather statistics and probability of its occurrence. In case of heating equipment, design temperatures are classified as 99% and 99.6% design temperatures. This refers to the annual percentage of time the outside air is above the design temperature (ASHRAE, 2009). The heating design temperatures in general have increased by 0.2°F compared to the 2005 edition. The 2005 edition design temperatures compared well with those of the 2001 edition (Thevenard and Humphries, 2005). Colliver and Burks (2000) compared the design temperatures between the 1993 and previous edition and found the mean of the difference in heating design temperatures was 1.03°F, indicating less extreme design conditions in the 1993 edition. The difference in the design temperatures among various editions can be attributed to the improvement in the estimation methodology, use of most recent periods of weather data and conversion from seasonal frequency to annual frequency estimation.

The buildings in the second half of the 1900s were designed with a heating safety factor of 10-20% to account for piping loss, and a 10-20% factor of safety for warm-up load along with additional safety factors determined by the designer. These steam heating systems are predominantly encountered in older buildings, and based on the above discussion these systems are designed for slightly more extreme outside air conditions and have a higher factor of safety compared to present day construction. Even though lowering steam pressure will reduce the heating capacity of the coil, the coil may have sufficient surface area to meet the heating load for all or a major portion of the operating period.

A lower operating steam pressure will result in increased airflow through the coil and reduce air stratification. It will also help in reducing the steam loss through traps, flashing of steam and reduced radiation loss.

4 STEAM HEATING SYSTEMS

Many current engineers and building operators are unfamiliar with the different types of steam heating systems and their inherent characteristics. This section provides a brief description of the various steam heating systems and different types of coil designs encountered in commercial buildings across the United States.

4.1 Classification of Steam Heating Systems

A steam heating system can be classified into various categories as shown in Figure 2. Based on the operating pressure it is classified into vacuum, low pressure, medium pressure and high pressure steam heating. In vacuum heating, the steam is supplied at sub atmospheric pressure (less than 15psia) to the coils or radiators while a low pressure system generally operates in the range of 2psig-14psig. A medium pressure system is supplied with steam in the range of 15psig-50psig and high pressure systems make use of steam at pressures higher than 50psig (ASHRAE GreenGuide, 2006).

Based on the arrangement of the distribution system, steam heating systems are also classified as one-pipe or two-pipe systems. This classification is based on the mode of returning the condensate to the boiler. In a one-pipe system, a single pipe is used to supply the steam to the coils and return the condensate back to the boiler while in a two-pipe system the steam is supplied through the supply line while the condensate is returned through a separate line.

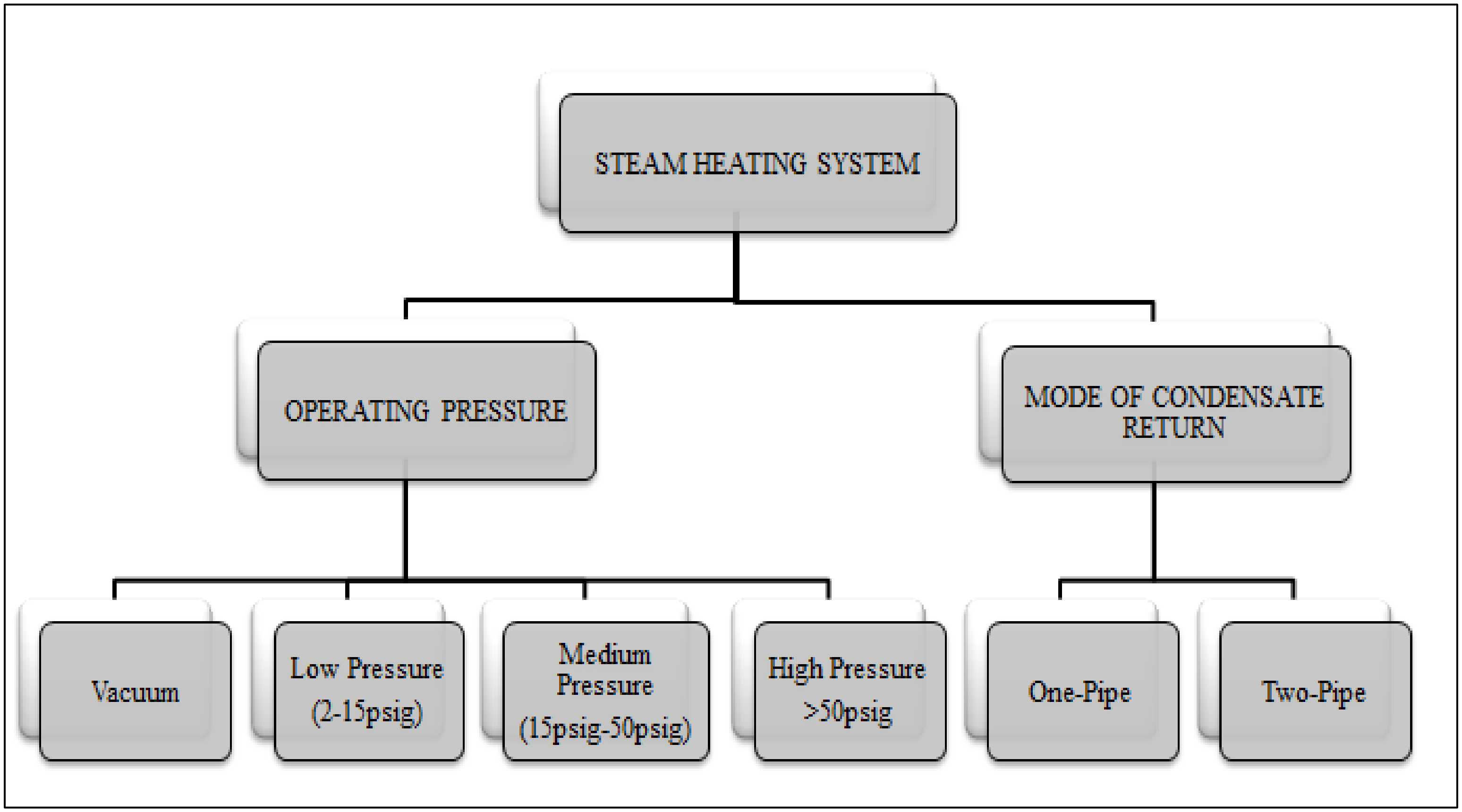

Figure 2: Steam Heating System Classification

The condensate can be brought back to the boiler with the help of gravity or through condensate return pumps.

4.2 Types of Steam Heating Coils

The steam heat coils are broadly classified based on the coil arrangement and mode of control as shown in Figure 3. The commonly installed steam heating coils are mentioned below.

Based on Coil Arrangement:

- *Single Row Parallel Coils:* This is the most prevalent type of steam heating coil. Steam is supplied to a series of parallel tubes through a common header on one end of the coil and the condensate is drained through an outlet header located on the other side. Since airflow across the coil is seldom uniform, there is a non-uniform condensation of steam in the coil resulting in erratic temperature distribution (Bowser, 1985).
- *Multiple Row Parallel Coils:* This coil type is equivalent to stacking of multiple single row coils. Steam is supplied to and condensate returned from all the rows through common headers located on the opposite sides of the coil. Since the first row of the coil is in contact with the coldest stream of air, more steam condenses in this row than in the succeeding rows. Consequently this row has the highest pressure drop and steam from the outer header may flow into the first row trapping the condensate. When maximum heat exchanger performance or freezing inlet air temperature is expected, it is recommended that multiple row coils be replaced with a tandem coil.

- *Tandem Coils:* A tandem coil is a modification to a multiple row parallel coil so as to eliminate the trapping of condensate due to varying pressure in each coil. The coil is designed such that the total amount of steam to the coil is supplied to the first row. The condensate formed in the first row and the remaining steam for the other rows flows into a larger header at the other end of first row. The header is provided with baffles to channel the steam to subsequent rows and the condensate is collected.
- *Distributing Tube Type Coil:* Air stratification similar to that witnessed in bypass type coils occurs in conventional steam heating coils during reduced load operation. During low load period when the valve is throttled, the rate at which the steam enters the coil is lesser than the condensation rate. If no provision is provided to drain the condensate from the coil, it restricts the flow of steam throughout the coil resulting in practically no heating effect as the air is in contact with condensate. To overcome this problem, the distributing tube type coil was developed. It essentially consists of two tubes arranged concentrically. The inner tube is perforated with holes evenly placed along the whole length. The steam flows axially through the tube and is released at fixed intervals and practically in equal amounts regardless of the quantity supplied and condenses in the annulus.

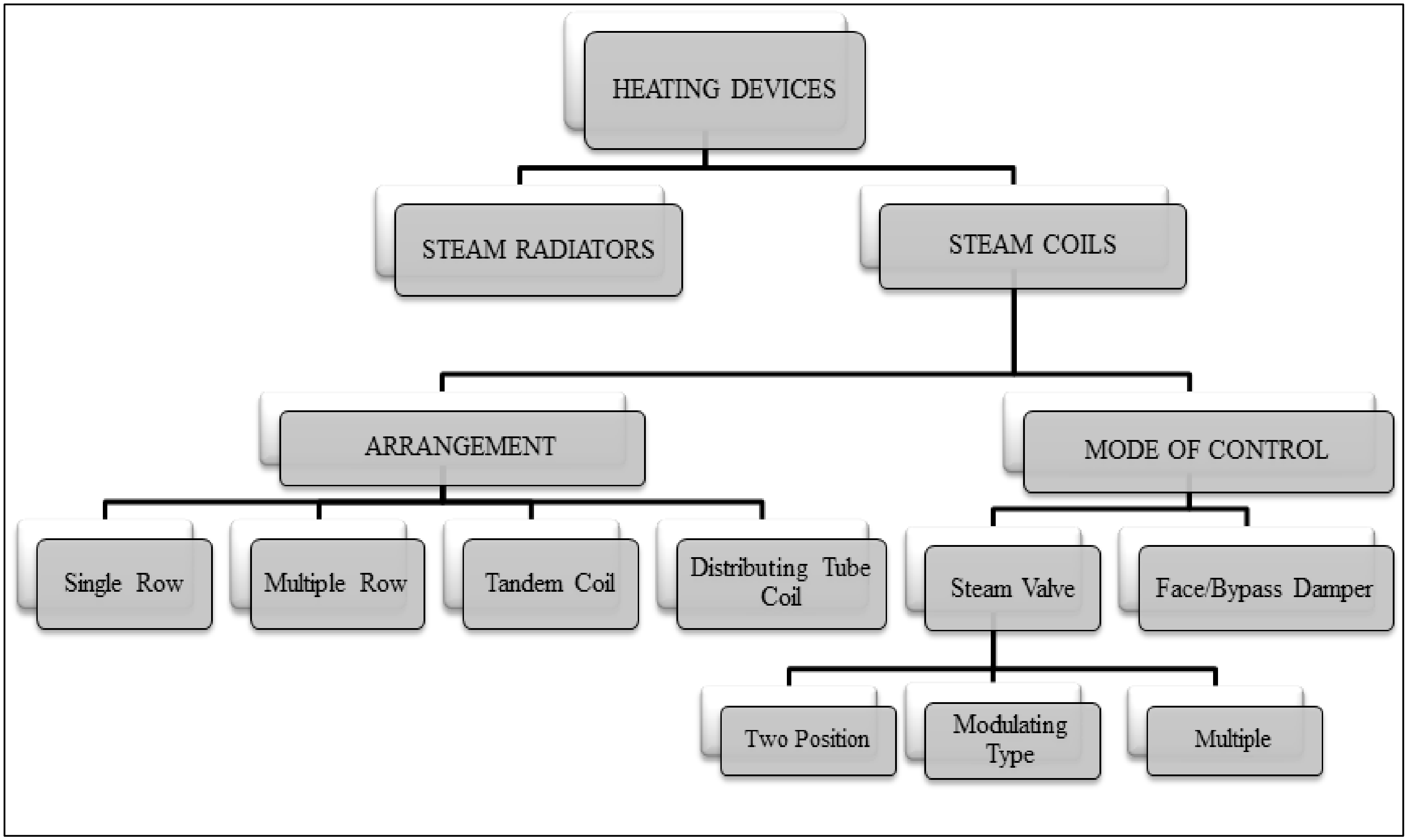

Figure 3: Classification of Steam Heating Devices

The steam condensate drains back to the condensate return header, staying warm as it is in continuous contact with steam flowing through the inner tube. In addition to providing an even heat distribution these coils are more difficult to freeze than the conventional steam coils.

Based on Mode of Control:

- *Coils with a Control Valve:* This type of coil has the simplest design and is very similar to a hot water coil. A control valve is installed to modulate the steam flow through the coil in order to maintain the preheat temperature. The control valve can be either a two-position or a modulating type.

In a two-position valve control configuration, the valve is either in a fully open or closed position with no intermediate state. This type of control configuration is not popular as it causes undesirable thermal stresses, water hammer issues and needs rapid readjustment of the chilled water or reheat valve control causing hunting of valves. This valve hunting leads to control instability and increased maintenance costs (Sheringer and Govan, 1985).

A modulating valve makes use of proportional control. The valve is modulated from a fully open position to fully close in order to maintain the air temperature. Figure 4 shows a typical arrangement of a steam control valve. The restriction due to modulation causes reduction in the steam flow rate to the coil resulting in reduced pressure in the coil. This reduced steam pressure helps in maintaining the discharge air temperature. Care needs to be taken to prevent sub-atmospheric conditions in the coil. If no vacuum breakers are installed it may lead to flooding of the coils with condensate and subsequent coil freezing.

Some designers make use of multiple preheat coils for improved temperature control and to eliminate the freezing hazard. One of the coils is controlled on-off based on the outside air temperature while the other coil makes use of a proportioning valve to maintain the discharge air temperature. Figure 5 shows a schematic of this type of arrangement.

In larger coils, for improved control, a single coil is provided with two control valves usually referred to as a 1/3rd valve and a 2/3rd valve. The valves are sized in such a fashion that the smaller valve accounts for 1/3rd of the total design flow while the rest is delivered through the other valve. When the system calls for heating, the smaller valve is opened first and modulated. When the heating effect provided by the smaller valve is not enough, the larger valve modulates. During the call for cooling the larger valve closes first followed by the smaller valve (Calabrese, 2003, Sheringer and Govan, 1985). Figure 6 illustrates a typical 1/3rd -2/3rd control valve configuration. This arrangement helps in reduced simultaneous heating and cooling compared to a single control valve configuration.

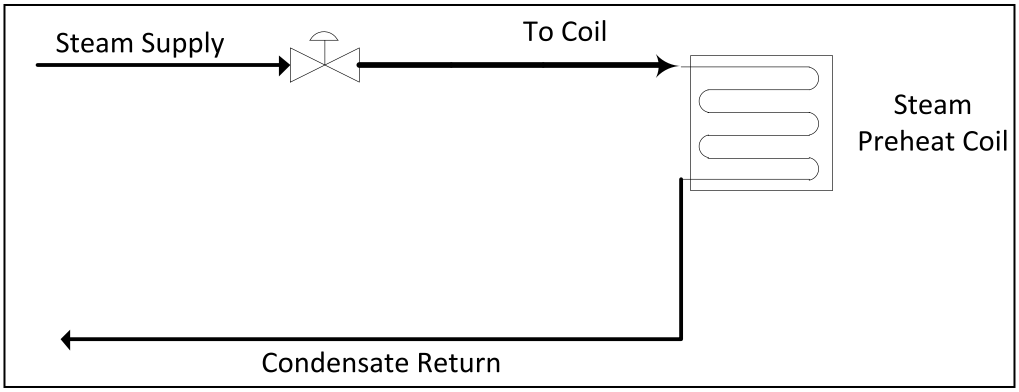

Figure 4: Preheat Coil with Control Valve Schematic

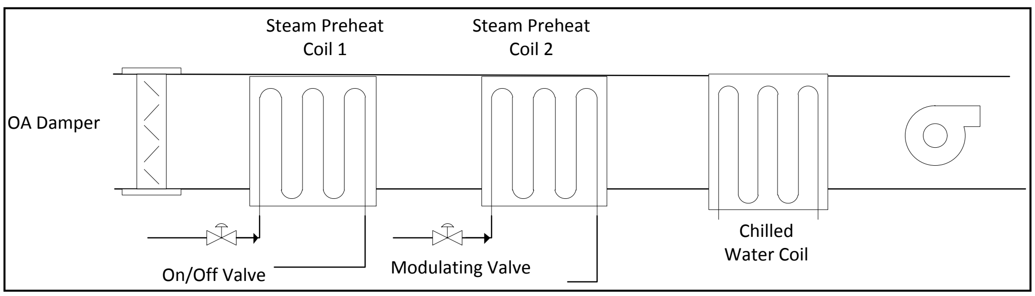

Figure 5: Multiple Preheat Coil Schematic

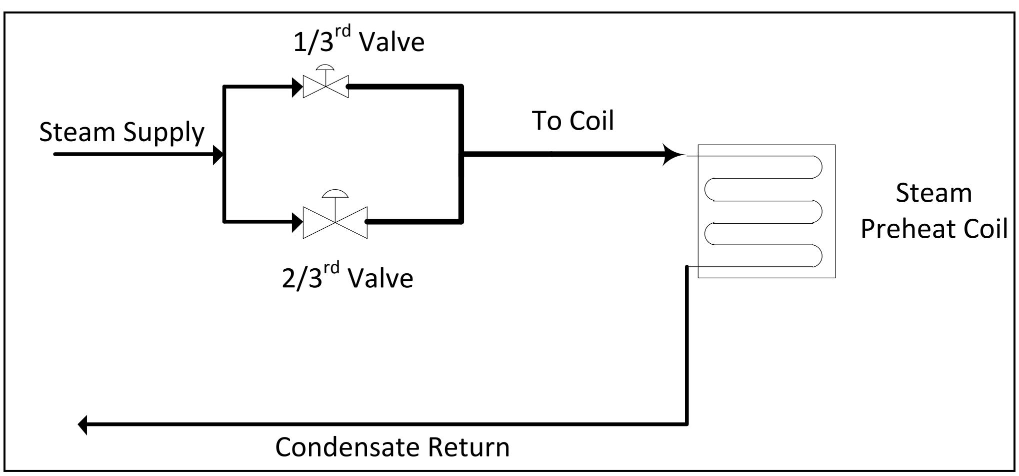

Figure 6: Preheat Coil with 1/3rd-2/3rd Valve Arrangement Schematic

To better understand the operation of these three different kinds of control valve arrangements let us assume a steam coil requires 400lb_m/hr. of steam at its design condition. In the first case the control valve is designed for a flow rate of 400lb_m/hr. and the valve is modulated to vary the steam flow and maintain the discharge air temperature.

In case of a multiple coil arrangement with an ON/OFF and modulating valve, *Steam Preheat Coil I* (ON/OFF type valve) can be assumed to be designed for a steam flow rate 75 lb_m/hr. while *Steam Preheat Coil II* (modulating valve) has a design flow rate 325 lb_m/hr. If the outside air temperature is above 25°F, steam flow through *Steam Preheat Coil II* is modulated to maintain the discharge air temperature. When the outside air temperature falls below 25°F *Steam Preheat Coil I* is enabled while flow through *Steam Preheat Coil II* is modulated to provide supplemental heating to maintain discharge air temperature.

For a 1/3rd -2/3rd valve arrangement, the 1/3rd valve is designed for a steam flow rate of 133 lb_m/hr. and the 2/3rd valve accounts for the remaining design steam flow. The 1/3rd valve is activated first and modulated to maintain the discharge air temperature and the 2/3rd valve is in a fully closed position. When the 1/3rd valve is fully open and the discharge air temperature setpoint not met the 2/3rd valve is commanded open and modulated to meet the heating requirement.

- *Face/Bypass Damper Steam Coil:* In this system the discharge air temperature is maintained by mixing hot and cold air in varying proportions rather than modulating steam flow. This type of system employs a set of dampers to direct incoming air either over the coil or bypass it depending on demand. A constant steam flow rate keeps the condensate moving out of the coil.

Variable static pressure drop, large space requirement and temperature overshoot in the bypass mode are the common problems with this type of system. Figure 7 provides a schematic of this type of heating system.

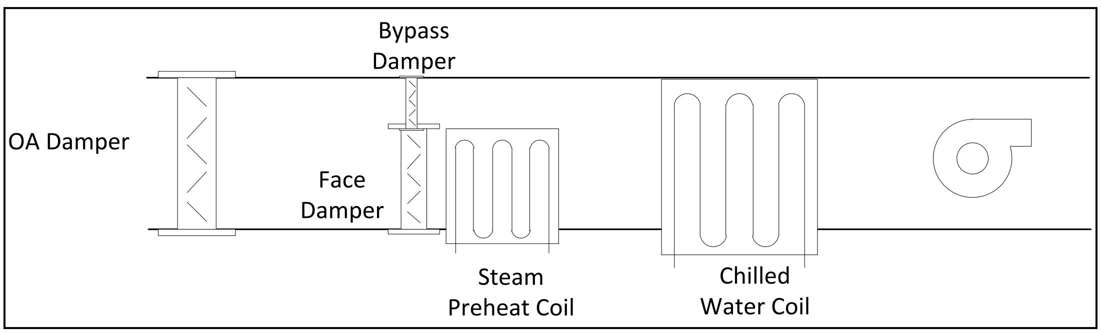

Figure 7: Built-in Face/Bypass Dampers

Integral face/bypass damper coils are an improvement over the built-in face/bypass coils discussed above. The dampers are an integral part of the coil and hence are compact and efficient. The dampers operate like a clamshell as shown in Figure 8 (published with permission from the author). They enclose the coil area when in the full bypass mode or block the bypass path during the face mode or in between while trying to maintain the discharge air temperature. The steam is usually supplied at a constant flow rate through a two-position control valve.

One of the disadvantages of the bypass method is the separation of air into hot and cold air streams. Unless proper mixing of the two streams occurs on the discharge side, the stratification of air streams can continue over long lengths of duct and cause comfort issues.

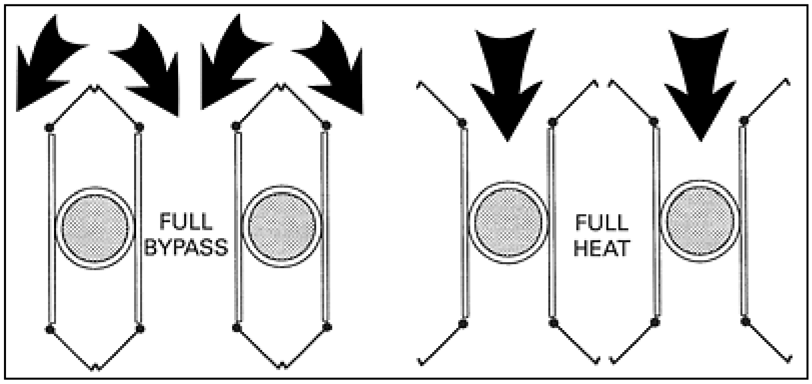

Figure 8: Integrated Face/Bypass Damper Arrangement (Owen, 2005)

The preheat coils can be installed either on the outside air intake duct or after the mixing chamber. Various combination of the basic types discussed above are widely used by designers for improved control and reduced maintenance. A list of commonly employed combinations is summarized below:

I. Heating coil with a two-position and modulating valve.

II. Distributing coil with face/bypass damper.

III. Face/bypass damper coil with a modulating valve.

IV. Face/bypass damper coil with 1/3rd and 2/3rd valve arrangement.

4.3 Steam Valve Sizing

In addition to the types of steam heating systems, it is important for commissioning engineers to be aware of the pressure drop across a control valve. This will help the engineers in determining the optimum line pressure for the system for improved control and reduced energy consumption. As discussed in the previous section, two types of control valves are used with steam heating coil, namely the *two-position* or *on-off* type and the *proportional or modulating* type.

The control valve for an application is selected based on the designed pressure drop since it determines the heating capacity of the coil. The two-position control valves are designed for a *minimum pressure drop (10% of inlet pressure)* as the coils used in this configuration are sized based on the supply line pressure and temperature.

In case of proportional controls the valves are desired to have a high authority as it helps in better regulating steam flow to the coil (Young, 1958). Valve authority is defined as the ratio of pressure drop across a fully open valve to the total pressure drop across the circuit. A high valve authority provides a linear steam flow rate through the valve when modulated thereby complementing the linear characteristic of a steam coil.

Usually a modulating valve is selected such that a *critical pressure* condition occurs downstream of the valve as explained below. This helps in reducing the valve size and its cost. A critical pressure drop will also provide a linear characteristic to the valve. For steam the critical pressure condition occurs when the pressure drop across the valve is *42% of the inlet steam pressure*. A detailed description of the critical pressure phenomena is provided in Appendix B.

5 OVERVIEW OF FIELD EXPERIMENT

5.1 Description of Facilities

The experimental investigations were carried out at three facilities located in the United States: Alexandria Veteran's Affairs Medical Center located in Pineville, Louisiana, McDonald Army Community Hospital, Fort Eustis, Virginia and Ireland Army Community Hospital, Fort Knox, Kentucky.

5.1.1 Alexandria Veteran's Affairs Medical Center

This facility is part of the South Central VA Health Care Network commonly referred to as VISN 16 (Veterans Integrated Service Network 16). It is composed of multiple buildings and houses various functions such as hospitals, a patient care center, office space and a laundry. A centralized boiler system supplies steam to the buildings for heating and other process needs. An aerial view of the campus is shown in. The high pressure steam (100psig) from the central plant is reduced by pressure regulating valves (PRVs) to medium pressure (25-30psig) at the buildings.

The experiment was carried out on a Single Duct Constant Air Volume (SDCAV) AHU. The steam coil is used for preheating of the outside air. The specifications of the AHU selected for experiment are provided in Table 1. The HVAC equipment in the facility is controlled through a Johnson Controls Inc. Metasys PMI BAS.

Table 1: Experimental AHU Equipment Nameplate Information

Unit Type	SDCAV
Airflow Rate	7,800 cfm
OA Percentage	100%
Heating Source	Steam
Steam Pressure at the Valve	26psig
Steam Pressure Entering Coil	15psig
Flow size	5' x 4'
Number of Points Measured	Single Point
Position	Center of Flow Path

5.1.2 McDonald Army Community Hospital, Virginia

This facility is part of the United States Army Medical Command and is located in Fort Eustis, Virginia. It is a three story community hospital with eight AHUs. Experiments were carried out on two AHUs, one equipped with a two-position valve with a face/bypass damper arrangement while the other was equipped with a $1/3^{rd}$ - $2/3^{rd}$ modulating valves arrangement. The specifications of the AHU selected for experiment are provided in Table 2. All the HVAC equipment was controlled through a Johnson Controls Inc. Metasys 5.0 BAS.

Table 2: AHU with $1/3^{rd}$ - $2/3^{rd}$ Valve Nameplate Information

Unit Type	SDCAV
Airflow Rate	11,030 cfm
OA Percentage	100%
Heating Source	Steam
Steam Flow rate	205 pph.
Steam Pressure at the Valve	12 psig
Steam Pressure Entering Coil	10.7 psig
Number of Points Measured	Averaging Sensor

5.1.3 Ireland Army Community Hospital, Kentucky

This facility is also part of the United States Army Medical Command and is located in Fort Knox, Kentucky. It is an eight story community hospital with over forty AHUs. Steam coils are used in these AHUs for preheating/heating based on the weather requirements. All the HVAC equipment is controlled through a Trane Tracer Summit BAS.

5.2 Conducting the Building Experiments

As a part of the research, the existing control strategies for the preheat coils were evaluated. Also the conditions of the control equipment, including control valves, face/bypass dampers and pressure reducing valves (PRVs) and the feasibility of reducing the steam pressure for the system were evaluated. Trends were set up to monitor critical parameters such as preheat air temperature, discharge air temperature, etc.

After the baseline data was obtained, changes were carried out on the preheat system and its effect on the discharge air temperature and system control was monitored. A list of changes carried out during this time period is provided below.

I. Effect of inlet steam pressure reduction on the discharge air temperature.

II. Modified control sequence.

5.3 Data Collection

The data corresponding to various parameters specified in Table 3 was monitored through the BAS. The data was collected at regular intervals and stored on the BAS server.

Table 3: Parameters Trended During Experiments

S. No	Parameter
1	Preheat Air Temperature
2	Discharge Air Temperature
3	Outside Air Temperature
4	Steam Valve Position
5	Chilled Water Valve Position

For the units which were not equipped with temperature sensors to measure the preheat air temperature, the data was collected by using external temperature probes. A single point temperature measurement was made for units which were not equipped with averaging sensors. The point at which the temperature was measured was selected such that it represented the average temperature of the air stream. Hobo TMC-HD series and Fluke 80PK-24 temperature probes were used and the data was collected on TSI and Hobo data loggers. Detailed specifications of the equipment used in the experiments are provided in the Appendix A.

6 PROJECT RESULTS

This section presents the detailed description of the existing control strategies for the specimen AHUs, modifications carried out and the results of these experiments.

6.1 Reduction of Steam Pressure

The effect of reduced steam pressure on the discharge air temperature was conducted at Veteran's Affair Medical Center, Alexandria, LA. Figure 9 provides a schematic of the specimen AHU. Initially the steam was supplied at 15psig to the preheat coil and the discharge air temperature was monitored with the help of external temperature probes. The steam pressure was reduced to 5psig by adjusting the PRV and data was collected. The steam pressure in the supply line (PG 1) was measured at 26psig and 15psig for a coil inlet steam conditions (PG 2) of 15psig and 5psig respectively. Figure 10 shows a 1-minute interval trend data from the experiment.

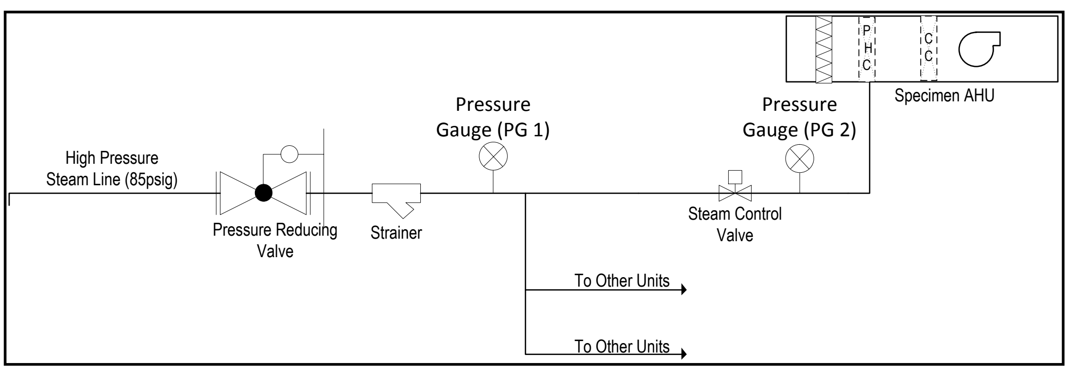

Figure 9: Schematic of Specimen AHU and its Steam Supply, Alexandria, LA

Based on the data collected, it is observed that the steam pressure reduction resulted in reducing the average temperature rise across the coil from 36°F to 30°F, corresponding to a 16% reduction in the preheat temperature difference.

Since the experiments were carried out for an outside air condition of 68.5°F, let us estimate the temperature rise when the steam pressure is reduced for sub-freezing outside air conditions (assume 22°F).

As discussed in Section 3.1 the heat transfer across a coil is proportional to the temperature difference of the steam temperature and the inlet air condition.

Saturation Temperature of Steam at 15psig T_{sat_15} = 250°F

Saturation Temperature of Steam at 5psig T_{sat_5} = 227°F

Let us assume the inlet air temperature condition of the air is 68.5°F.

According to Eq. 3.4

$$q \propto (T_{s\_p} - T_{a\_i})$$

Therefore, heat transfer for 15psig steam in the coil and 68.5°F inlet air condition is

$$Q1 \propto (250 - 68.5)$$

$$Q1 \propto 181.5$$

For a steam pressure of 5psig and similar inlet air condition the heat transfer is proportional to

$$Q2 \propto (227 - 68.5)$$

$$Q2 \propto 158.5$$

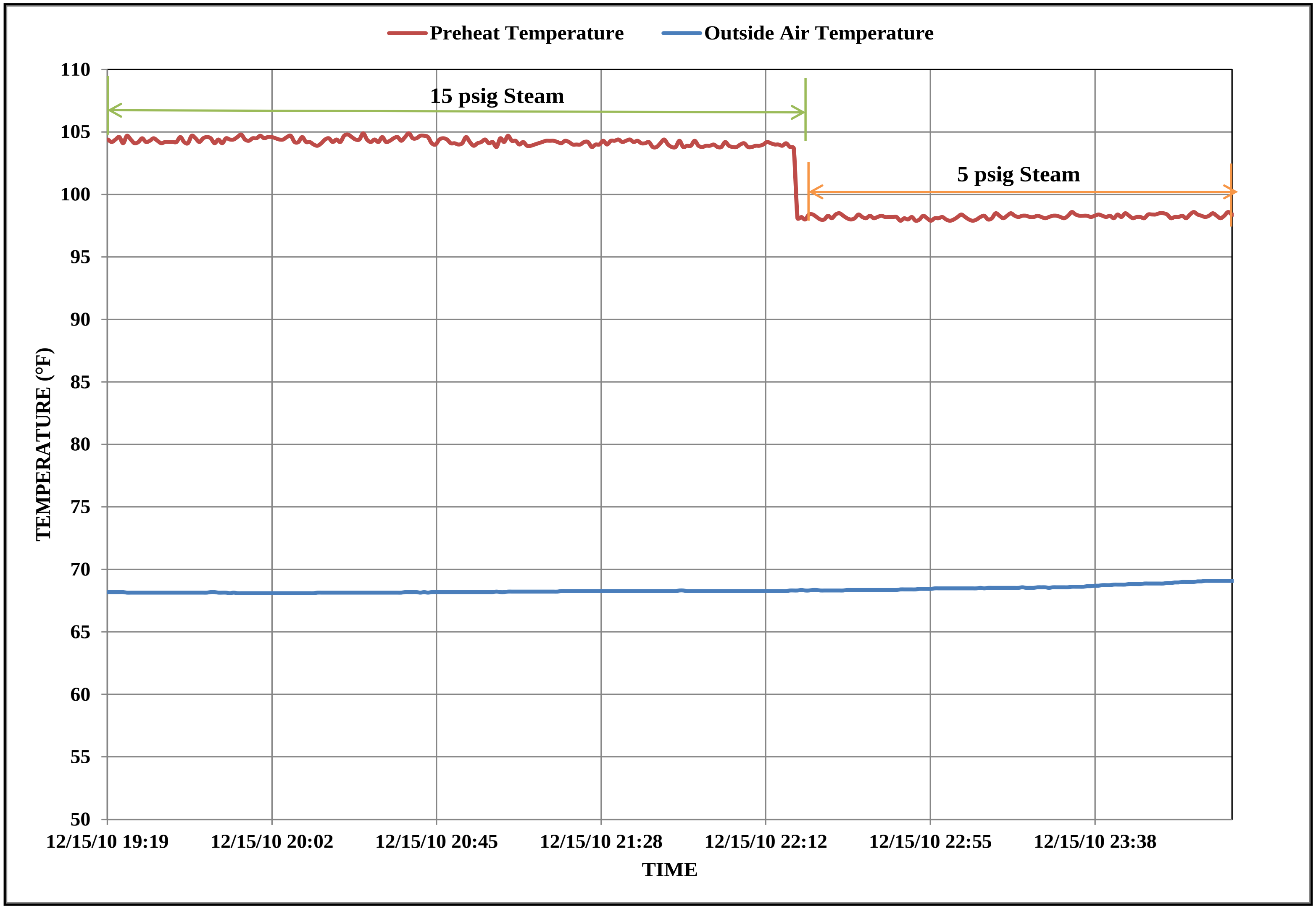

Figure 10: Preheat Temperature Plot for Steam Pressure Reduction

From Eq. 3.5 we can infer

$$q \propto (T_{a\_o} - T_{a\_i})$$

$$q \propto \Delta T_{air}$$

$$q = e * C_{min} * \left(T_{s\_p\_} - T_{a\_i}\right) = m_a * Cp_a * (T_{a\_o} - T_{a\_i})$$

Hence, the ratio of temperature rise of the air for the two steam pressure condition is given by

$$(\Delta T1)/\,(\Delta T2) = 181.5/158.5$$

$$(\Delta T1)/\,(\Delta T2) = 1.14$$

If the inlet air condition is changed to 22°F, based on the discussion provided above if the steam pressure is reduced from 15psig to 5psig, the ratio of temperature rise is

$$(\Delta T1)/\,(\Delta T2) = 228/205$$

$$(\Delta T1)/\,(\Delta T2) = 1.11$$

The percentage change in the temperature rise when the inlet air condition is changed from 68°F to 22°F is a 3% change.

From Eq. 3.5

$$e * C_{min} * \left(T_{s\_p\_} - T_{a\_i}\right) = m_a * Cp_a * (T_{a\_o} - T_{a\_i})$$

In case of a condenser type heat exchanger$C_{min} = m_a * Cp_a$, the effectiveness of heat exchanger can be calculated based on the temperature rise obtained for 15psig and 5psig steam pressure.

$$e * \left(T_{s\_p\_} - T_{a\_i}\right) = (T_{a\_o} - T_{a\_i})$$

$$e * (250 - 68.5) = (104.2 - 68.2)$$

$$e * 181.5 = 36$$

$$e = 0.198$$

For the given steam heating coil, if the inlet air temperature was reduced to 22°F, the temperature rise of air stream for 15psig and 5psig steam can be calculated from the equations developed in Section 3.1.

For 15psig steam pressure,

$$e * \left(T_{s\_p\_} - T_{a\_i}\right) = \left(T_{a\_o} - T_{a\_i}\right)$$

$$e * \left(T_{s\_p\_} - T_{a\_i}\right) = \left(T_{a\_o} - T_{a\_i}\right)$$

$$0.198 * (250 - 22) = \left(T_{a\_o} - T_{a\_i}\right)$$

$$\left(T_{a\_o} - T_{a\_i}\right) = 45.15°F$$

For 5psig steam pressure,

$$e * \left(T_{s\_p\_} - T_{a\_i}\right) = \left(T_{a\_o} - T_{a\_i}\right)$$

$$0.198 * (227 - 22) = \left(T_{a\_o} - T_{a\_i}\right)$$

$$\left(T_{a_o} - T_{a_i}\right) = 40.59°F$$

The temperature rise across the coil would increase to 45.15°F and 40.6°F when the outside air temperature falls to 22°F for 15psig and 5psig steam respectively. Table 4 provides binned TMY3 weather data for Alexandria, LA. Based on this weather data to maintain a discharge air temperature setpoint of 55°F, it can be concluded from the experiments and theoretical calculations carried out that 5psig steam will provide sufficient heating during the entire heating season. During severe cold conditions the steam pressure can be raised to prevent freezing.

Table 4: Bin Data for Alexandria, LA

Dry Bulb Temp. Bins		Totals	
		# of hours	Mean Coincident Wet Bulb Temp.
14	18	0	
18	22	2	21
22	26	14	23
26	30	114	27
30	34	223	31
34	38	220	35
38	42	201	38
42	46	394	41
46	50	489	45
50	54	598	49
54	58	572	53
58	62	496	57
62	66	629	60
66	70	1060	65
70	74	1021	69
74	78	751	72
78	82	639	74
82	86	499	74
86	90	493	76
90	94	234	78
94	98	81	78
98	102	13	75
102	106	17	76
106	110	0	
Total		**8760**	

6.1.1 Benefits of Reduced Steam Pressure

It is recommended that the steam coils be supplied with steam at the lowest feasible pressure. While determining the lowest feasible pressure required for the coil, detailed measurement and analysis should be conducted to ensure that there will be adequate heating capacity at design conditions.

The benefit of using steam at lower pressure is provided below with reference to the above mentioned AHU.

I. *Improved Steam Quality:* The amount of liquid water present in the steam /saturated condensate mixture determines the quality of steam and is measured by its dryness fraction. Dryness fraction is defined as the ratio of mass of dry steam present in a known mass of steam. Even though boiler manufacturers recommended operating steam boilers at their maximum design pressure, this is seldom practiced. When the steam demand is low or when the process/heating load requires pressure reducing station, boilers are operated at substantially less than design pressure. This low pressure operation reduces the steam quality exiting the boiler (Hahn, 1998).

The heating capacity for wet steam is given by:

$$h_{a\_w} = x * h_g + (1 - x) * h_f$$

From the above equation we can conclude that the heating capacity is linearly dependent on the steam quality. In addition to reduced heating capacity, a lower steam quality causes erosion of pipes, valves, coils etc. due to the presence of water droplets. Since pressure reduction using a PRV is an iso-enthalpic process, a portion of the energy associated with higher pressure steam is utilized in converting the water particles into steam. A lower operating pressure at the coils would help provide a better quality steam

The steam at Alexandria is supplied at 100psig to the various buildings located on campus including the steam sterilizers. The sterilization process typically requires steam with a dryness fraction of 0.97 (Moore, 2008), at an operating pressure of 60psig.

The steam pressure required at the valve inlet (PG 1) for the two scenarios tested as part of the steam pressure reduction feasibility is 26psig and 15psig.

If,

$$x = Dryness\ Fraction = 0.97$$

$$h_{60\_0.97} = Actual\ Enthalpy\ of\ 95\%\ Wet\ Steam\ at\ 60psig\ \left(\frac{Btu}{lb}\right)$$

$$h_{g\_60} = Enthalpy\ of\ Dry\ Saturated\ Steam\ at\ 60psig\ \left(\frac{Btu}{lb}\right) = 1183$$

$$h_{f\_60} = Enthalpy\ of\ \ Saturated\ Water\ at\ 60psig\ \left(\frac{Btu}{lb}\right) = 277$$

From the above mentioned discussion, the actual enthalpy of 97% dry steam at 60psig is given by:

$$h_{60\_0.97} = x * h_{g\_60} + (1 - x) * h_{f\_60}$$

$$h_{60\_0.97} = 1183 * 0.97 + 0.03 * 277$$

$$= 1155.8\frac{Btu}{lb}$$

Based on the enthalpy values of saturated steam and water, the steam quality at the exit of the PRV for an outlet pressure of 26 psig is computed as 0.986. The steam quality is improved to 0.991 when the steam pressure is further reduced to 15 psig

II. *Reduced Air Stratification:* Since the coil surface temperature is proportional to the operating steam pressure, a lowered steam pressure will help in reducing the air stratification in a face/bypass type heating coil as less air bypasses the coil. Table 5 shows the percentage of air flow required through the coil to maintain a discharge air setpoint of 55°F before and after reduction in steam pressure for the AHU in Alexandria (Design Airflow: 7,800 cfm).

Table 5: Comparison of Face/Bypass Flow Before and After Steam Pressure Reduction

Bin Data		**15psig steam**			**5psig steam**		
Average Dry Bulb Temp (F)	# of hours	Airflow Facing the Coil (cfm)	Airflow through Bypass (cfm)	% of Face to Bypass Flow	Airflow Facing the Coil (cfm)	Airflow through Bypass (cfm)	% of Face to Bypass Flow
20	2	6003	1797	77%	6657	1143	85%
24	14	5411	2389	69%	6012	1788	77%
28	114	4798	3002	62%	5342	2458	68%
32	223	4162	3638	53%	4644	3156	60%
36	220	3503	4297	45%	3916	3884	50%
40	201	2818	4982	36%	3158	4642	40%
44	394	2107	5693	27%	2366	5434	30%
48	489	1367	6433	18%	1540	6260	20%
52	598	598	7202	8%	675	7125	9%

The procedure followed to estimate the airflow for a face/bypass system before and after steam pressure reduction is provided in Appendix E.

Steam heating systems are usually encountered in older buildings, and are predominantly pneumatically controlled. Pneumatic controllers require regular maintenance work to ensure a clean and dry source of compressed air. If a proper maintenance schedule is not adopted by the facility, the pneumatic lines may be contaminated with oil or water which could hamper the performance of controllers (Gupton, 2001, Turner and Doty, 2009).

Lack of maintenance coupled with wear and tear of the pneumatic lines/actuators over time makes these controllers more susceptible to failure. These older buildings do not have a regular operation and maintenance program and most of the equipment is close to the end of its lifetime. Most of the pneumatic actuators observed in the VISN 16 hospitals are either stuck or don't respond to the control commands. In fact, one of the drawbacks of using an AHU with a modulating steam valve control is that the signal to the steam valves tends to fail due to the age of the controllers and actuators. In the event of actuator failure, a lower operating steam pressure can reduce the amount of simultaneous heating and cooling as discussed below.

According to the maintenance operators at the facility, the pneumatic signal to the steam valve tends to fail open around 10% of the time during a year. Since steam valves normally fail in the open position, it is estimated that simultaneous heating and cooling can be reduced by 81 MMBtu annually by lowering the steam pressure.

Table 6 provides a breakdown of heating and cooling energy for the two steam pressure conditions. This simultaneous heating and cooling energy waste can be brought down by 122 MMBtu and 162 MMBtu annually if the valve failure rate increases to 15% and 20% respectively.

If a face/bypass damper type steam heating system is employed and assuming the pneumatic signal fails with the damper position in 70% face mode, it is estimated that the amount of simultaneous heating and cooling can be reduced by 121 MMBtu/yr. by lowering the steam pressure from 15psig to 5psig. Table 7 provides a breakdown of heating and cooling energy for the two steam pressure system. The procedure used to estimate the energy savings is provided in Appendix E for reference.

Table 6: Comparison of Simultaneous Heating and Cooling for Modulating Steam Valve Coil

Bin Data		5 psig Steam			15 psig Steam		
Average OAT (F)	# of Hrs.	Average DAT(F)	Heating Energy Consumption (MMBtu)	Cooling Energy Consumption (MMBtu)	Average DAT (F)	Heating Energy Consumption (MMBtu)	Cooling Energy Consumption (MMBtu)
20	1	61	0	(0.0)	65	0	0.0
24	4	64	0	(0.0)	69	1	0.2
28	34	67	4	0.3	72	4	1.6
32	67	71	7	1.3	75	8	3.8
36	66	74	7	2.0	78	8	4.3
40	60	77	6	2.5	82	7	4.5
44	118	80	12	6.3	85	14	9.9
48	147	83	15	9.5	88	16	13.6
52	179	87	17	13.6	91	20	18.2
56	172	90	16	14.9	94	18	19.0
60	149	93	14	14.6	98	16	17.8
64	189	96	17	20.7	101	19	24.3
68	318	100	28	38.4	104	32	43.7
72	306	103	26	40.4	107	30	44.9
76	225	106	19	32.3	110	22	35.0
80	192	109	16	29.6	114	18	31.5
84	150	112	12	24.8	117	14	26.0
88	148	116	11	26.2	120	13	27.0
92	70	119	5	13.2	123	6	13.4
96	24	122	2	4.8	126	2	4.9
100	4	125	0	0.8	130	0	0.8
104	5	128	0	1.1	133	0	1.1
108	0	132	-	-	136	-	-
Total	**-**	**-**	**236**	**297**	**-**	**269**	**345**

Table 7: Comparison of Simultaneous Heating and Cooling for Integrated Face/Bypass Damper Coil

Bin Data		**5 psig Steam**			**15 psig Steam**		
Average OAT (F)	# of Hrs.	Average MAT(F)	Heating Energy Consumption (MMBtu)	Cooling Energy Consumption (MMBtu)	Average MAT (F)	Heating Energy Consumption (MMBtu)	Cooling Energy Consumption (MMBtu)
20	2	49	0	(0.1)	52	1	(0.1)
24	14	52	3	(0.3)	55	4	0.0
28	114	56	26	0.5	59	29	3.6
32	223	59	51	7.5	62	57	13.5
36	220	62	49	13.8	66	55	19.8
40	201	66	44	18.4	69	49	23.9
44	394	69	84	47.6	73	95	58.4
48	489	73	102	73.3	76	115	86.7
52	598	76	123	107.0	80	138	123.5
Total	**2255**	**-**	**483**	**268**	**-**	**542**	**329**

6.2 AHU with Face/Bypass Damper and Two-position Valve

An improved control sequence for a steam heating system equipped with a two-position valve and a face/bypass damper configuration was developed to provide a stable discharge air temperature. It also eliminates the hunting of preheat and chilled water valves which is a common issue with a steam heating system when not properly controlled. The sequence was tested on a dual duct AHU with a steam coil. The steam coil is provided with an integrated face/bypass damper arrangement (clam type dampers). The steam coil served as a heating coil for the hot deck of the AHU in addition to prevent the chilled water coil from freezing. A schematic of the AHU is shown in Figure 11.

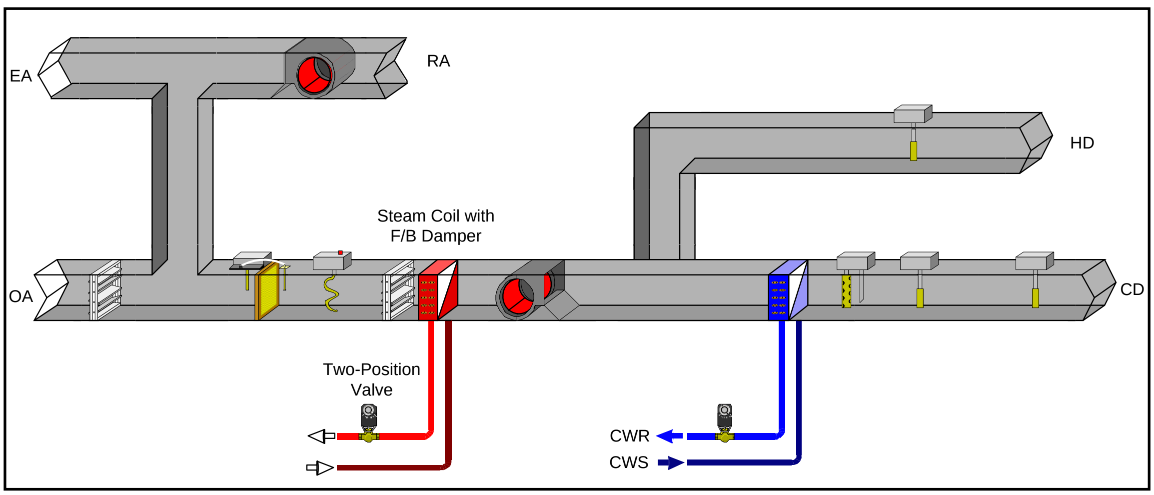

Figure 11: Schematic of AHU with Two-Position Steam Valve and F/B Damper

6.2.1 Existing Control Sequence

The existing control sequence used Proportional-Integral (PI) loops for temperature control of the preheat valve and face/bypass damper. The hot deck temperature setpoint was reset based on the coldest zone temperature. During the period of experimentation the hot deck temperature was set at 77°F. The preheat valve was turned on and the face/bypass damper was modulated to maintain the hot deck temperature with the help of a PI loop. When the air temperature reached the *hot deck setpoint* the preheat valve was turned off with a dead band of 5°F. This control strategy led to fluctuation of the cold and hot deck temperature and valve hunting. In order to eliminate the hunting of valves and to provide a stable temperature control, an improved control sequence was implemented. A detailed description of the control sequence for the valve and dampers is provided below.

6.2.2 Improved Control Sequence - Steam Valve Control

The steam coil serves as a heating source for the hot deck and the steam valve is commanded to be in the open position when the outside air temperature (OAT) is below 75°F. When the OAT is above 75°F, the steam valve is commanded OFF. A dead band of 2°F is provided to provide control stability for the valve. The OAT setpoint is adjustable and needs to be determined by the commissioning engineer based on the location of the facility and the area served by the AHU.

6.2.3 Improved Control Sequence - Face/Bypass Damper Control

The face/bypass dampers were either enabled or disabled based on the steam valve command. If the preheat steam valve is commanded to the fully closed position the face/bypass dampers were commanded to go to a fully BYPASS mode. This arrangement will prevent any unnecessary heating of the air stream if any steam is leaking through the valve. It also mitigates any radiant heat pick up from the coil. When the preheat valve is commanded to the fully open position, the face/bypass dampers are modulated to maintain the hot deck temperature setpoint. The dampers can be in either full face mode, bypass mode or in between in order to maintain the temperature setpoint. The various scenarios for the discharge air temperature and the subsequent control of the face/bypass damper are discussed below.

If the hot deck temperature (HDT) is lower than the setpoint by 5°F then the damper is set in FACE mode. If the HDT is equal to or higher than the setpoint, the FACE/BYPASS dampers should be modulated to maintain the hot deck temperature at its setpoint with the help of a PI loop. In addition, if the face/bypass damper is more than 80% (adjustable) in BYPASS mode for fifteen minutes to maintain the DAT setpoint, then turn OFF the steam valve.

A temperature dead band of +/-2°F (adjustable) was provided to prevent any valve hunting. The temperature measurements were carried out using an averaging type temperature sensor, which is a type of sensor that goes across the air stream and reads an average air stream temperature. Figure 12 and Figure 13 provide a thirty second trend of the critical parameters measured over a five hour period. The outside air temperature was trended with a one hour interval and is assumed to be constant during that one-hour period. Figure 12 provides the variation of the hot deck temperature before and after the improved CC® sequence was implemented while Figure 13 represents the cold deck discharge air temperature during that period.

The hot deck temperature initially varied from 52°F to 84°F and this variation was brought down to +/-5°F of the hot deck temperature setpoint. The cold deck temperature varied from 45°F to 55°F while the setpoint during that period was set at 50°F. The chilled water valve position varied from 0% to 60% open every three minutes due to the fluctuating temperature conditions at the exit of the steam coil. After the implementation of the improved control sequence, the cold deck temperature was maintained at 50°F with a +/- 1°F fluctuation while the control valve position varied from 45% to 50% during that period. This reduction in the cycling of the valve results in the following benefits:

I. Provides a stable discharge air temperature to the conditioned space, thereby eliminating any occupant discomfort.

II. Reduces the wear and tear on the valve seat, hence reducing maintenance cost, increasing lifetime of the valve.

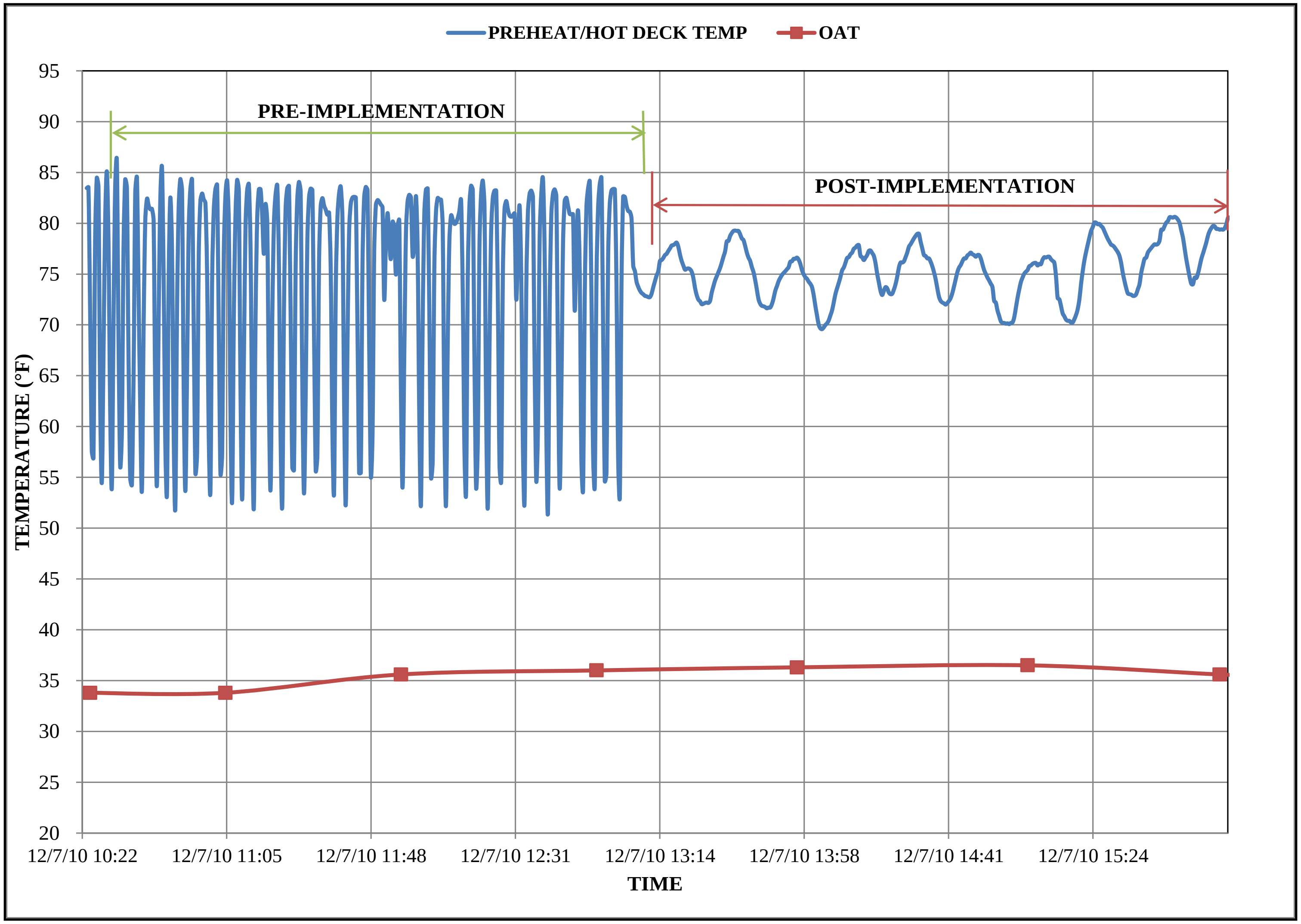

Figure 12: Preheat Temperature Plot for Improved Face/Bypass Control

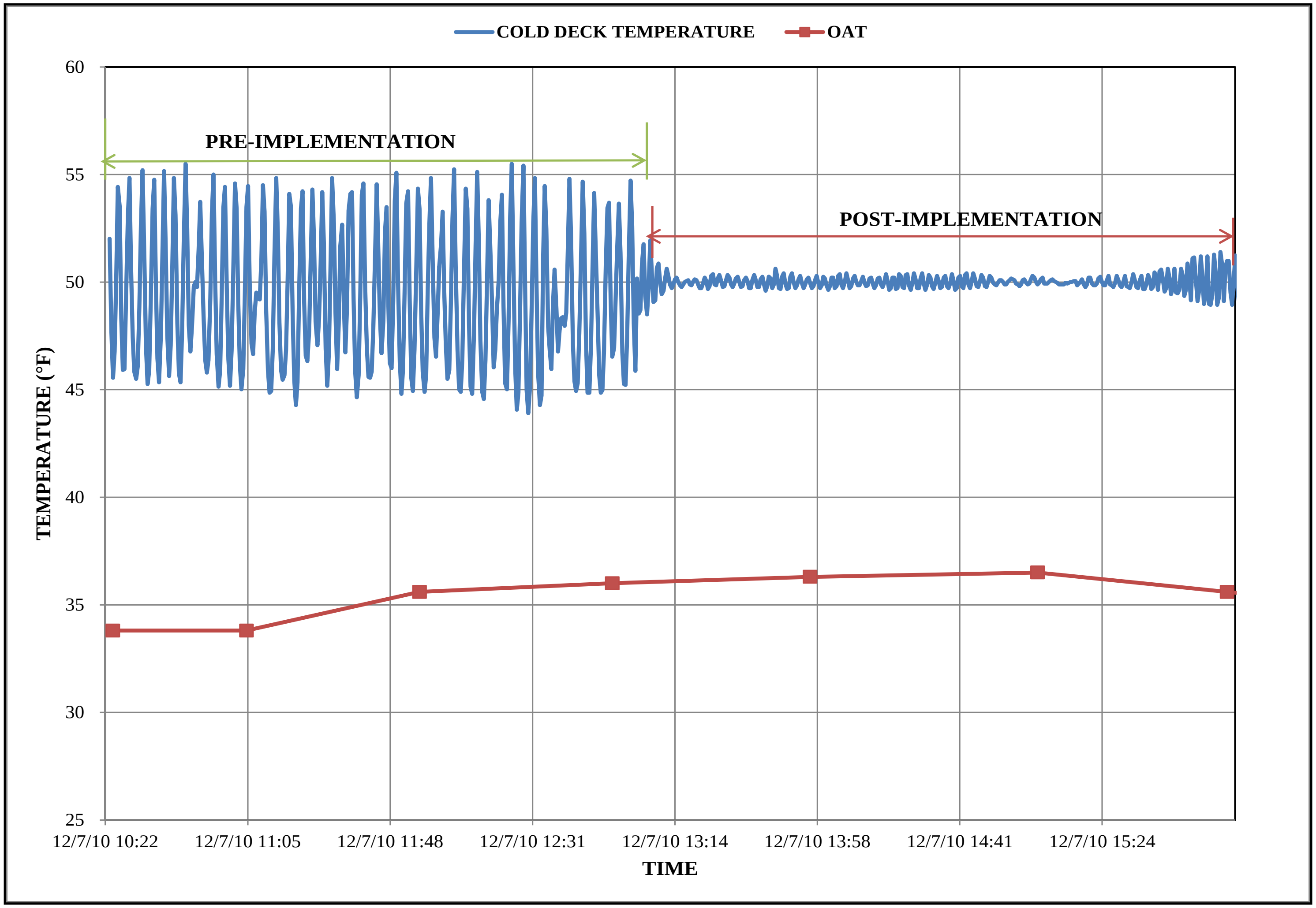

Figure 13: Cold Deck Temperature Plot for Improved Face/Bypass Control

6.3 AHU with Modulating Valve Configuration

An improved control sequence for a steam coil with a modulating valve was developed with the intention to optimize the energy consumption of the unit while preventing the freezing of the coils during cold outside air conditions. By implementing the improved control sequence it is estimated that annual energy savings of 114 MMBtu can be achieved.

6.3.1 Existing Control Sequence

The air handling unit in McDonald Army Community Hospital that serves the surgery area is a dedicated outside air unit and makes use of a $1/3^{rd}$ -$2/3^{rd}$ valve arrangement on the preheat coil. These valves are modulated through a PI loop to maintain the discharge air temperature at a fixed setpoint (48°F). The preheat setpoint is maintained at 45°F to account for the temperature rise across the supply fan (3°F). The steam is supplied at 10psig to the coil when the valve is in the fully open position.

A schematic of the steam coil is shown in Figure 14. When the outside air temperature (OAT) falls below 35°F the $1/3^{rd}$ valve is commanded to the fully open position as a safety measure. This is done to prevent a sub-atmospheric condition in the coil and subsequent freezing of the condensate. When the OAT reaches 15°F or below, both the valves are commanded fully open.

Table 8 summarizes the existing control sequence.

Figure 15 provides a 10-minute trend of the preheat air temperature, discharge air temperature, OAT and preheat valve position for the AHU. Based on the plots it can be observed that a large amount of simultaneous heating and cooling of air occurs when the outside air temperature falls below freezing.

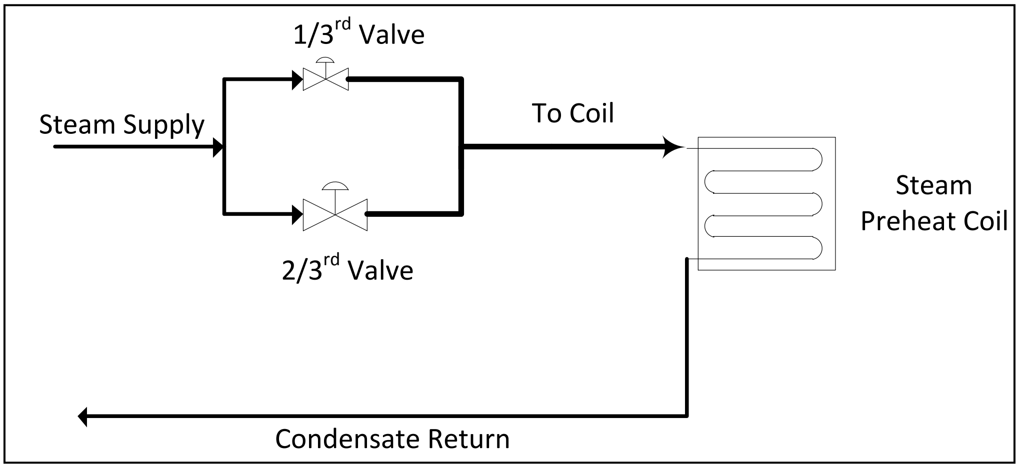

Figure 14: Schematic of the Steam Coil at McDonald Army Community Hospital, Fort Eustis, VA

Table 8 Existing Control Sequence for AHU with Modulating Valve

If OAT>35°F	Modulate $Valve_{1/3}$ & $Valve_{2/3}$
15°F<OAT<35°F	$Valve_{1/3}$:Fully Open, $Valve_{2/3}$:Modulate
OAT<15°F	$Valve_{1/3}$ & $Valve_{2/3}$-Fully Open

6.3.2 Improved Control Sequence

As discussed in Section 3.1, the mass flow rate of steam changes when a valve is modulated, resulting in variation of steam pressure within the coil. In locations where the outside air temperature falls below freezing, it is important that the modulation of steam valves be regulated to prevent sub-atmospheric conditions in the coil (gravity return system). Maintaining a positive differential pressure across the steam trap ensures continuous draining of condensed steam and prevents the coil from freezing. In addition, it also improves the heat transfer efficiency of the coil. An improved control sequence was developed keeping in mind these constraints.

This sequence will determine the operating steam pressure for various valve positions instead of commanding the valve to be fully open during cold outside air conditions. A valve position corresponding to a steam pressure of 2.5 psig (adjustable) is defined as the lowest limit to which the valve can modulate when OAT is less than 35°F. The chosen value of the steam pressure (2.5 psig in this case) must be sufficient to drain the condensate. The output of the preheat valve PI loop is then compared with the lowest limit described above and the higher of the two is chosen as the signal transmitted to the valve. Table 9 summarizes the control strategy for the modulating steam valve arrangement. This sequence will prevent freezing of the coil during cold weather and will also optimize the cooling/heating energy consumption for the unit.

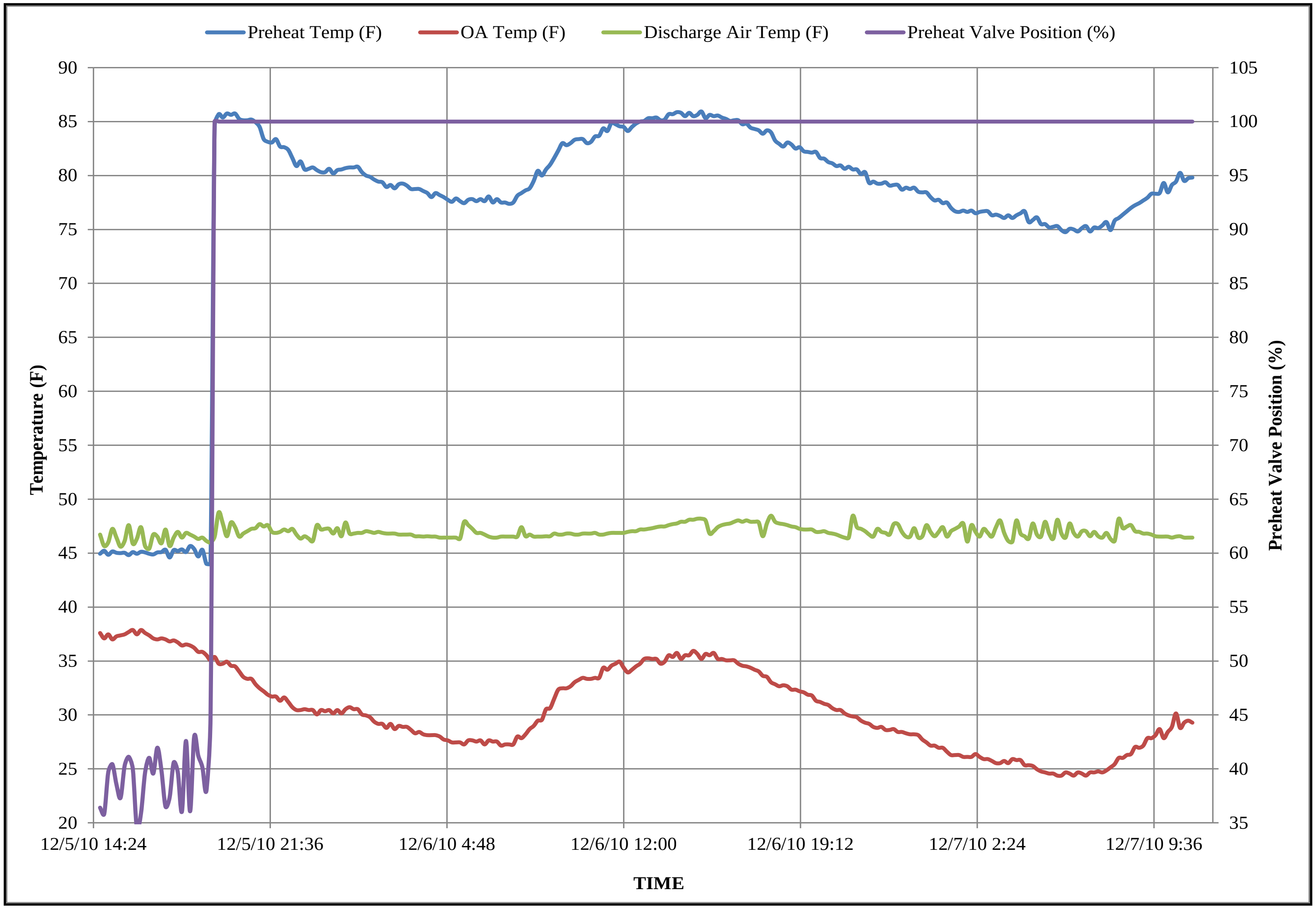

Figure 15: Temperature Plot for Modulating Valve Arrangements

The procedure to determine the steam pressure corresponding to various valve positions is provided below.

Table 9 Improved Control Sequence for AHU with Modulating Valve

<table>
<tr><td>If OAT>35°F</td><td>Modulate $Valve_{1/3}$ & $Valve_{2/3}$</td></tr>
<tr><td>15°F<OAT<35°F</td><td rowspan="2">$Valve_{1/3}$: Modulate valve with lower limit set such that positive pressure is maintained in the coil
$Valve_{2/3}$: Modulates to supply supplemental heating</td></tr>
<tr><td>If OAT<15°F</td></tr>
</table>

6.3.3 Procedure to Determine Pressure in Coil for Various Valve Positions

The heat transfer when air temperature is raised from T_1 to T_2 is given by,

$$\dot{Q} = \dot{m} * C_{Pa} * (T_2 - T_1) \qquad \textit{Eq. 6.1}$$

The heat transfer in a heat exchanger can also be denoted by,

$$\dot{Q} = U * A * \Delta T_{LMTD} \qquad \textit{Eq. 6.2}$$

By equating Eqs. 6.1 & 6.2 the log mean temperature difference for a given airflow rate can be calculated.

Since a steam coil can be compared to a condenser type heat exchanger, the saturation temperature of steam for a given inlet and exit condition of the air stream can be calculated from Log-Mean Temperature Differential (ΔT_{LMTD}) mentioned below

$$\Delta T_{LMTD} = (T_2 - T_1)/\ln\{\frac{T_s - T_2}{T_s - T_1}\} \qquad \textit{Eq. 6.3}$$

Based on the steam tables for a given saturation temperature the steam pressure can be computed.

In case the temperature rise of the air for various valve positions is not known, the steam pressure in the coil can be estimated based on the design parameters of the valve and mass of steam required to maintain the discharge air temperature. Assuming that steam enters as a saturated vapor and leaves the coil as saturated liquid, the mass flow rate of steam can be computed as,

$$\dot{m}_s = Q/h_{fg} \qquad \textit{Eq. 6.4}$$

For a valve designed for a critical pressure drop, the valve coefficient C_v for various steam flow rates is given by,

$$\mathrm{C_v}(\frac{\mathrm{lb}}{\mathrm{hr-psia}}) = \dot{\mathrm{m}}_\mathrm{s}/(1.61 * \mathrm{P_i}) \qquad \textit{Eq. 6.5}$$

For a valve designed for non-critical pressure drop, the valve coefficient is given by,

$$\mathrm{C_v}(\frac{\mathrm{lb}}{\mathrm{hr-psia}}) = \dot{\mathrm{m}}_\mathrm{s}/(2.1 * \left\{\sqrt{(\mathrm{P_i} + \mathrm{P_o})} * (\Delta\mathrm{P})\right\}) \qquad \textit{Eq. 6.6}$$

For various inlet air conditions we can estimate the steam flow required at the coil to maintain the discharge air temperature. The valve coefficient for various steam flow rates can be consequently determined. Based on the valve coefficient required and the design valve coefficient for the steam valve we can estimate the percentage of valve opening to maintain the heating setpoint,

$$\%\ of\ Valve\ Open = C_v/C_{v-d} \qquad \textit{Eq. 6.7}$$

For the specimen AHU, field work was carried out to determine the temperature rise across the coil for various valve positions. Table 10 provides the measured data for the unit. Since the heat transfer through steam coils varies linearly with the change in flow a first order curve was fit to the field measurements. Based on the curve temperature rise for various valve positions is determined. Figure 16 provides a plot of the temperature rise across the coil for various valve positions. The temperature measurements were carried out for an outside air temperature of 32.5°F.

Table 10: Temperature Rise for Various Valve Positions

Valve Position	Temperature Rise (°F)
30	5.0
40	8.0
60	21.0
70	28.0
100	50.4

Using the experimental data and the above mentioned procedure, the pressure in the coil is determined for various valve positions. Table 11 provides steam temperature for different valve positions. Based on the saturated steam tables, a valve position of 90% open corresponds to a steam pressure of 2.5 psig and is defined as the lowest limit for the valve during cold outside air conditions.

Bin data based on the TMY3 weather data for Fort Eustis, VA was generated. The energy consumption for the existing and improved control sequences was evaluated.

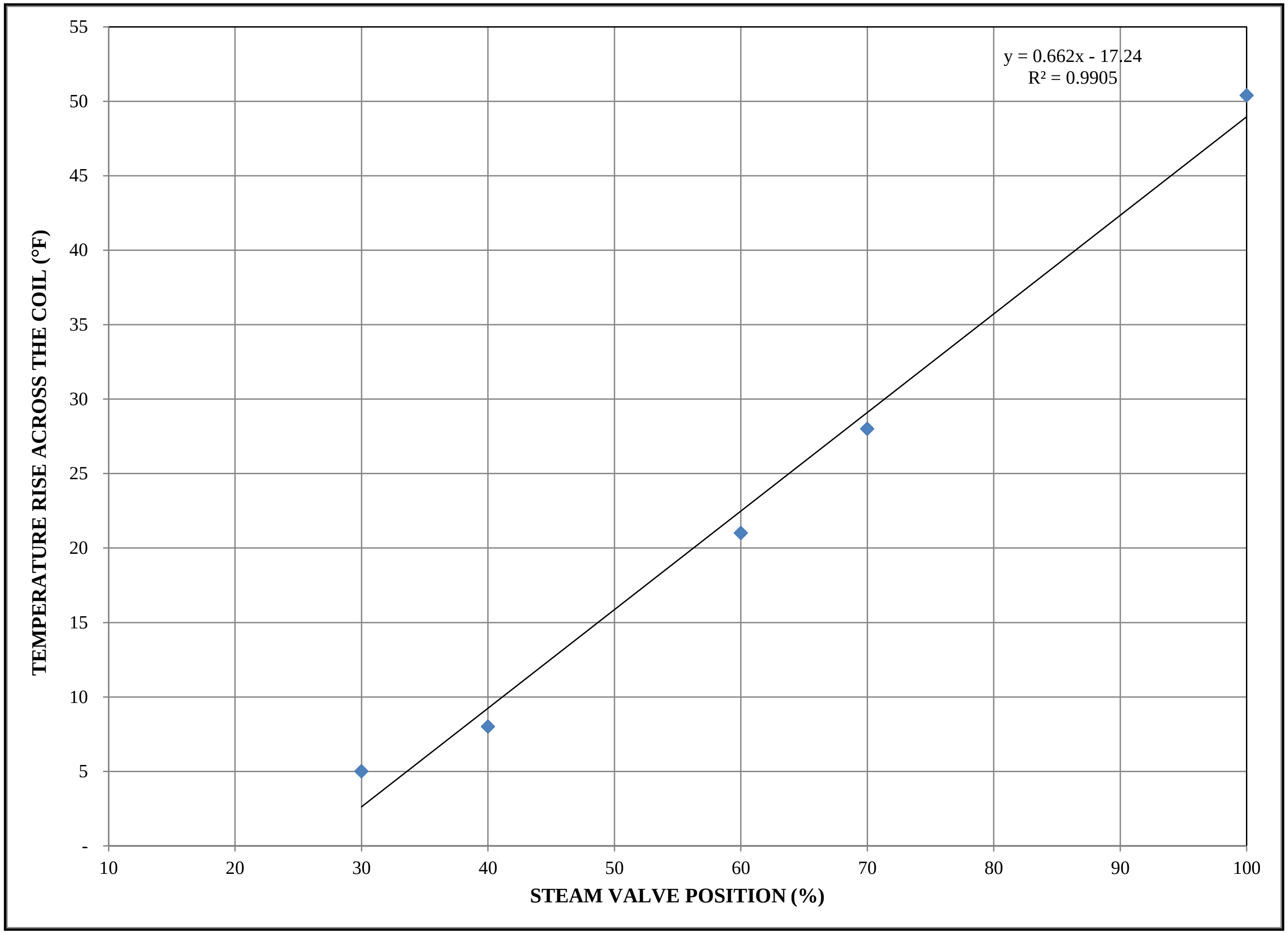

Figure 16: Temperature Plot for Modulating Valve Arrangement

It is estimated that a total energy reduction of 114 MMBtu/year can be realized by implementing the improved control sequence amounting to a 5% reduction in energy consumption annually. If a hot water system is used for preheat it will result in 420MMBtu of reduction in energy usage, amounting to a 23% reduction over improved control sequence.

Figure 17 and Figure 18 provide a comparison of total energy consumption for the three control methods, i.e., the existing control sequence, improved control sequence and the heating hot water control. The heating and cooling energy consumptions for the three configurations is calculated based on a number of operating hours in 5°F temperature bins for the temperature range of 5°F to 35°F. Table 12 and Table 13 provide the estimated energy consumption for various temperature bins for the above mentioned scenarios. Since the existing control sequence maintains the preheat temperature at its setpoint with no simultaneous heating and cooling when the outside air is above 35°F, the energy consumption for heating and cooling have been represented on the plots for the outside air temperature range of 5°F to 35°F. The procedure followed to estimate the energy savings is provided in Appendix E for reference.

Table 11: Steam Temperature for Various Valve Positions

Valve Position	Temperature Rise (°F)	LMTD	Steam Temperature
60	22.5	56	144.1
70	29.1	69	171.1
80	35.7	81	196.1
90	**42.3**	**91**	**219.3**
91	43.0	93	221.5
92	43.7	94	223.7

Table 11: Continued

Valve Position	Temperature Rise (°F)	LMTD	Steam Temperature
93	44.3	95	225.9
94	45.0	96	228.1
95	45.7	97	230.3
96	46.3	98	232.4
97	47.0	99	234.6
98	47.6	100	236.7
99	48.3	101	238.8
100	49.0	102	240.9

Even though a hot water system would completely eliminate simultaneous heating and cooling, it is not recommended to retrofit the air handling unit. Since the coil serves as preheat (for Fort Eustis, VA close to 8% of annual hours is below freezing condition), the coils would require an anti-freeze (glycol) mixed with the hot water to prevent freezing.

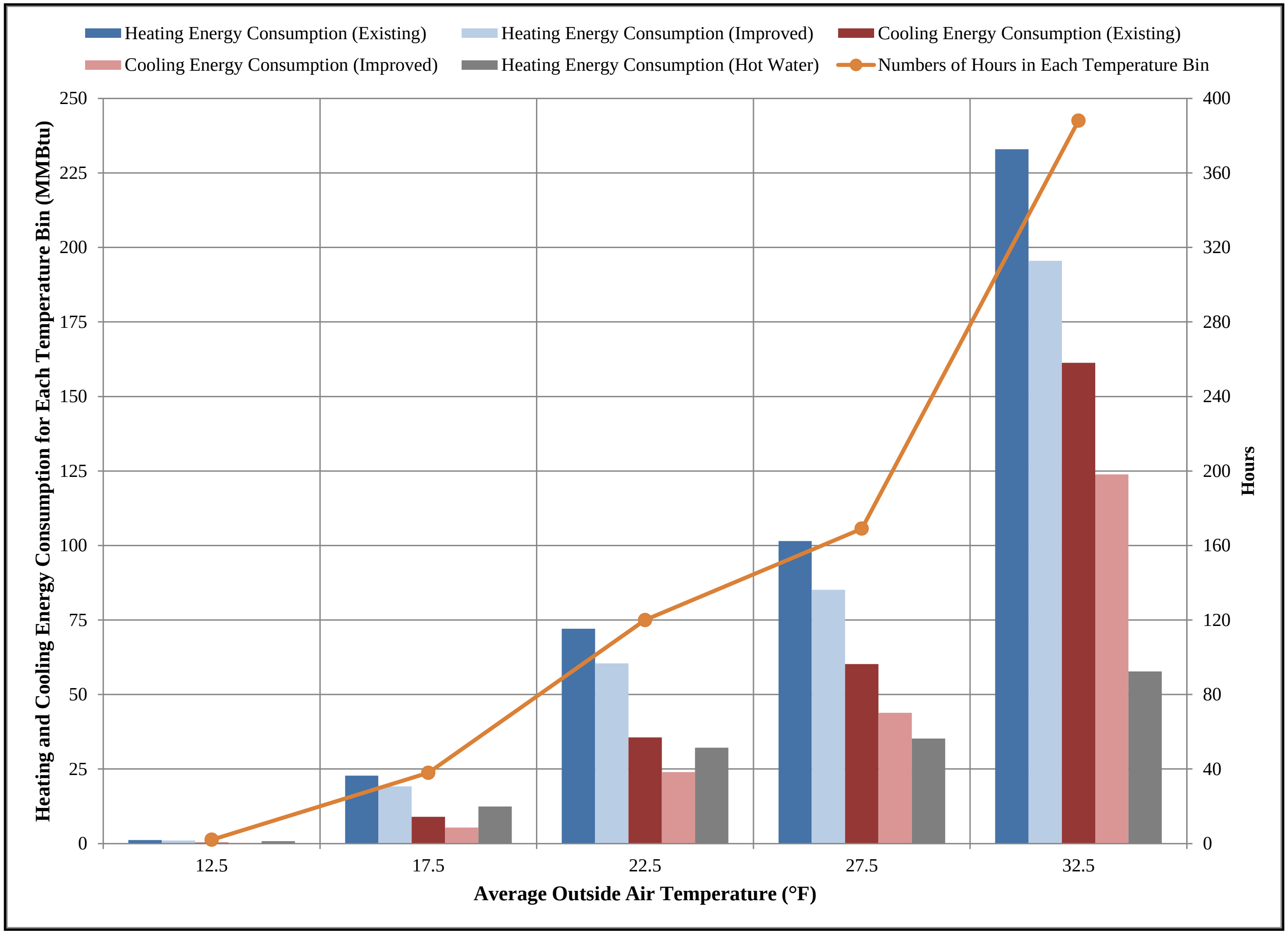

Figure 17: Comparison of Total Energy Consumption for Temperature Bins 5°F to 35°F

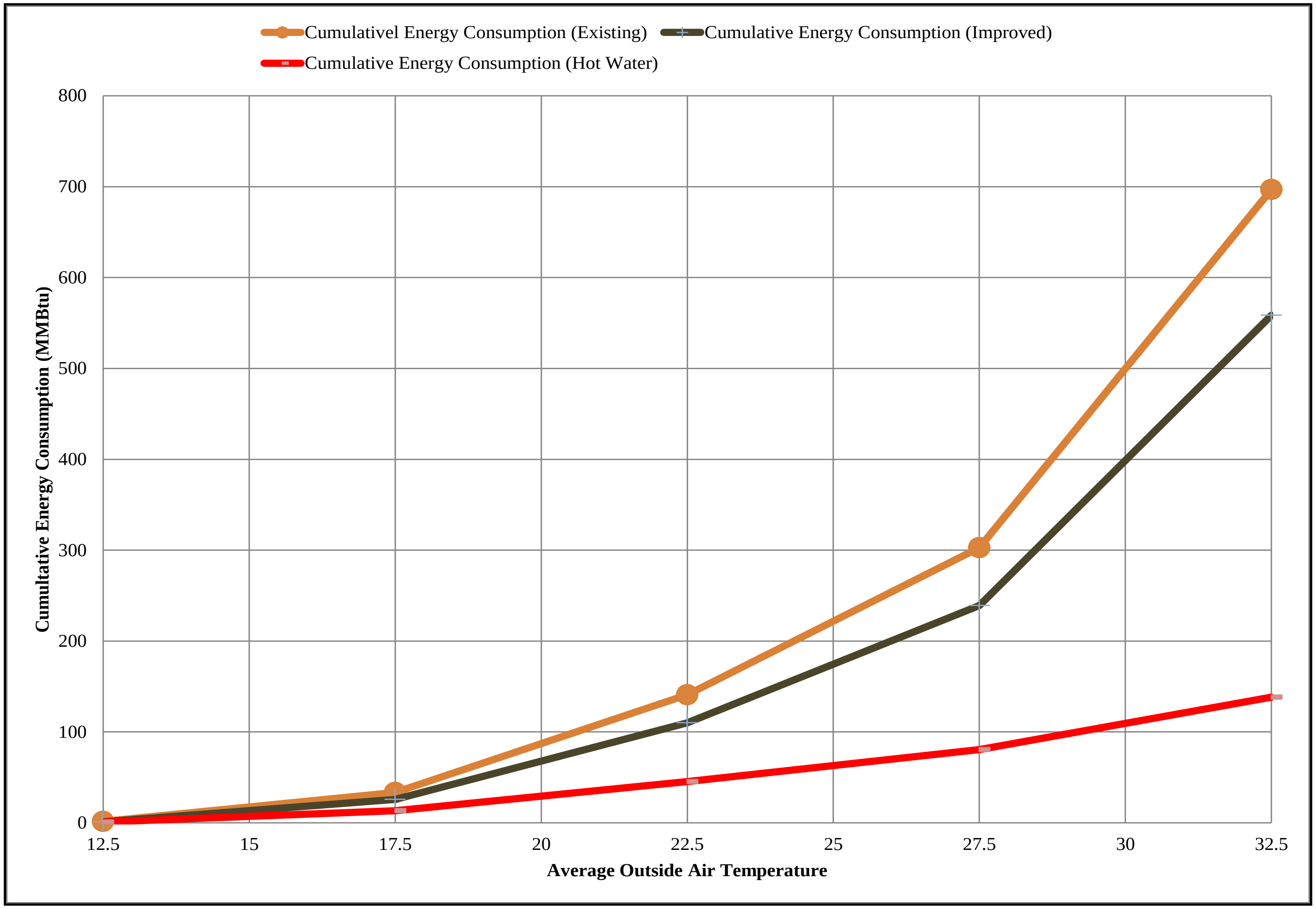

Figure 18: Cumulative Total Energy Consumption for Temperature Bins 5°F to 35°F

Table 12: Comparison of Energy Consumption for Existing and Improved Control Sequence/Hot Water

Dry Bulb Temp. Bin Data		# of hours	Average Temp. (F)	Original Control				Improved Control			
				Temp. Rise (F)	Heating (MMBtu)	Cooling (MMBtu)	Total Energy Consumption (MMBtu)	Temp. Rise (F)	Heating (MMBtu)	Cooling (MMBtu)	Total Energy (MMBtu)
5	10	0	7.5	-	0	0	0	-	0	0	0
1	15	2	12.5	49.0	1	0	1	42.3	1	0	1
1	20	38	17.5	49.0	22	8	31	42.3	19	5	24
2	25	120	22.5	49.0	70	34	104	42.3	60	24	84
2	30	169	27.5	49.0	99	57	156	42.3	85	44	129
3	35	388	32.5	49.0	226	155	381	42.3	196	124	319
3	40	621	37.5	7.5	55	0	55	7.5	55	0	55
4	45	624	42.5	2.5	19	0	19	2.5	19	0	19
4	50	633	47.5	0	0	0	0	0	0	0	0
5	55	801	52.5	0	0	43	43	0	0	43	43
5	60	631	57.5	0	0	71	71	0	0	71	71
6	65	990	62.5	0	0	171	171	0	0	171	171
6	70	841	67.5	0	0	195	195	0	0	195	195
7	75	789	72.5	0	0	230	230	0	0	230	230
7	80	915	77.5	0	0	322	322	0	0	322	322
8	85	687	82.5	0	0	282	282	0	0	282	282
8	90	373	87.5	0	0	176	176	0	0	176	176
9	95	101	92.5	0	0	54	54	0	0	54	54
9	100	37	97.5	0	0	22	22	0	0	22	22
Total		**8760**	**-**	**-**	**493**	**1820**	**2313**		**212**	**1566**	**1778**

Table 13: Comparison of Energy Consumption for Existing and Improved Control Sequence/Hot Water

Dry Bulb Temp. Bin Data		# of hours	Average Temp (F)	Heating Hot Water Control			
				Temperature Rise (F)	Heating Total (MMBtu)	Cooling Energy (MMBtu)	Total Energy (MMBtu)
5	10	0	7.5	-	0	0	0
10	15	2	12.5	32.5	1	0	1
15	20	38	17.5	27.5	12	0	12
20	25	120	22.5	22.5	32	0	32
25	30	169	27.5	17.5	35	0	35
30	35	388	32.5	12.5	58	0	58
35	40	621	37.5	7.5	55	0	55
40	45	624	42.5	2.5	19	0	19
45	50	633	47.5	0	0	0	0
50	55	801	52.5	0	0	43	43
55	60	631	57.5	0	0	71	71
60	65	990	62.5	0	0	171	171
65	70	841	67.5	0	0	195	195
70	75	789	72.5	0	0	230	230
75	80	915	77.5	0	0	322	322
80	85	687	82.5	0	0	282	282
85	90	373	87.5	0	0	176	176
90	95	101	92.5	0	0	54	54
95	100	37	97.5	0	0	22	22
Total		**8760**	**-**		**212**	**1566**	**1778**

A glycol heating system needs to be installed at the facility which would require cost investment. In addition to the cost, the various problems associated glycol are:

I. Glycol has an increased creep capacity compared to water, i.e., it has a greater rate of leaking through joints compared to water and hence requires Teflon coating at the joints to prevent the loss of water-glycol mixture.

II. Presence of dissolved solids (chlorine) would have a higher rate of corrosion in case of water-glycol systems compared to water. Dissolved solids also use up the inhibitors and buffers (inhibitors and buffers used in the water-glycol system to provide a protective film to the metal surface and maintain neural pH level of water) and further increase the corrosion of the heat exchanger.

III. A water-glycol system needs to be maintained oxygen free. If glycol oxidizes it turns into lactic acid which would corrode the coils. This could lead to coolant leakage, reduced heat transfer and clogging of cooling system passages (Miller, 2005).

IV. Glycol is a toxic fluid and its ingestion may lead to death. Proper care needs to be taken to avoid any leakage of the fluid into potable water supply.

V. If proper maintenance program is not followed, it may lead to breakdown of glycol and would require complete flushing of the fluid which is expensive.

VI. The heat transfer capability for a glycol hot water system is low since glycol has lower thermal conductivity, reduced heating capacity and reduced flow due to high viscosity compared to water only system. This would result in installing a hot water coil with larger surface area.

Due to these reasons it is not advised to replace the existing steam preheating system with hot water coil. The improved control strategy would help in preventing coil freeze and reduce the energy waste.

6.4 AHU with Face/Bypass Damper and Modulating Valve

The AHUs in Ireland Army Community Hospital are equipped with steam coils for preheating purposes. The steam flow to the coil is controlled through a modulating type control valve. The units are also provided with a face/bypass damper arrangement for better control over the discharge air temperature. The steam coil is used for preheating as well as the heating source for the AHUs.

In case of a recirculation unit the steam valve is controlled based on the mixed air temperature and in fresh air units the outside air temperature drives the steam valve. The face/bypass damper is modulated to maintain the discharge air temperature. Figure 19 & Figure 20 provides a schematic of the specimen AHUs.

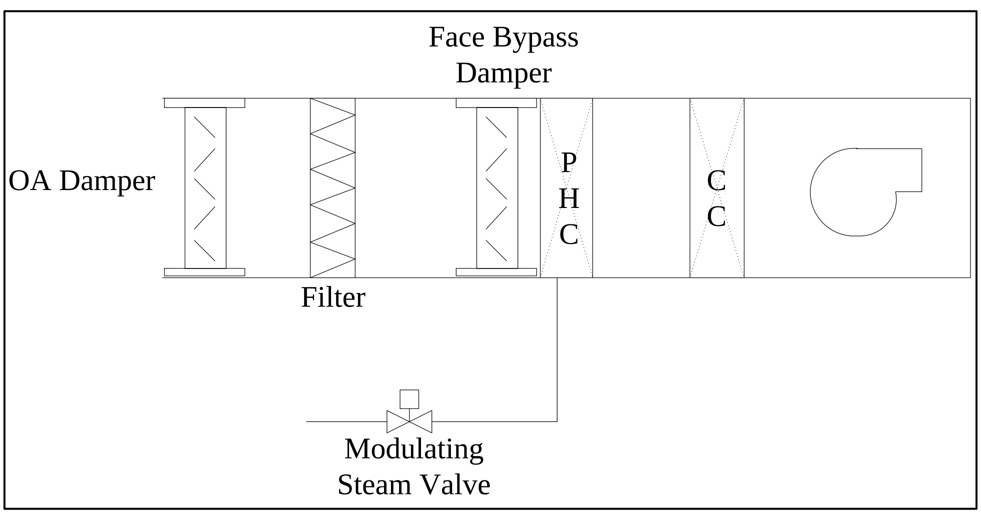

Figure 19: 100% OA Unit with Face/Bypass Damper & Modulating Steam Valve

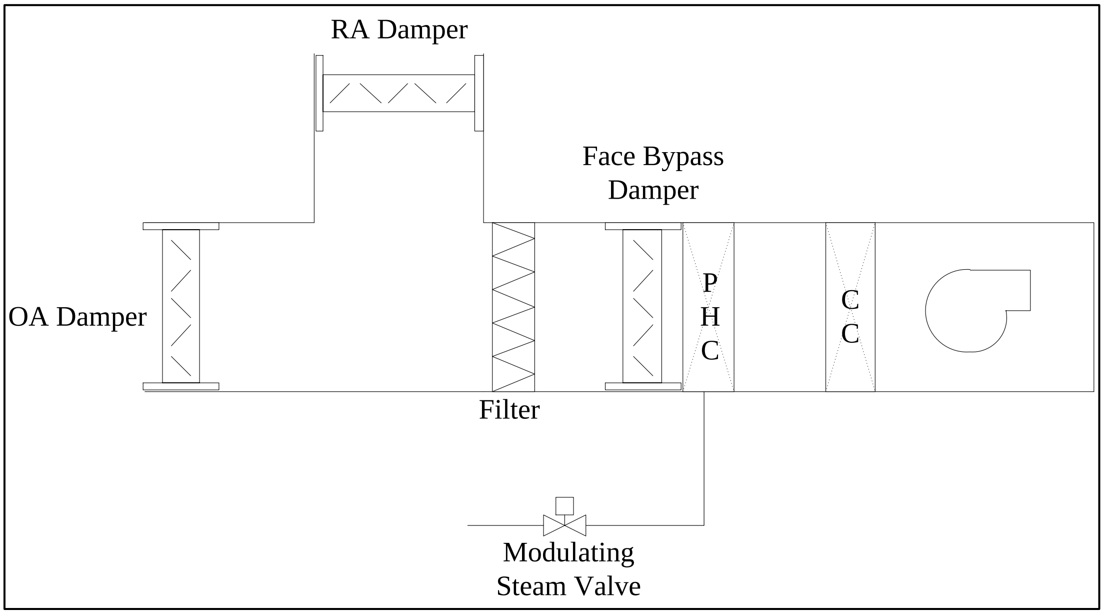

Figure 20: **Re-circulation type AHU with Face/Bypass Damper & Modulating Steam Valve**

6.4.1 Existing Control Sequence

The steam valve position is reset based on the mixed air temperature (MAT) or outside air temperature (OAT) depending on the unit type. The steam valve position is reset from 0% to 100% when the MAT/OAT varies from 52°F to 35°F. A 2°F dead band is provided to prevent hunting. The face/bypass damper was modulated to maintain the discharge air temperature. This control strategy leads to insufficient heating capacity at the coil resulting in the actual discharge air temperature being lower than the setpoint. Figure 21 provides a fifteen minute temperature trend for a four-hour period. The discharge air temperature was an average of 7°F lower than the setpoint.

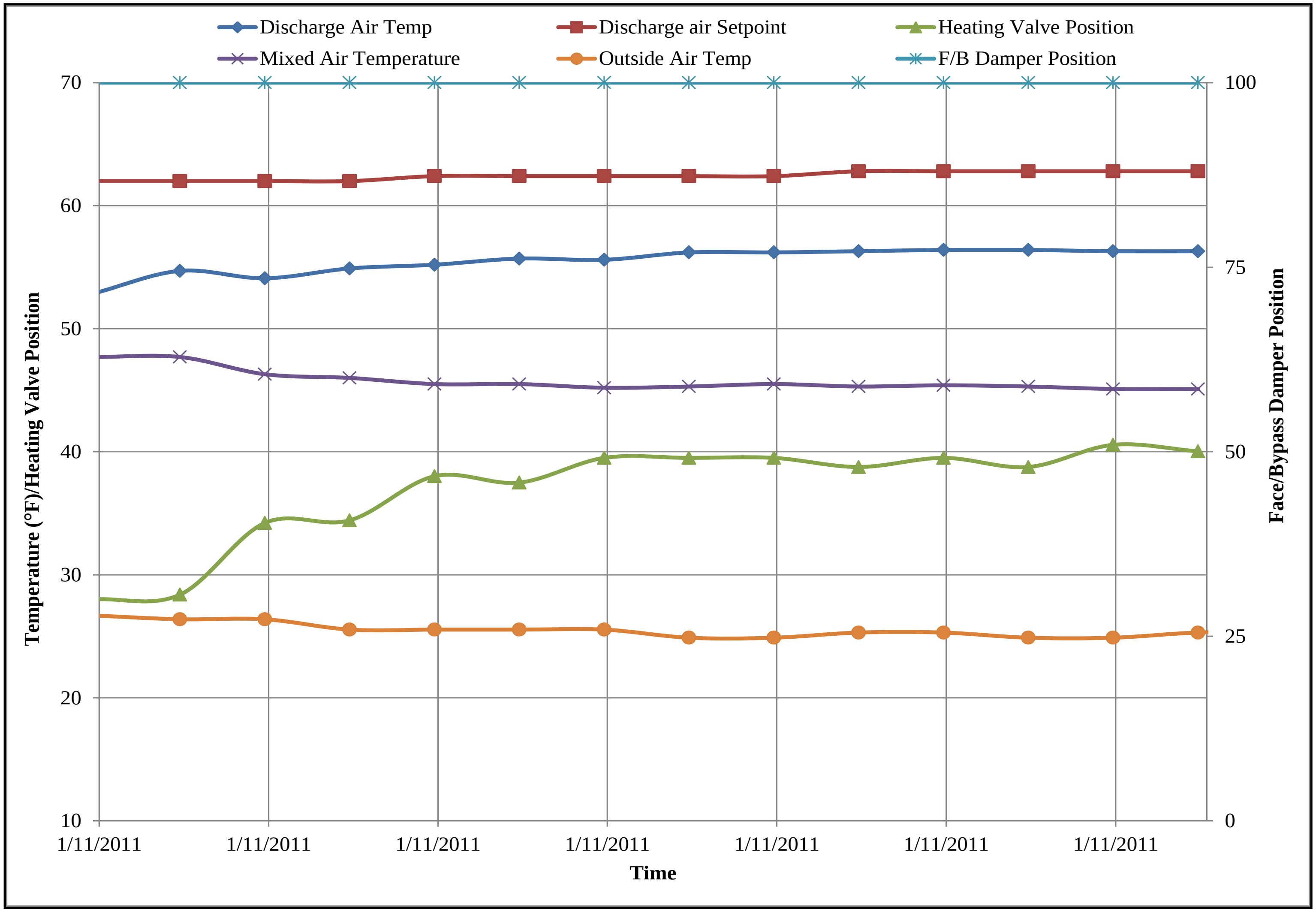

Figure 21: **Temperature Plot for Original Face/Bypass Control**

Since the control enabling the face/bypass damper is not linked to the steam valve activation there is a possibility that the dampers will either modulate or remain in the partially open position even when the steam valve is turned OFF. This could result in an increased restriction to the airflow and excess fan power consumption. Figure 21 provides fifteen minute interval trend data of the critical parameters. The outside air temperature varied from 25°F to 28°F at the time of field measurements. An improved control strategy was developed to resolve these issues and improve overall control of the system.

6.4.2 Steam Valve Control

The steam valve was controlled based on the mixed air temperature or outside air temperature depending on the type of unit. In case of AHUs with a return fan, the steam heating was only provided when the mixed air temperature was lower than a setpoint at a minimum outside air intake. This type of control will prevent any interference of the steam heating coil with the economizer operation.

In case of recirculation units the steam valve is turned ON and modulated to maintain the discharge air temperature with the help of a PI loop. If the outside air damper is in its minimum position and the mixed air temperature is 3°F (adjustable) lower than the discharge air temperature setpoint for the recirculation unit for more than fifteen minutes ***or*** if the outside air temperature is 3°F (adjustable) lower than the discharge air temperature setpoint for the fresh air units for more than fifteen ELSE the steam valve is to remain fully closed.

In case of recirculation units the steam valve is turned ON and modulated to maintain the discharge air temperature with the help of a PI loop. The steam loop is to remain fully closed if the outside air damper is in its minimum position and the mixed air temperature is 3°F (adjustable) lower than the discharge air temperature setpoint for the recirculation unit for more than fifteen minutes. It will also remain fully closed if the outside air temperature is 3°F (adjustable) lower than the discharge air temperature setpoint for the fresh air units for more than fifteen minutes.

If the outside air temperature is below 35°F, a lower limit for the steam valve is set to prevent sub-atmospheric conditions in the coil. This lock out will avoid accumulation of condensate in the coil and prevent the coil from freezing. The lower limit for the steam valve can be determined using the procedure described in Section 6.2.2.

6.4.3 Face/Bypass Damper Control

The face/bypass damper arrangement will be used only when the steam valve is at the lower limit position due to an outside air temperature lock out as described above. If the steam valve is at its lower limit and the discharge air temperature is greater than the setpoint, modulate the face/bypass damper to maintain the temperature with the help of a PI loop.

If the face/bypass damper is in full FACE mode and the discharge air temperature is 2°F (adjustable) lower than setpoint for fifteen minutes, the interlock on the steam valve is to be disabled. This control arrangement will prevent the overlap of the steam valve and face/bypass dampers and provides a stable control of the equipment.

If the steam preheat valve is OFF, the dampers are to be commanded to the BYPASS mode to reduce the flow restriction, reduce the radiant heat pick-up by the air stream and prevent any simultaneous heating and cooling due to leaking valves. Table 14 summarizes the improved control sequence for valve and face/bypass damper

The experimental results for this control strategy are not available at this time. Based on experiments carried out for a face/bypass damper and modulation valve configuration system at Fort Eustis as discussed in Sections 6.2 & 6.3, it can be concluded that this strategy will provide a good control for the heating system. The setpoints will require fine-tuning to maximize the control stability and reduced energy consumption.

Table 14: Improved Control Sequence for AHU with Face/Bypass Damper and Modulating Valve Arrangement

If OAT or MAT>35°F	Set F/B damper in 100% face mode, modulate steam valve
If OAT or MAT<35°F	Modulate steam valve with a low limit defined to maintain positive pressure, when valve reaches low limit modulate the F/B dampers to maintain setpoint If F/B damper in 80% bypass mode (adj.) turn off the valve

7 SUMMARY AND RECOMMENDATIONS

7.1 Summary

Current research involved documenting various types of steam heating systems prevalent in the commercial building sector. A brief description of the systems and their inherent characteristics has been provided to help educate the commissioning engineers. The research also looked into the effect of reduced steam pressure on the discharge air temperature and its benefits. Existing control strategies for various steam heating systems were analyzed, and improved control sequences developed. Stabilized temperature control and reduced energy consumption were the objectives of these improved strategies.

Experiments were carried out to estimate the effect of reduced operating pressure for the steam coils. Based on the data collected it is observed that the steam pressure reduction from 15psig to 5psig reduced the temperature rise across the coil by 16%. The coil with lowered steam pressure had enough capacity to prevent coil freezing even in the most severe weather condition. It is recommended that steam coils be operated at the lowest feasible pressure as it improves the system efficiency; reduces steam flashing/radiation losses, lowers maintenance cost and energy consumption.

In the case of a steam preheat system with a face/bypass damper arrangement; a lower steam pressure will result in a greater percentage of air passing through the coil thereby mitigating stratification of air.

For a built-in face/damper arrangement, the bypass area generally constitutes less than 20% of the total area of cross section available for the airflow through the coil. This result in more flow constriction compared to air flowing through the face damper/steam coil. A lower steam pressure translates to lower surface temperature of the coil resulting in larger portion of air flowing through the coil to maintain setpoint. Consequently this results in reduced fan power consumption for the AHU system due to lower restriction to airflow.

Proper care needs to be taken during operation of steam systems so that a sub-atmospheric condition in the coil is prevented. The extent to which the control valve can be modulated to avoid such conditions needs to be determined when any reduction in the supply pressure is carried out. Commissioning engineers can determine the feasibility of reducing the steam pressure on a heating system by using the following procedure:

I. Identify the type of steam heating system and the current steam pressure supplied to the coil.

II. Perform field tests to determine the temperature rise across the coil for the various steam pressures values.

III. Collect the location's TMY3 weather data and bin the data to identify the minimum pressure required at the coil to maintain the preheat/discharge air temperature for 99% of the total heating season. The commissioning engineer needs to make sure the operating pressure is chosen so sub-atmospheric conditions are prevented during low outside air temperature.

In addition to determining the feasibility of reducing steam pressure for a steam coil and evaluating its benefits, the current work also focused on the reviewing various control strategies employed by facilities to operate steam heating systems. Various steam heating systems were analyzed and the shortcomings of these control methods were identified. The improved control sequences developed during the course of this research addressed the shortcomings, such as valve hunting, control instability and insufficient heating. A standardized operating sequence developed for these systems is provided below. These sequences can be used for various configurations of steam heating systems by making the necessary adjustments to the basic control strategy.

7.1.1 Standard Operating Sequence for Steam Coil with Integrated Face/Bypass Dampers and Two-position Control Valve

I. The face/bypass damper should be enabled and disabled based on the steam valve position. If the steam valve is turned off, command the dampers to full bypass mode to prevent radiant heating in case of leaking valves. In case of built-in face/bypass dampers, the dampers are to be commanded to face mode if the steam valve is turned off. This will reduce the restriction of the airflow.

II. The steam valve should be controlled based on the outside air temperature or mixed air temperature depending on the type of unit and if the steam coil serves as preheat only or for heating purpose as well.

   a. If the coil is for only preheating purpose, In case of 100% outside air unit enable the steam heating if the outside air temperature is below 35°F. For return air type unit enable the steam preheat coil if the mixed air temperature is below 35°F (adjustable). In case of AHUs where the steam coil serves as preheating and heating purpose, the steam coil is enabled as mentioned below.

b. For a return type unit, enable steam heating if the mixed air temperature is lower than setpoint by 5°F (adjustable) for more than fifteen minutes (adjustable) with the outside air dampers in minimum position. In case of a 100% outside air unit, enable steam heating if the outside air temperature is 3°F below the discharge air.

III. If the discharge air temperature is lower than the setpoint by 5°F (adjustable), set the damper in full face mode. If the discharge air temperature is equal to or higher than setpoint, modulate the face/bypass damper.

IV. If the face/bypass dampers are more than 80% (adjustable) in bypass mode for more than fifteen minutes, turn off the steam valve.

V. A dead band of +/-2°F can be given for control stability.

7.1.2 Standard Operating Sequence for Steam Coil with a Modulating Steam Control Valve

I. The steam valve should be controlled based on the outside air temperature or mixed air temperature depending on the type of unit and if the steam coil serves as preheat only or for heating purposes as well.

a. If the coil is for preheating purposes only, as in the case of 100% outside air unit, enable the steam heating if the outside air temperature is below 35°F (adjustable). For return air type unit, enable the steam preheat coil if the mixed air temperature is below 35°F (adjustable). In case of AHUs where the steam coil serves as preheating and heating purposes, the steam coil should be enabled as mentioned below.

b. For a return type unit enable steam heating if the mixed air temperature is lower than setpoint by 5°F (adjustable) for more than fifteen minutes (adjustable) with the outside air dampers in minimum position or if outside air temperature is 3°F (adjustable) below the discharge air setpoint in case of 100% outside air unit enable steam heating.

II. The steam valve should be modulated using a PI loop to maintain the discharge air temperature at its setpoint. To prevent coil from freezing when the outside air temperatures is below freezing, a lower limit on the steam valve should be determined. This will prevent the sub-atmospheric condition in the coil and eliminate the freezing of condensate. The minimum position should be determined based on the valve specifications and coil heating capacity.

III. If the outside air temperature is below 35°F (adjustable) , the output of the PI loop for the steam valve is compared with the lowest limit determined and the higher of the two is selected as the valve output.

IV. A dead band of +/-2°F can be given for control stability.

7.1.3 Standard Operating Sequence for Steam Coil with Integrated Face/Bypass Dampers and Modulating Control Valve

I. The steam valve should be controlled based on the outside air temperature or mixed air temperature depending on the type of unit and if the steam coil serves as preheat only or for heating purpose as well.

   a. If the coil is for only preheating purpose, as in the case of a 100% outside air unit, the steam heating should be enabled if the outside air temperature is below 35°F (adjustable). For a return air type unit, enable the steam preheat coil if the mixed air temperature is below 35°F (adjustable). In case of AHUs where the steam coil serves as preheating and heating purpose, the steam coil is enabled as mentioned below.

   b. For a return type unit, enable steam heating if the mixed air temperature is lower than setpoint by 5°F (adjustable) for more than fifteen minutes (adjustable) with the outside air dampers in minimum position or if outside air temperature is 3°F (adjustable) below the discharge air setpoint. In the case of 100% outside air unit, enable steam heating.

II. If the steam valve is enabled maintain the face/bypass damper in full face mode. If the steam valve is turned off, position the dampers to full bypass mode to prevent any unnecessary heating due to leaking valves.

III. The steam valve should be modulated to maintain the discharge air temperature at its setpoint. To prevent the coil freezing when the outside air temperatures are below freezing, a lower limit on the steam valve is provided and the valve is locked at this position. This will prevent the sub-atmospheric condition in the coil and eliminate freezing of condensate. The minimum position should be determined based on the valve specifications and coil heating capacity.

IV. If the outside air temperature is below 35°F (adjustable), the steam valve is at its lock-out position and the discharge air temperature is greater than setpoint, modulate the face/bypass damper to maintain setpoint.

V. If the steam valve is in its lock-out position, the face/damper is in full face mode and the discharge air temperature is 3°F (adjustable) lower than setpoint for more than fifteen minutes disable the lock-out on the steam valve.

VI. A dead band of +/-2°F can be given for control stability.

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APPENDIX A

FIELD MEASUREMENT INSTRUMENTATION

The temperature measurements were carried out by Hobo TMC-HD series and Fluke 80PK-24 temperature sensors shown in Figure 22. The data was collected using the TSI VelociCalc 9555 and HOBO U12 data loggers shown in Figure 23. A detailed measurement specification for the equipment is provided in Table 15.

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Figure 22: TMCX-HD (Left) and Fluke 80PK-24 (Right) Temperature Probes (Onset Computer Corporation and Fluke Corporation, 2010)

Figure 23: TSI VelociCalc 9555 (Left) and Hobo U12 Data Logger (Right) (TSI Performance Measurement Tools & Onset Computer Corporation, 2010)

Table 15: Measurement Specifications for Temperature Probes

Parameter	Hobo TMC-HD	Fluke 80PK-24
Range	-40° to 212°F in air	-40 °F to 559.4 °F
Accuracy	±0.9° at 68°F	±2.2 °C
Resolution	0.05°F at 68°F	
Response Time	< 3 minutes typical to 90% in 1 m/sec air flow	3.0 seconds in 100 °C air moving at 3.33 meters/second (10.9 feet/ second) at sea level pressure (5 time constants = 1 complete step change, i.e., 15 seconds).

APPENDIX B

CRITICAL PRESSURE FOR STEAM

The mass flow rate of steam through a control valve is dependent on the pressure differential occurring across the control valve. The mass flow of steam through a valve increases as the differential pressure is increased until a condition referred to as *critical pressure* is reached.

The phenomena of critical pressure can be explained by understanding the flow of fluid through a convergent-divergent nozzle. A control valve is in principle similar to a convergent-divergent nozzle with difference being the purpose they serve. A nozzle is used to increase the velocity of steam to produce work while a control valve is used as a throttling device to produce a significant pressure drop.

The inlet of the control valve can be compared to the high pressure region. The inlet area between the valve plug and its seat may be viewed as the convergent area, with the narrowest region between the valve plug and seat as the throat and the outlet area as the low pressure region of a convergent-divergent nozzle.

In a convergent-divergent nozzle, high pressure steam enters through one end and leaves the nozzle at a lower pressure at the other end. In the convergent section of the nozzle, the steam velocity increases due to the reduction in cross-sectional area resulting in reduction of steam pressure. Though specific volume also increases with a lowered steam pressure, the rate of increase in steam velocity is much greater than the rate of change of specific volume; consequently the required flow area through this part of the nozzle becomes less. At a certain point the specific volume begins to increase more rapidly than the velocity, and the required flow area must increase. At this point the steam velocity is sonic and the flow area is at its minimum. The steam pressure at this minimum area referred to as the *throat* is described as the *critical pressure*. The ratio of absolute pressure at the throat to the inlet steam pressure is found to be close to 0.58 in case of saturated steam (Spirax-Sarco Limited, 2005). This same principle can be applied to a control valve.

If the valve is designed for a pressure drop greater than the critical pressure drop, the critical pressure occurs at the throat. The steam will expand after passing the throat such that if the outlet area has been correctly sized, the required downstream pressure is achieved.

Should the nozzle outlet be either too large or small, turbulence will occur at the nozzle outlet thereby reducing the capacity of the valve and increased noise.

I. If the nozzle outlet is too small, the steam has not expanded enough and has to continue expanding outside the nozzle until it reaches the required downstream pressure.

II. If the nozzle outlet is too large, the steam will expand too far in the nozzle and steam pressure exiting the nozzle is lower than required. This causes the steam to recompress outside the outlet of low pressure region.

The mass flow rate of steam is greatest when the designed steam pressure drop at the throat equals the critical pressure drop. Further reduction of downstream pressure doesn’t affect the mass flow rate. For a given coil capacity, choosing a control valve which will provide a critical pressure drop will result in reduced valve size and provide a linear characteristic.

APPENDIX C

BIN DATA FOR FORT EUSTIS, VA

Table 16: Bin Data for Fort Eustis, VA

Dry Bulb Temp. Bins		Totals	
		# of hours	Mean Coincident Wet Bulb Temp.
5	10	0	
10	15	2	13
15	20	38	17
20	25	120	20
25	30	169	25
30	35	388	30
35	40	621	34
40	45	624	39
45	50	633	43
50	55	801	48
55	60	631	52
60	65	990	58
65	70	841	62
70	75	789	68
75	80	915	70
80	85	687	72
85	90	373	74
90	95	101	77
95	100	37	78
100	105	0	

APPENDIX D

FIELD MEASUREMENTS

Table 17: Reduced Steam Pressure Trend Data, VAMC, Alexandria, LA

OAT (F)	Preheat Temp. (F)	OAT (F)	Preheat Temp. (F)	OAT (F)	Preheat Temp. (F)
68.2	104.4	68.1	104.1	68.1	104.4
68.2	104.2	68.1	104.5	68.1	104.2
68.2	104.4	68.1	104.4	68.1	104.7
68.2	104.6	68.1	104.4	68.1	104.8
68.2	104.1	68.1	104.6	68.1	104.6
68.1	104.7	68.1	104.8	68.1	104.4
68.1	104.4	68.1	104.4	68.1	104.4
68.1	104.1	68.1	104.3	68.1	104.9
68.1	104.2	68.1	104.5	68.1	104.4
68.1	104.5	68.1	104.5	68.1	104.2
68.1	104.2	68.1	104.7	68.1	104.4
68.1	104.3	68.1	104.5	68.1	104.2
68.1	104.5	68.1	104.6	68.1	104.6
68.1	104.3	68.1	104.6	68.1	104.2
68.1	104.1	68.1	104.5	68.1	104.3
68.1	104.2	68.1	104.4	68.1	104.5
68.1	104.2	68.1	104.4	68.1	104.6
68.1	104.2	68.1	104.6	68.1	104.3
68.1	104.2	68.1	104.7	68.2	104.6
68.1	104.6	68.1	104.2	68.2	104.9
68.1	104.2	68.1	104.2	68.2	104.5
68.1	104.1	68.1	104.6	68.2	104.5
68.1	104.7	68.1	104.2	68.1	104.7
68.1	104.5	68.1	104.2	68.2	104.7
68.1	104.2	68.1	104.0	68.1	104.6
68.1	104.5	68.1	103.9	68.2	104.1
68.1	104.6	68.1	104.1	68.2	104.0
68.2	104.5	68.1	104.4	68.2	104.4
68.2	104.1	68.1	104.5	68.2	104.5
68.1	104.4	68.1	104.2	68.2	104.4

OAT (F)	Preheat Temp. (F)
68.2	104.1
68.2	104.1
68.2	104.0
68.2	104.1
68.2	104.6
68.2	104.2
68.2	103.9
68.2	104.1
68.2	104.2
68.2	104.4
68.2	104.1
68.2	104.2
68.2	103.8
68.2	104.5
68.2	104.2
68.2	104.7
68.2	104.3
68.2	104.3
68.2	104.0
68.2	104.2
68.2	103.9
68.2	103.9
68.2	104.0
68.2	104.1
68.2	104.2
68.2	104.3
68.2	104.3
68.2	104.3
68.2	104.2
68.3	104.1
68.3	104.3
68.3	104.2
68.3	104.0
68.3	104.0
68.3	104.0
68.3	104.2
68.3	104.2

OAT (F)	Preheat Temp. (F)
68.3	103.8
68.3	104.0
68.3	104.0
68.3	104.3
68.3	104.0
68.3	104.3
68.3	104.3
68.3	104.4
68.3	104.2
68.3	104.3
68.3	104.4
68.3	104.2
68.3	104.3
68.3	104.1
68.3	104.1
68.3	104.2
68.3	103.8
68.3	103.8
68.3	104.1
68.3	104.4
68.3	104.0
68.3	103.8
68.3	103.8
68.3	104.3
68.3	103.8
68.3	103.9
68.3	103.9
68.3	104.3
68.3	103.9
68.3	103.8
68.3	103.9
68.3	103.9
68.3	104.0
68.3	103.8
68.3	103.8
68.3	104.2
68.3	103.9

OAT (F)	Preheat Temp. (F)
68.3	103.8
68.3	103.8
68.3	104.0
68.3	104.1
68.3	103.8
68.3	103.8
68.3	103.9
68.3	103.9
68.3	104.0
68.3	104.2
68.3	104.1
68.3	104.0
68.3	104.0
68.3	103.9
68.3	104.1
68.3	103.8
68.3	103.7
68.3	98.1
68.4	98.2
68.3	98.0
68.3	98.4
68.4	98.4
68.4	98.2
68.3	98.0
68.3	98.0
68.3	98.3
68.3	98.1
68.3	98.4
68.3	98.5
68.3	98.3
68.4	98.1
68.4	98.0
68.4	98.1
68.4	98.4
68.4	98.2
68.4	98.1
68.4	98.3

OAT (F)	Preheat Temp. (F)
68.4	98.1
68.4	98.2
68.4	98.3
68.4	98.2
68.4	98.2
68.4	98.2
68.4	98.2
68.4	97.9
68.4	98.1
68.4	98.0
68.4	98.2
68.4	97.9
68.4	98.0
68.4	98.3
68.4	98.1
68.4	97.9
68.5	98.1
68.5	98.1
68.5	98.2
68.5	98.0
68.5	97.9
68.5	98.0
68.5	98.2
68.5	98.4
68.5	98.2
68.5	98.0
68.5	97.9
68.5	98.0
68.5	98.2
68.5	98.3
68.5	98.0
68.5	98.1
68.5	98.5
68.5	98.3
68.5	98.1
68.5	98.3
68.5	98.5

OAT (F)	Preheat Temp. (F)
68.5	98.3
68.5	98.2
68.6	98.3
68.5	98.3
68.5	98.2
68.5	98.2
68.6	98.3
68.6	98.2
68.6	98.1
68.5	98.2
68.6	98.3
68.6	98.3
68.6	98.2
68.6	98.1
68.6	98.3
68.6	98.6
68.6	98.4
68.6	98.3
68.6	98.3
68.7	98.3
68.7	98.2
68.7	98.3
68.7	98.4
68.7	98.3
68.7	98.2
68.7	98.3
68.8	98.1
68.8	98.4
68.8	98.2
68.8	98.5
68.8	98.3
68.8	98.1
68.8	98.2
68.8	98.2
68.9	98.1
68.9	98.4
68.9	98.4

OAT (F)	Preheat Temp. (F)
68.9	98.4
68.9	98.5
68.9	98.5
68.9	98.4
68.9	98.1
69.0	98.2
69.0	98.2
69.0	98.3
69.0	98.1
69.0	98.4
69.0	98.6
69.0	98.4
69.0	98.3
69.1	98.2
69.1	98.3
69.1	98.5
69.1	98.3
69.1	98.1
69.1	98.3
69.1	98.6
69.1	98.4
69.1	98.4
69.1	98.4
69.1	98.5
69.1	98.5
69.1	98.5
69.1	98.5
69.1	98.5
69.1	98.2
69.1	98.5

Table 18: F/B Damper and Two Position Valve Trend Data, McDonald Army Community Hospital, Fort Eustis, VA

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 2:35:26 PM	79.3	52.0	36.5
12/7/10 2:35:56 PM	83.4	47.7	36.5
12/7/10 2:36:26 PM	84.3	45.5	36.5
12/7/10 2:36:56 PM	69.8	46.8	36.5
12/7/10 2:37:26 PM	55.4	50.2	36.5
12/7/10 2:37:56 PM	57.4	54.4	36.5
12/7/10 2:38:26 PM	76.9	53.5	36.5
12/7/10 2:38:56 PM	82.2	48.3	36.5
12/7/10 2:39:26 PM	82.9	45.9	36.5
12/7/10 2:39:56 PM	82.3	46.0	36.5
12/7/10 2:40:26 PM	82.0	49.3	36.5
12/7/10 2:40:56 PM	75.1	53.1	36.5
12/7/10 2:41:26 PM	59.0	54.8	36.5
12/7/10 2:41:56 PM	53.7	50.5	36.5
12/7/10 2:42:26 PM	74.2	47.5	36.5
12/7/10 2:42:56 PM	81.3	45.8	36.5
12/7/10 2:43:26 PM	83.4	45.5	36.5
12/7/10 2:43:56 PM	83.8	46.1	36.5
12/7/10 2:44:26 PM	69.6	48.8	36.5
12/7/10 2:44:56 PM	55.3	52.6	36.5
12/7/10 2:45:26 PM	57.3	55.2	36.5
12/7/10 2:45:56 PM	76.9	51.1	36.5
12/7/10 2:46:26 PM	82.8	47.0	36.5
12/7/10 2:46:56 PM	84.2	45.3	36.5
12/7/10 2:47:26 PM	79.9	45.9	36.5
12/7/10 2:47:56 PM	61.4	49.2	36.5
12/7/10 2:48:26 PM	52.7	53.4	36.5
12/7/10 2:48:56 PM	70.7	54.8	36.5
12/7/10 2:49:26 PM	80.8	50.1	36.5
12/7/10 2:49:56 PM	84.3	46.4	36.5
12/7/10 2:50:26 PM	83.1	45.3	36.5
12/7/10 2:50:56 PM	63.5	47.5	36.5
12/7/10 2:51:26 PM	52.8	51.4	36.5
12/7/10 2:51:56 PM	66.3	55.2	36.5
12/7/10 2:52:26 PM	79.7	51.7	36.5
12/7/10 2:52:56 PM	83.3	47.6	36.5
12/7/10 2:53:26 PM	83.8	45.6	36.5
12/7/10 2:53:56 PM	77.6	47.1	36.5
12/7/10 2:54:26 PM	60.1	50.7	36.5
12/7/10 2:54:56 PM	52.1	54.8	36.5
12/7/10 2:55:26 PM	70.7	53.1	36.5
12/7/10 2:55:56 PM	80.6	48.1	36.5
12/7/10 2:56:26 PM	83.3	45.7	36.5
12/7/10 2:56:56 PM	83.3	45.3	36.5
12/7/10 2:57:26 PM	80.8	47.8	36.5
12/7/10 2:57:56 PM	77.0	51.9	36.5
12/7/10 2:58:26 PM	81.9	55.5	36.5
12/7/10 2:58:56 PM	80.1	52.5	36.5
12/7/10 2:59:26 PM	62.9	48.0	36.5
12/7/10 2:59:56 PM	53.9	46.8	36.5
12/7/10 3:00:26 PM	70.5	48.0	36.5
12/7/10 3:00:56 PM	80.0	49.8	36.5
12/7/10 3:01:26 PM	82.8	50.0	36.5
12/7/10 3:01:56 PM	83.7	49.8	36.5
12/7/10 3:02:26 PM	75.7	51.5	36.5
12/7/10 3:02:56 PM	58.8	53.7	36.5
12/7/10 3:03:26 PM	52.3	50.7	36.5
12/7/10 3:03:56 PM	71.5	48.3	36.5
12/7/10 3:04:26 PM	81.5	46.4	36.5
12/7/10 3:04:56 PM	83.5	45.8	36.5
12/7/10 3:05:26 PM	83.7	46.4	36.5
12/7/10 3:05:56 PM	70.9	49.2	36.5

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 3:06:26 PM	55.9	53.2	36.5
12/7/10 3:06:56 PM	55.7	55.0	36.5
12/7/10 3:07:26 PM	75.8	50.3	36.5
12/7/10 3:07:56 PM	82.4	46.7	36.5
12/7/10 3:08:26 PM	84.1	45.1	36.5
12/7/10 3:08:56 PM	83.0	45.9	36.5
12/7/10 3:09:26 PM	63.9	49.5	36.5
12/7/10 3:09:56 PM	53.4	53.4	36.5
12/7/10 3:10:26 PM	63.2	54.4	36.5
12/7/10 3:10:56 PM	78.3	49.7	36.5
12/7/10 3:11:26 PM	82.5	46.4	36.5
12/7/10 3:11:56 PM	83.4	45.2	36.5
12/7/10 3:12:26 PM	83.3	46.8	36.5
12/7/10 3:12:56 PM	69.2	50.3	36.5
12/7/10 3:13:26 PM	55.7	54.6	36.5
12/7/10 3:13:56 PM	57.3	53.4	36.5
12/7/10 3:14:26 PM	76.2	49.1	36.5
12/7/10 3:14:56 PM	81.7	46.2	36.5
12/7/10 3:15:26 PM	82.5	45.0	36.5
12/7/10 3:15:56 PM	81.7	46.0	36.5
12/7/10 3:16:26 PM	81.3	49.5	36.5
12/7/10 3:16:56 PM	80.8	53.5	36.5
12/7/10 3:17:26 PM	81.0	54.5	36.5
12/7/10 3:17:56 PM	75.4	50.0	36.5
12/7/10 3:18:26 PM	59.4	47.1	36.5
12/7/10 3:18:56 PM	53.6	46.7	36.5
12/7/10 3:19:26 PM	73.4	48.6	36.5
12/7/10 3:19:56 PM	80.6	49.5	36.5
12/7/10 3:20:26 PM	82.9	49.2	36.5
12/7/10 3:20:56 PM	83.6	49.2	36.5
12/7/10 3:21:26 PM	80.4	51.0	36.5
12/7/10 3:21:56 PM	62.6	54.5	36.5
12/7/10 3:22:26 PM	52.3	53.3	36.5
12/7/10 3:22:56 PM	68.0	48.7	36.5
12/7/10 3:23:26 PM	79.2	45.9	36.5
12/7/10 3:23:56 PM	81.9	44.8	36.5
12/7/10 3:24:26 PM	82.5	44.9	36.5
12/7/10 3:24:56 PM	82.6	46.9	36.5
12/7/10 3:25:26 PM	82.5	51.2	36.5
12/7/10 3:25:56 PM	69.6	54.8	36.5
12/7/10 3:26:26 PM	55.4	52.3	36.5
12/7/10 3:26:56 PM	55.5	48.1	36.5
12/7/10 3:27:26 PM	76.0	45.9	36.5
12/7/10 3:27:56 PM	82.5	45.5	36.5
12/7/10 3:28:26 PM	83.6	45.7	36.5
12/7/10 3:28:56 PM	83.2	46.9	36.5
12/7/10 3:29:26 PM	68.1	50.1	36.5
12/7/10 3:29:56 PM	55.1	54.1	36.5
12/7/10 3:30:26 PM	59.2	53.5	36.5
12/7/10 3:30:56 PM	77.2	48.3	36.5
12/7/10 3:31:26 PM	81.8	45.4	36.5
12/7/10 3:31:56 PM	82.3	44.3	36.5
12/7/10 3:32:26 PM	82.1	45.3	36.5
12/7/10 3:32:56 PM	81.8	49.5	36.5
12/7/10 3:33:26 PM	81.5	53.4	36.5
12/7/10 3:33:56 PM	72.5	54.2	36.5
12/7/10 3:34:26 PM	78.0	49.1	36.5
12/7/10 3:34:56 PM	81.0	46.6	36.5
12/7/10 3:35:26 PM	78.4	46.3	36.5
12/7/10 3:35:56 PM	76.5	47.8	36.5
12/7/10 3:36:26 PM	80.1	51.2	36.5
12/7/10 3:36:56 PM	79.5	54.3	36.5
12/7/10 3:37:26 PM	74.9	52.0	36.5
12/7/10 3:37:56 PM	79.7	48.2	36.5
12/7/10 3:38:26 PM	80.3	47.1	36.5
12/7/10 3:38:56 PM	63.4	48.3	36.5
12/7/10 3:39:26 PM	54.0	50.8	36.5
12/7/10 3:39:56 PM	67.0	54.2	36.5
12/7/10 3:40:26 PM	79.0	51.9	36.5
12/7/10 3:40:56 PM	82.1	46.9	36.5

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 3:41:26 PM	82.8	45.2	36.5
12/7/10 3:41:56 PM	82.5	46.7	36.5
12/7/10 3:42:26 PM	76.8	50.6	36.5
12/7/10 3:42:56 PM	77.9	54.8	36.5
12/7/10 3:43:26 PM	82.7	52.7	36.5
12/7/10 3:43:56 PM	77.2	48.5	36.5
12/7/10 3:44:26 PM	60.8	46.0	36.5
12/7/10 3:44:56 PM	52.4	48.0	36.5
12/7/10 3:45:26 PM	70.1	51.7	36.5
12/7/10 3:45:56 PM	81.0	52.7	36.5
12/7/10 3:46:26 PM	83.3	49.1	36.5
12/7/10 3:46:56 PM	83.4	46.7	36.5
12/7/10 3:47:26 PM	71.2	48.7	36.5
12/7/10 3:47:56 PM	55.0	53.3	36.5
12/7/10 3:48:26 PM	56.6	54.1	36.5
12/7/10 3:48:56 PM	75.8	54.2	36.5
12/7/10 3:49:26 PM	81.2	49.2	36.5
12/7/10 3:49:56 PM	82.5	45.8	36.5
12/7/10 3:50:26 PM	82.3	44.6	36.5
12/7/10 3:50:56 PM	82.3	45.7	36.5
12/7/10 3:51:26 PM	79.7	49.3	36.5
12/7/10 3:51:56 PM	60.8	54.1	36.5
12/7/10 3:52:26 PM	52.3	54.6	36.5
12/7/10 3:52:56 PM	68.1	50.0	36.5
12/7/10 3:53:26 PM	78.3	46.9	36.5
12/7/10 3:53:56 PM	80.8	45.6	36.5
12/7/10 3:54:26 PM	80.3	45.5	36.5
12/7/10 3:54:56 PM	79.9	45.8	36.5
12/7/10 3:55:26 PM	80.1	47.8	35.1
12/7/10 3:55:56 PM	80.5	51.3	35.1
12/7/10 3:56:26 PM	81.2	54.6	35.1
12/7/10 3:56:56 PM	82.3	51.4	35.1
12/7/10 3:57:26 PM	73.2	48.4	35.1
12/7/10 3:57:56 PM	57.8	46.7	35.1
12/7/10 3:58:26 PM	53.5	48.2	35.1

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 3:58:56 PM	74.7	52.3	35.1
12/7/10 3:59:26 PM	81.8	53.5	35.1
12/7/10 3:59:56 PM	83.7	49.3	35.1
12/7/10 4:00:26 PM	83.3	46.2	35.1
12/7/10 4:00:56 PM	66.5	46.0	35.1
12/7/10 4:01:26 PM	54.0	49.1	35.1
12/7/10 4:01:56 PM	58.6	53.8	35.1
12/7/10 4:02:26 PM	77.7	55.1	35.1
12/7/10 4:02:56 PM	82.9	50.1	35.1
12/7/10 4:03:26 PM	84.2	46.2	35.1
12/7/10 4:03:56 PM	81.5	44.8	35.1
12/7/10 4:04:26 PM	61.6	46.0	35.1
12/7/10 4:04:56 PM	52.0	49.7	35.1
12/7/10 4:05:26 PM	67.7	53.7	35.1
12/7/10 4:05:56 PM	79.2	54.2	35.1
12/7/10 4:06:26 PM	82.5	49.2	35.1
12/7/10 4:06:56 PM	83.2	45.5	35.1
12/7/10 4:07:26 PM	83.2	44.9	35.1
12/7/10 4:07:56 PM	72.8	46.9	35.1
12/7/10 4:08:26 PM	56.2	50.7	35.1
12/7/10 4:08:56 PM	54.6	54.3	35.1
12/7/10 4:09:26 PM	75.0	53.4	35.1
12/7/10 4:09:56 PM	81.4	48.7	35.1
12/7/10 4:10:26 PM	82.2	45.8	35.1
12/7/10 4:10:56 PM	81.4	44.9	35.1
12/7/10 4:11:26 PM	80.8	45.7	35.1
12/7/10 4:11:56 PM	80.6	49.7	35.1
12/7/10 4:12:26 PM	80.8	52.7	35.1
12/7/10 4:12:56 PM	81.0	54.8	35.1
12/7/10 4:13:26 PM	72.5	50.8	35.1
12/7/10 4:13:56 PM	77.4	47.4	35.1
12/7/10 4:14:26 PM	81.7	46.5	35.1
12/7/10 4:14:56 PM	74.7	48.5	35.1
12/7/10 4:15:26 PM	57.9	50.7	35.1
12/7/10 4:15:56 PM	52.6	51.7	35.1

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 4:16:26 PM	73.4	53.3	35.1
12/7/10 4:16:56 PM	81.0	49.6	35.1
12/7/10 4:17:26 PM	82.9	47.1	35.1
12/7/10 4:17:56 PM	83.2	45.5	35.1
12/7/10 4:18:26 PM	82.6	45.9	35.1
12/7/10 4:18:56 PM	66.3	49.0	35.1
12/7/10 4:19:26 PM	54.6	53.0	35.1
12/7/10 4:19:56 PM	60.0	55.2	35.1
12/7/10 4:20:26 PM	78.4	50.7	35.1
12/7/10 4:20:56 PM	82.8	46.9	35.1
12/7/10 4:21:26 PM	84.4	45.0	35.1
12/7/10 4:21:56 PM	77.2	44.9	35.1
12/7/10 4:22:26 PM	59.7	46.4	35.1
12/7/10 4:22:56 PM	51.6	50.3	35.1
12/7/10 4:23:26 PM	71.5	54.3	35.1
12/7/10 4:23:56 PM	80.8	53.4	35.1
12/7/10 4:24:26 PM	82.9	48.2	35.1
12/7/10 4:24:56 PM	83.3	45.1	35.1
12/7/10 4:25:26 PM	82.6	44.8	35.1
12/7/10 4:25:56 PM	66.3	47.6	35.1
12/7/10 4:26:26 PM	53.9	51.7	35.1
12/7/10 4:26:56 PM	63.4	55.1	35.1
12/7/10 4:27:26 PM	78.0	51.9	35.1
12/7/10 4:27:56 PM	82.2	47.3	35.1
12/7/10 4:28:26 PM	82.5	44.9	35.1
12/7/10 4:28:56 PM	81.7	44.6	35.1
12/7/10 4:29:26 PM	80.9	46.4	35.1
12/7/10 4:29:56 PM	80.9	50.1	35.1
12/7/10 4:30:26 PM	80.9	53.8	35.1
12/7/10 4:30:56 PM	71.4	51.3	35.1
12/7/10 4:31:26 PM	77.5	49.0	35.1
12/7/10 4:31:56 PM	81.2	46.0	35.1
12/7/10 4:32:26 PM	75.3	46.9	35.1
12/7/10 4:32:56 PM	56.7	48.7	35.1
12/7/10 4:33:26 PM	53.8	50.1	35.1

Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 4:33:56 PM	72.7	52.8	35.1
12/7/10 4:34:26 PM	81.5	54.1	35.1
12/7/10 4:34:56 PM	83.4	48.9	35.1
12/7/10 4:35:26 PM	84.1	45.3	35.1
12/7/10 4:35:56 PM	73.1	45.0	35.1
12/7/10 4:36:26 PM	57.3	48.2	35.1
12/7/10 4:36:56 PM	54.2	53.0	35.1
12/7/10 4:37:26 PM	74.2	55.5	35.1
12/7/10 4:37:56 PM	81.5	50.9	35.1
12/7/10 4:38:26 PM	83.9	46.0	35.1
12/7/10 4:38:56 PM	84.5	44.1	35.1
12/7/10 4:39:26 PM	69.6	44.8	35.1
12/7/10 4:39:56 PM	54.6	48.6	35.1
12/7/10 4:40:26 PM	55.5	52.8	35.1
12/7/10 4:40:56 PM	76.7	55.4	35.1
12/7/10 4:41:26 PM	81.9	50.0	35.1
12/7/10 4:41:56 PM	83.2	45.0	35.1
12/7/10 4:42:26 PM	83.4	43.9	35.1
12/7/10 4:42:56 PM	83.3	45.0	35.1
12/7/10 4:43:26 PM	73.9	48.9	35.1
12/7/10 4:43:56 PM	56.8	52.8	35.1
12/7/10 4:44:26 PM	53.2	54.8	35.1
12/7/10 4:44:56 PM	74.2	49.9	35.1
12/7/10 4:45:26 PM	82.6	45.8	35.1
12/7/10 4:45:56 PM	82.6	44.3	35.1
12/7/10 4:46:26 PM	81.7	44.7	35.1
12/7/10 4:46:56 PM	81.2	51.2	35.1
12/7/10 4:47:26 PM	81.1	54.5	35.1
12/7/10 4:47:56 PM	80.5	52.5	32.7
12/7/10 4:48:26 PM	72.8	47.9	32.7
12/7/10 4:48:56 PM	79.7	46.8	32.7
12/7/10 4:49:26 PM	80.6	46.0	32.7
12/7/10 4:49:56 PM	75.3	48.9	32.7
12/7/10 4:50:26 PM	78.8	50.6	32.7
12/7/10 4:50:56 PM	81.0	48.8	32.7

Time	Preheat Temp	Cold Deck Temp	OAT	Time	Preheat Temp	Cold Deck Temp	OAT
12/7/10 4:51:26 PM	69.1	46.8	32.7	12/7/10 5:08:56 PM	80.4	49.3	32.7
12/7/10 4:51:56 PM	57.4	47.8	32.7	12/7/10 5:09:26 PM	80.6	51.7	32.7
12/7/10 4:52:26 PM	54.1	48.3	32.7	12/7/10 5:09:56 PM	77.9	53.7	32.7
12/7/10 4:52:56 PM	60.0	48.4	32.7	12/7/10 5:10:26 PM	55.3	51.4	32.7
12/7/10 4:53:26 PM	77.0	48.0	32.7	12/7/10 5:10:56 PM	52.1	46.7	32.7
12/7/10 4:53:56 PM	82.2	48.6	32.7	12/7/10 5:11:26 PM	70.7	45.3	32.7
12/7/10 4:54:26 PM	83.1	51.5	32.7	12/7/10 5:11:56 PM	80.2	45.2	32.7
12/7/10 4:54:56 PM	83.3	54.8	32.7	12/7/10 5:12:26 PM	82.5	47.4	32.7
12/7/10 4:55:26 PM	78.0	52.1	32.7	12/7/10 5:12:56 PM	83.0	51.2	32.7
12/7/10 4:55:56 PM	59.6	47.1	32.7	12/7/10 5:13:26 PM	82.9	54.7	32.7
12/7/10 4:56:26 PM	51.5	45.4	32.7	12/7/10 5:13:56 PM	76.7	52.6	32.7
12/7/10 4:56:56 PM	71.2	44.9	32.7	12/7/10 5:14:26 PM	78.1	48.1	32.7
12/7/10 4:57:26 PM	80.6	45.3	32.7	12/7/10 5:14:56 PM	83.3	45.9	32.7
12/7/10 4:57:56 PM	83.3	47.6	32.7	12/7/10 5:15:26 PM	76.7	50.2	32.7
12/7/10 4:58:26 PM	84.3	51.3	32.7	12/7/10 5:15:56 PM	58.7	48.5	32.7
12/7/10 4:58:56 PM	75.8	54.7	32.7	12/7/10 5:16:26 PM	51.4	48.7	32.7
12/7/10 4:59:26 PM	57.9	52.3	32.7	12/7/10 5:16:56 PM	71.4	50.9	32.7
12/7/10 4:59:56 PM	51.2	47.3	32.7	12/7/10 5:17:26 PM	80.6	51.8	32.7
12/7/10 5:00:26 PM	75.7	44.8	32.7	12/7/10 5:17:56 PM	82.9	49.4	32.7
12/7/10 5:00:56 PM	80.5	44.8	32.7	12/7/10 5:18:26 PM	83.4	48.5	31.5
12/7/10 5:01:26 PM	82.5	47.1	32.7	12/9/10 2:30:01 PM	79.7	49.8	31.5
12/7/10 5:01:56 PM	82.9	50.7	32.7	12/9/10 2:30:31 PM	80.2	51.9	31.5
12/7/10 5:02:26 PM	82.8	54.1	32.7	12/9/10 2:31:01 PM	80.3	50.4	31.5
12/7/10 5:02:56 PM	79.9	51.4	32.7	12/9/10 2:31:31 PM	80.1	49.1	31.5
12/7/10 5:03:26 PM	61.3	48.5	32.7	12/9/10 2:32:01 PM	80.3	49.1	31.5
12/7/10 5:03:56 PM	52.0	45.7	32.7	12/9/10 2:32:31 PM	80.4	50.7	31.5
12/7/10 5:04:26 PM	67.7	44.8	32.7	12/9/10 2:33:01 PM	80.4	50.9	31.5
12/7/10 5:04:56 PM	79.0	45.0	32.7	12/9/10 2:33:31 PM	80.6	50.0	31.5
12/7/10 5:05:26 PM	81.4	46.8	32.7	12/9/10 2:34:01 PM	80.6	49.4	31.5
12/7/10 5:05:56 PM	80.9	50.9	32.7	12/9/10 2:34:31 PM	80.8	49.6	31.5
12/7/10 5:06:26 PM	80.0	53.4	32.7	12/9/10 2:35:01 PM	81.0	50.2	31.5
12/7/10 5:06:56 PM	79.9	53.7	32.7	12/9/10 2:35:31 PM	81.1	50.6	31.5
12/7/10 5:07:26 PM	80.0	49.2	32.7	12/9/10 2:36:01 PM	81.2	50.3	31.5
12/7/10 5:07:56 PM	80.0	46.5	32.7	12/9/10 2:36:31 PM	81.1	49.9	31.5
12/7/10 5:08:26 PM	80.3	46.9	32.7	12/9/10 2:37:01 PM	80.2	49.7	31.5

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 2:37:31 PM	80.1	49.9	31.5
12/9/10 2:38:01 PM	79.5	50.1	31.5
12/9/10 2:38:31 PM	78.8	50.2	31.5
12/9/10 2:39:01 PM	77.6	50.0	31.5
12/9/10 2:39:31 PM	76.7	49.8	31.5
12/9/10 2:40:01 PM	76.0	49.8	31.5
12/9/10 2:40:31 PM	75.4	49.9	31.5
12/9/10 2:41:01 PM	75.1	50.0	31.5
12/9/10 2:41:31 PM	74.9	50.0	31.5
12/9/10 2:42:01 PM	75.0	50.1	31.5
12/9/10 2:42:31 PM	75.2	49.9	31.5
12/9/10 2:43:01 PM	75.4	49.8	31.5
12/9/10 2:43:31 PM	75.7	50.0	31.5
12/9/10 2:44:01 PM	76.3	50.1	31.5
12/9/10 2:44:31 PM	77.0	50.1	31.5
12/9/10 2:45:01 PM	77.7	49.9	31.5
12/9/10 2:45:31 PM	78.4	49.7	31.5
12/9/10 2:46:01 PM	79.0	49.7	31.5
12/9/10 2:46:31 PM	79.4	49.9	31.5
12/9/10 2:47:01 PM	79.8	50.2	31.5
12/9/10 2:47:31 PM	80.1	50.0	31.5
12/9/10 2:48:01 PM	80.3	49.7	31.5
12/9/10 2:48:36 PM	80.5	49.8	31.5
12/9/10 2:49:06 PM	80.6	50.3	31.5
12/9/10 2:49:36 PM	80.8	50.4	31.5
12/9/10 2:50:06 PM	81.0	49.9	31.5
12/9/10 2:50:36 PM	81.0	49.8	31.5
12/9/10 2:51:06 PM	81.2	50.2	31.5
12/9/10 2:51:36 PM	81.2	50.3	31.5
12/9/10 2:52:06 PM	81.2	50.0	31.5
12/9/10 2:52:36 PM	81.2	49.8	31.5
12/9/10 2:53:06 PM	81.1	49.8	31.5
12/9/10 2:53:36 PM	80.9	50.1	31.5
12/9/10 2:54:06 PM	80.4	50.3	31.5
12/9/10 2:54:36 PM	80.3	50.2	31.5

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 2:55:06 PM	80.5	49.9	31.5
12/9/10 2:55:36 PM	80.6	49.8	31.5
12/9/10 2:56:06 PM	80.6	49.9	31.5
12/9/10 2:56:36 PM	80.4	50.2	31.5
12/9/10 2:57:06 PM	79.5	50.3	31.5
12/9/10 2:57:36 PM	79.4	50.0	31.5
12/9/10 2:58:06 PM	78.7	49.8	31.5
12/9/10 2:58:36 PM	77.6	49.8	31.5
12/9/10 2:59:06 PM	76.9	50.1	31.5
12/9/10 2:59:36 PM	76.5	50.2	31.5
12/9/10 3:00:06 PM	76.3	50.0	31.5
12/9/10 3:00:36 PM	76.2	49.7	31.5
12/9/10 3:01:06 PM	76.3	49.7	31.5
12/9/10 3:01:36 PM	76.1	50.1	31.5
12/9/10 3:02:06 PM	76.0	50.3	31.5
12/9/10 3:02:36 PM	76.3	50.1	37.4
12/9/10 3:03:06 PM	76.5	49.8	37.4
12/9/10 3:03:36 PM	76.9	49.8	37.4
12/9/10 3:04:06 PM	77.5	50.1	37.4
12/9/10 3:04:36 PM	78.0	50.3	37.4
12/9/10 3:05:06 PM	78.6	49.9	37.4
12/9/10 3:05:36 PM	79.0	49.6	37.4
12/9/10 3:06:06 PM	79.2	49.8	37.4
12/9/10 3:06:36 PM	79.4	50.3	37.4
12/9/10 3:07:06 PM	79.8	50.2	37.4
12/9/10 3:07:36 PM	80.1	49.7	37.4
12/9/10 3:08:06 PM	80.3	49.9	37.4
12/9/10 3:08:36 PM	80.8	50.6	37.4
12/9/10 3:09:06 PM	80.9	50.3	37.4
12/9/10 3:09:36 PM	81.1	49.7	37.4
12/9/10 3:10:06 PM	81.3	49.7	37.4
12/9/10 3:10:36 PM	81.5	50.3	37.4
12/9/10 3:11:06 PM	81.5	50.4	37.4
12/9/10 3:11:36 PM	81.5	49.9	37.4
12/9/10 3:12:06 PM	81.2	49.7	37.4

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 3:12:36 PM	80.8	49.7	37.4
12/9/10 3:13:06 PM	80.3	50.3	37.4
12/9/10 3:13:36 PM	80.1	50.4	37.4
12/9/10 3:14:06 PM	80.0	50.0	37.4
12/9/10 3:14:36 PM	79.9	49.7	37.4
12/9/10 3:15:06 PM	79.8	49.8	37.4
12/9/10 3:15:36 PM	79.4	50.1	37.4
12/9/10 3:16:06 PM	79.0	50.3	37.4
12/9/10 3:16:36 PM	78.3	50.0	37.4
12/9/10 3:17:06 PM	76.6	49.7	37.4
12/9/10 3:17:36 PM	76.5	49.9	37.4
12/9/10 3:18:06 PM	75.9	50.1	37.4
12/9/10 3:18:36 PM	75.6	50.2	37.4
12/9/10 3:19:06 PM	75.6	50.0	37.4
12/9/10 3:19:36 PM	75.5	49.7	37.4
12/9/10 3:20:06 PM	75.6	49.8	37.4
12/9/10 3:20:36 PM	75.9	50.1	37.4
12/9/10 3:21:06 PM	76.5	50.3	37.4
12/9/10 3:21:36 PM	77.1	50.0	37.4
12/9/10 3:22:06 PM	77.8	49.7	37.4
12/9/10 3:22:36 PM	78.5	49.8	37.4
12/9/10 3:23:06 PM	79.2	50.3	37.4
12/9/10 3:23:36 PM	79.7	50.2	37.4
12/9/10 3:24:06 PM	79.9	49.8	37.4
12/9/10 3:24:36 PM	80.2	49.6	37.4
12/9/10 3:25:06 PM	80.3	49.8	37.4
12/9/10 3:25:36 PM	80.5	50.2	37.4
12/9/10 3:26:06 PM	80.4	50.2	37.4
12/9/10 3:26:36 PM	80.4	49.7	37.4
12/9/10 3:27:06 PM	80.5	49.8	37.4
12/9/10 3:27:36 PM	80.6	50.3	37.4
12/9/10 3:28:06 PM	80.8	50.4	37.4
12/9/10 3:28:36 PM	81.0	49.9	37.4
12/9/10 3:29:06 PM	81.0	49.7	37.4
12/9/10 3:29:36 PM	81.3	50.0	37.4

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 3:30:06 PM	81.3	50.4	37.4
12/9/10 3:30:36 PM	81.2	50.1	37.4
12/9/10 3:31:06 PM	81.2	49.7	37.4
12/9/10 3:31:36 PM	81.2	49.8	37.4
12/9/10 3:32:06 PM	81.2	50.2	37.4
12/9/10 3:32:36 PM	81.1	50.3	37.4
12/9/10 3:33:06 PM	80.8	50.0	37.4
12/9/10 3:33:36 PM	80.3	49.8	37.4
12/9/10 3:34:06 PM	79.7	49.8	37.4
12/9/10 3:34:36 PM	79.0	50.0	37.4
12/9/10 3:35:06 PM	78.3	50.3	37.4
12/9/10 3:35:36 PM	77.1	50.0	37.4
12/9/10 3:36:06 PM	76.3	49.8	37.4
12/9/10 3:36:36 PM	75.6	49.8	37.4
12/9/10 3:37:06 PM	75.1	50.2	37.4
12/9/10 3:37:36 PM	75.1	50.2	37.4
12/9/10 3:38:06 PM	75.1	49.9	37.4
12/9/10 3:38:36 PM	75.0	49.7	37.4
12/9/10 3:39:06 PM	75.5	49.8	37.4
12/9/10 3:39:36 PM	76.0	50.2	37.4
12/9/10 3:40:06 PM	76.8	50.2	37.4
12/9/10 3:40:36 PM	77.5	49.9	37.4
12/9/10 3:41:06 PM	78.1	49.8	37.4
12/9/10 3:41:36 PM	78.5	50.0	37.4
12/9/10 3:42:06 PM	79.0	50.4	37.4
12/9/10 3:42:36 PM	79.3	50.0	37.4
12/9/10 3:43:06 PM	79.7	49.6	37.4
12/9/10 3:43:36 PM	79.8	49.7	37.4
12/9/10 3:44:06 PM	79.9	50.2	37.4
12/9/10 3:44:36 PM	80.1	50.2	37.4
12/9/10 3:45:06 PM	80.1	49.7	37.4
12/9/10 3:45:36 PM	80.3	49.7	37.4
12/9/10 3:46:06 PM	80.3	50.4	37.4
12/9/10 3:46:36 PM	80.4	50.3	37.4
12/9/10 3:47:06 PM	80.4	49.8	37.4

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 3:47:36 PM	80.3	49.7	37.4
12/9/10 3:48:06 PM	80.2	50.3	37.4
12/9/10 3:48:36 PM	79.8	50.4	37.4
12/9/10 3:49:06 PM	79.5	49.8	37.4
12/9/10 3:49:36 PM	79.2	49.6	37.4
12/9/10 3:50:06 PM	79.0	50.0	37.4
12/9/10 3:50:36 PM	78.8	50.4	37.4
12/9/10 3:51:06 PM	78.8	50.1	37.4
12/9/10 3:51:36 PM	78.9	49.7	37.4
12/9/10 3:52:06 PM	78.9	49.7	37.4
12/9/10 3:52:36 PM	78.8	50.2	37.4
12/9/10 3:53:06 PM	78.6	50.4	37.4
12/9/10 3:53:36 PM	78.2	49.9	37.4
12/9/10 3:54:06 PM	77.5	49.6	37.4
12/9/10 3:54:36 PM	76.6	49.9	37.4
12/9/10 3:55:06 PM	75.9	50.3	36.0
12/9/10 3:55:36 PM	75.5	50.2	36.0
12/9/10 3:56:06 PM	75.4	49.8	36.0
12/9/10 3:56:36 PM	75.1	49.7	36.0
12/9/10 3:57:06 PM	75.4	50.0	36.0
12/9/10 3:57:36 PM	75.5	50.3	36.0
12/9/10 3:58:06 PM	75.8	50.0	36.0
12/9/10 3:58:36 PM	76.7	49.7	36.0
12/9/10 3:59:06 PM	78.1	49.8	36.0
12/9/10 3:59:36 PM	79.4	50.2	36.0
12/9/10 4:00:06 PM	80.9	50.3	36.0
12/9/10 4:00:36 PM	81.9	49.8	36.0
12/9/10 4:01:06 PM	82.6	49.7	36.0
12/9/10 4:01:36 PM	83.2	49.9	36.0
12/9/10 4:02:06 PM	83.6	50.4	36.0
12/9/10 4:02:36 PM	84.0	50.0	36.0
12/9/10 4:03:06 PM	84.2	49.6	36.0
12/9/10 4:03:36 PM	84.4	49.7	36.0
12/9/10 4:04:06 PM	84.5	50.2	36.0
12/9/10 4:04:36 PM	84.7	50.3	36.0

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 4:05:06 PM	84.8	49.8	36.0
12/9/10 4:05:36 PM	84.3	49.7	36.0
12/9/10 4:06:06 PM	83.4	50.3	36.0
12/9/10 4:06:36 PM	82.4	50.4	36.0
12/9/10 4:07:06 PM	81.4	49.9	36.0
12/9/10 4:07:36 PM	80.6	49.7	36.0
12/9/10 4:08:06 PM	80.1	50.0	36.0
12/9/10 4:08:36 PM	79.6	50.4	36.0
12/9/10 4:09:06 PM	79.4	50.2	36.0
12/9/10 4:09:36 PM	79.2	49.8	36.0
12/9/10 4:10:06 PM	79.0	49.8	36.0
12/9/10 4:10:36 PM	79.0	50.0	36.0
12/9/10 4:11:06 PM	78.7	50.3	36.0
12/9/10 4:11:36 PM	78.1	50.2	36.0
12/9/10 4:12:06 PM	77.3	49.8	36.0
12/9/10 4:12:36 PM	76.3	49.8	36.0
12/9/10 4:13:06 PM	75.4	50.0	36.0
12/9/10 4:13:36 PM	75.0	50.3	36.0
12/9/10 4:14:06 PM	74.7	50.2	36.0
12/9/10 4:14:36 PM	74.7	50.0	36.0
12/9/10 4:15:06 PM	74.7	49.9	36.0
12/9/10 4:15:36 PM	74.8	49.8	36.0
12/9/10 4:16:06 PM	75.0	50.0	36.0
12/9/10 4:16:36 PM	75.3	50.1	36.0
12/9/10 4:17:06 PM	76.5	50.1	36.0
12/9/10 4:17:36 PM	76.6	50.0	36.0
12/9/10 4:18:06 PM	77.3	49.9	36.0
12/9/10 4:18:36 PM	77.9	49.9	36.0
12/9/10 4:19:06 PM	78.3	50.0	36.0
12/9/10 4:19:36 PM	78.7	50.1	36.0
12/9/10 4:20:06 PM	79.2	50.2	36.0
12/9/10 4:20:36 PM	79.5	50.1	36.0
12/9/10 4:21:06 PM	79.7	50.1	36.0
12/9/10 4:21:36 PM	79.9	49.9	36.0
12/9/10 4:22:06 PM	80.1	49.9	36.0

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 4:22:36 PM	80.1	49.8	36.0
12/9/10 4:23:06 PM	80.4	49.9	36.0
12/9/10 4:23:36 PM	80.6	50.1	36.0
12/9/10 4:24:06 PM	80.8	50.1	36.0
12/9/10 4:24:36 PM	81.0	50.0	36.0
12/9/10 4:25:06 PM	81.0	49.9	36.0
12/9/10 4:25:36 PM	80.8	49.9	36.0
12/9/10 4:26:06 PM	80.5	50.1	36.0
12/9/10 4:26:36 PM	80.2	50.3	36.0
12/9/10 4:27:06 PM	79.8	50.2	36.0
12/9/10 4:27:36 PM	79.5	50.0	36.0
12/9/10 4:28:06 PM	79.4	49.9	36.0
12/9/10 4:28:36 PM	79.4	50.0	36.0
12/9/10 4:29:06 PM	79.2	50.1	36.0
12/9/10 4:29:36 PM	78.8	50.2	36.0
12/9/10 4:30:06 PM	78.3	49.9	36.0
12/9/10 4:30:36 PM	77.7	49.9	36.0
12/9/10 4:31:06 PM	76.9	49.9	36.0
12/9/10 4:31:36 PM	75.8	50.0	36.0
12/9/10 4:32:06 PM	75.2	50.1	36.0
12/9/10 4:32:36 PM	75.1	50.1	36.0
12/9/10 4:33:06 PM	75.1	50.0	36.0
12/9/10 4:33:36 PM	75.1	50.0	36.0
12/9/10 4:34:06 PM	75.2	50.0	36.0
12/9/10 4:34:36 PM	75.6	49.9	36.0
12/9/10 4:35:06 PM	76.1	49.9	36.0
12/9/10 4:35:36 PM	76.8	49.9	36.0
12/9/10 4:36:06 PM	77.5	49.9	36.0
12/9/10 4:36:36 PM	78.1	50.0	36.0
12/9/10 4:37:06 PM	79.2	49.9	36.0
12/9/10 4:37:36 PM	79.4	50.0	36.0
12/9/10 4:38:06 PM	79.9	50.0	36.0
12/9/10 4:38:36 PM	80.3	50.0	36.0
12/9/10 4:39:06 PM	80.8	50.0	36.0
12/9/10 4:39:36 PM	81.2	50.0	36.0

Time	Preheat Temp	Cold Deck Temp	OAT
12/9/10 4:40:06 PM	81.3	49.8	36.0
12/9/10 4:40:36 PM	81.5	49.9	36.0
12/9/10 4:41:06 PM	81.6	50.0	36.0
12/9/10 4:41:36 PM	81.6	50.1	36.0
12/9/10 4:42:06 PM	81.7	49.9	36.0
12/9/10 4:42:36 PM	81.9	49.8	36.0
12/9/10 4:43:06 PM	81.9	49.9	36.0
12/9/10 4:43:36 PM	81.9	50.2	36.0
12/9/10 4:44:06 PM	81.9	50.2	36.0
12/9/10 4:44:36 PM	81.7	49.9	36.0
12/9/10 4:45:06 PM	81.1	49.8	36.0
12/9/10 4:45:36 PM	80.6	49.8	36.0
12/9/10 4:46:06 PM	80.3	49.9	36.0
12/9/10 4:46:36 PM	80.1	50.2	36.0
12/9/10 4:47:06 PM	80.1	50.3	36.0
12/9/10 4:47:36 PM	80.1	50.0	36.0
12/9/10 4:48:06 PM	80.1	49.8	36.0
12/9/10 4:48:36 PM	80.0	49.8	36.0
12/9/10 4:49:06 PM	79.6	50.1	36.0
12/9/10 4:49:36 PM	78.8	50.3	36.0
12/9/10 4:50:06 PM	77.8	49.9	36.0
12/9/10 4:50:36 PM	76.9	49.8	36.0
12/9/10 4:51:06 PM	76.5	49.8	36.0
12/9/10 4:51:36 PM	76.2	50.0	36.0
12/9/10 4:52:06 PM	76.0	50.3	36.0
12/9/10 4:52:36 PM	76.0	50.1	36.0
12/9/10 4:53:06 PM	76.1	49.8	36.0
12/9/10 4:53:36 PM	76.6	49.8	36.0
12/9/10 4:54:06 PM	78.0	50.0	36.0
12/9/10 4:54:36 PM	79.1	50.3	36.0
12/9/10 4:55:06 PM	80.0	49.8	36.0
12/9/10 4:55:36 PM	80.8	49.8	36.0
12/9/10 4:56:06 PM	81.7	49.7	31.6

Table 19: F/B Damper and Modulating Valve Trend Data, Ireland Army Community Hospital, Fort Knox, KY

TIME	OAT	DAT	DAT SETPOINT	HEATING VALVE POSITION	MAT	F/B DAMPER POSITION
1/11/2011 8:00	25.5	56.5	62.8	31	44.9	100
1/11/2011 7:45	25.15	56.3	62.8	40	45.1	100
1/11/2011 7:30	24.8	56.3	62.8	41	45.1	100
1/11/2011 7:15	25.15	56.4	62.8	39	45.3	100
1/11/2011 7:00	25.5	56.4	62.8	40	45.4	100
1/11/2011 6:45	25.15	56.3	62.8	39	45.3	100
1/11/2011 6:30	24.8	56.2	62.4	40	45.5	100
1/11/2011 6:15	25.35	56.2	62.4	40	45.3	100
1/11/2011 6:00	25.9	55.6	62.4	40	45.2	100
1/11/2011 5:45	26.6	55.7	62.4	37	45.5	100
1/11/2011 5:30	27.0	55.2	62.4	38	45.5	100
1/11/2011 5:15	27.3	54.9	62	34	46	100
1/11/2011 5:00	27.3	54.1	62	34	46.3	100
1/11/2011 4:45	27.55	54.7	62	28	47.7	100
1/11/2011 4:30	27.8	52.9	62	28	47.7	100

APPENDIX E

SPREADSHEET DATA FILES

This section of the appendix provides the input files used to estimate the energy savings associated with steam pressure reduction and improved control sequences mentioned in the main body of this thesis.

E.1 INPUT FILE FOR REDUCED AIR STRATIFICATION DUE TO STEAM PRESSURE REDUCTION

This input file is used to estimate the amount of air that flows through the coil and the amount which bypasses the coil in order to maintain a discharge air temperature of 55°F before and after the steam pressure was reduced for the specimen AHU in Alexandria, Louisiana.

The input conditions for simulating the effect of reduced steam pressure are provided in the Section “Given Input Conditions” in the spreadsheet (Rows 1-7). The effectiveness of heat transfer is initially computed based on the field experiments carried out at the facility and is found to be 0.198. Based on the calculated effectiveness, the temperature rise across the coil for various inlet air conditions (Rows 25-30) is computed by using the formula

$$\Delta T = e * (T_{sat} - T_{OA}) \qquad \textit{Eq. E.1.1}$$

For a given inlet air condition, the airflow through the coil and that bypassing the coil is computed in Columns 3,4 and 6-7 (C3, C4, C6 & C7) for Rows 33-42 (R33-R42) using the formula provided in Row 18 and Row 24, i.e.,

Total Design Flow = Flow through the Coil + Flow Bypassing the Coil

$$\mathbf{V_{TOT} = V_F + V_B} \qquad \textit{Eq. E.1.2}$$

$$Airflow\ through\ the\ coil\ (\boldsymbol{V_F}) =$$

$$Design\ Airflow\ * \frac{Discharge\ Air\ Temp - Outside\ Air\ Temp}{Temperature\ Rise\ Across\ the\ Coil} \qquad \textit{Eq. E.1.3}$$

$$\mathbf{V_F = V_{TOT} * (T_{DA} - T_{OA})/(T_{PH} - T_{OA})}$$

The percentage of airflow through the coil with respect to the total design airflow in Columns 5 and 8 (C5 & C8) and in Rows 33-42 (R33-R42) is given by

% of airflow through coil = $\mathbf{V_F} \div \mathbf{V_{TOT}}$ *Eq. E.1.4*

	C1	C2	C3
R1	**Given Input Conditions**	Value	Units
R2	Design Airflow	7,800	cfm
R3	Discharge Air Temperature Setpoint	55	°F
R4	Heat Exchanger Effectiveness 15psig (e)	0.198	
R5	Heat Exchanger Effectiveness 5psig (e)	0.198	
R6	Saturation Temperature of Steam @ 15psig (T_{sat_1})	250	°F
R7	Saturation Temperature of Steam @ 5psig (T_{sat_2})	227	°F

	C1	C2	C3
R8	**Variables**	**Definition**	**Units**
R9	$\mathbf{T_{OA}}$	Outside Air Temperature	°F
R10	$\mathbf{T_{DA}}$	Discharge Air Temperature Setpoint	°F
R11	$\mathbf{T_{MA}}$	Mixed Air Temperature	°F
R12	$\mathbf{\Delta T}$	Temperature Rise across Preheat Coil	
R13	$\mathbf{T_{PH}}$	Temperature Exiting Preheat Coil	°F
R14	$\mathbf{V_F}$	Airflow Through the Coil	cfm
R15	$\mathbf{V_B}$	Airflow Bypassing the Coil	cfm
R16	$\mathbf{V_{TOT}}$	Design Airflow for the AHU	cfm

	C1	C2	C3
	Formulae		
R17	$\Delta T = e * (T_{sat} - T_{OA})$		Eq. E.1.1
R18	$V_{TOT} = V_F + V_B$		Eq. E.1.2
R19	$V_F * T_{PH} + V_B * T_{OA} = V_{TOT} * T_{DA}$		
R20	From R18 & R19		
R21	$V_F * T_{PH} + (V_{TOT} - V_F) * T_{OA} = V_{TOT} * T_{DA}$		
R22	$V_{TOT} * T_{DA} = V_F * T_{PH} + (V_{TOT} - V_F) * T_{OA}$		
R23	$VF * (T_{PH} - T_{OA}) = V_{TOT} * (T_{DA} - T_{OA})$		
R24	$V_F = V_{TOT} * (T_{DA} - T_{OA})/(T_{PH} - T_{OA})$		Eq. E.1.3

	C1	C2	C3
	OA Temperature °F	**Estimated Temperature Rise °F (15psig)**	**Estimated Temperature Rise °F (5psig)**
R25	22	45.1	40.6
R26	30	43.5	39.0
R27	40	41.5	37.0
R28	50	39.5	35.1
R29	68.5	37.6	31.4
R30	80	33.6	29.1
R31	90	31.6	27.1
R32	100	29.6	25.2

	Bin Data		**15psig steam**			**5psig steam**		
	C1	C2	C3	C4	C5	C6	C7	C8
	Average Dry Bulb Temp (F)	# of hours	Airflow Through Coil (cfm)	Airflow through Bypass (cfm)	% of Face to Bypass Flow	Airflow Through Coil (cfm)	Airflow through Bypass (cfm)	% of Face to Bypass Flow
R33	-	-	-	-	-	-	-	-
R34	20	2	6003	1797	77%	6657	1143	85%
R35	24	14	5411	2389	69%	6012	1788	77%
R36	28	114	4798	3002	62%	5342	2458	68%
R37	32	223	4162	3638	53%	4644	3156	60%
R38	36	220	3503	4297	45%	3916	3884	50%

R39	40	201	2818	4982	36%	3158	4642	40%
R40	44	394	2107	5693	27%	2366	5434	30%
R41	48	489	1367	6433	18%	1540	6260	20%
R42	52	598	598	7202	8%	675	7125	9%

E.2 INPUT FILE FOR REDUCED ENERGY WASTE IN CASE OF FACE/BYPASS DAMPER STUCK AT 70% FACE MODE

This input file is used to estimate the total heating and cooling energy consumption in case face/bypass damper actuator failure occurs.

The input conditions for determining the reduction in energy waste when the face/bypass damper is stuck at 70% face mode is provided in the Section "Given Input Conditions" in the spreadsheet (Rows 1-6). From the known temperature rise across preheat coil, the air temperature exiting preheat coil is found using the formula

$$T_{PH} = T_{OA} + \Delta T \qquad \textit{Eq. E.2.1}$$

Based on the preheat temperature (T_{PH}), outside air temperature (T_{OA}) and the airflow through the coil (V_F) and bypassing the coil (V_B), the exiting mixed air temperature is calculated in Columns 3 and 6 and for Rows 46-54 (R46-R54) using the formula mentioned in Row 20 (R20).

$$V_F * T_{PH} + V_B * T_{OA} = V_{TOT} * T_{MA} \qquad \textit{Eq. E.2.2}$$

The total heating energy consumption for a each temperature bin is calculated in Columns 4 and 7 (C4 and C7) and for Rows 45-54 (R45-R54) using the airflow through the coil (V_F) and the temperature rise across the coil (ΔT) for a given outside air temperature as mentioned below

$$Heating\ Energy\ Consumption\ (Btu/hr)\ =\ 1.08\ *\ Airflow\ through\ the\ coil\ *\ Temperature\ Rise\ Across\ the\ coil$$

$$Heating\ Energy\ Consumption\ (Btu/hr) = 1.08\ *\ \mathrm{V_F} * \Delta \mathrm{T} \qquad \textit{Eq. E.2.3}$$

$$Total\ Heating\ Energy\ Consumption\ (MMBtu) = \frac{Heating\ Energy\ Consumption\ (\frac{Btu}{hr}) *\ No.of\ Hrs}{10^6} \qquad \textit{Eq. E.2.4}$$

Similarly the total cooling energy for each temperature is calculated in Columns 5 and 8 (C5 and C8) and for Rows 46- 55 (R46-R55) based on the total airflow and the discharge air temperature setpoint (from Rows 3 and 6) and the mixed air temperature calculated in Columns 3 and 6 (C3 and C6)

$$Cooling\ Energy\ Consumption\ \left(\frac{Btu}{hr}\right) = 1.08\ *\ V_{TOT}\ *\ (T_{DA}\ - T_{MA}) \qquad \textit{Eq. E.2.5}$$

$$Total\ Cooling\ Energy\ Consumption\ (MMBtu) = \frac{Cooling\ Energy\ Consumption\ (\frac{Btu}{hr}) *\ No.of\ Hrs}{10^6} \qquad \textit{Eq. E.2.6}$$

	C1	C2	C3
R2	**Given Input Conditions**	Value	Units
R3	Design Airflow	7,800	cfm
R4	Damper Stuck	70%	
R5	Airflow through the coil (70%)	5,460	cfm
R6	Discharge Air Temperature Setpoint	55	°F

	C1	C2	C3
R8	**Variables**	**Description**	**Units**
R9	$\mathbf{T_{OA}}$	Outside Air Temperature	°F

R10	**T_{DA}**	Discharge Air Temperature Setpoint	°F
R11	**T_{MA}**	Mixed Air Temperature	°F
R12	**ΔT**	Temperature Rise across Preheat Coil	
R13	**T_{PH}**	Temperature Exiting Preheat Coil	°F
R14	**V_F**	Airflow Through the Coil	cfm
R15	**V_B**	Airflow Bypassing the Coil	cfm
R16	**V_{TOT}**	Design Airflow for the AHU	cfm

	C1	C2	C3
R19	**Formulae**		
R20	$T_{PH} = T_{OA} + \Delta T$		Eq. E.2.1
R21	$V_F * T_{PH} + V_B * T_{OA} = V_{TOT} * T_{MA}$		Eq. E.2.2
R22	Heating Energy Consumption (Btu/hr) = 1.08 * V_F* ΔT		Eq. E.2.3
R33	Total Heating Energy Consumption (MMBtu) = Heating Energy Consumption * No. of Hrs /10^6		Eq. E.2.4
R24	Cooling Energy Consumption (Btu/hr) = 1.08 * V_{TOT} * (T_{DA} -T_{MA})		Eq. E.2.5
R25	Total Cooling Energy Consumption (MMBtu) = Cooling Energy Consumption * No. of Hrs /10^6		Eq. E.2.6

	C1	C2	C3	C4	C5	C6	C7	C8
	Bin Data		**5 psig Steam**			**15 psig Steam**		
	Average OAT (F)	**# of Hrs.**	**Average MAT(F)**	**Heating Energy Consumption (MMBtu)**	**Cooling Energy Consumption (MMBtu)**	**Average MAT (F)**	**Heating Energy Consumption (MMBtu)**	**Cooling Energy Consumption (MMBtu)**
R46	20	2	49	0	(0.1)	52	1	(0.1)
R47	24	14	52	3	(0.3)	55	4	0.0
R48	28	114	56	26	0.5	59	29	3.6
R49	32	223	59	51	7.5	62	57	13.5
R50	36	220	62	49	13.8	66	55	19.8
R51	40	201	66	44	18.4	69	49	23.9
R52	44	394	69	84	47.6	73	95	58.4
R53	48	489	73	102	73.3	76	115	86.7
R54	52	598	76	123	107.0	80	138	123.5
R55	**Total**	**2255**	**-**	**483**	**268**	**-**	**542**	**329**

E.3 INPUT FILE FOR REDUCED ENERGY WASTE IN CASE OF STEAM VALVE FAILURE

This input file is used to estimate the total heating and cooling energy consumption in case steam valve failure occurs.

The input conditions for determining the reduction in energy waste when the steam valve fails is provided in the Section “Given Input Conditions” in the spreadsheet (Rows 1-4). From the known temperature rise across preheat coil, the air temperature exiting preheat coil is found using the formula

$$T_{PH} = T_{OA} + \Delta T \qquad \textit{Eq. E.3.1}$$

The total heating energy consumption for each temperature bin is calculated in Columns 4 and 7 (C4 and C7) and for Rows 40-63 (R40-R63) using the total airflow through the coil (V_{TOT}) and the temperature rise across the coil (ΔT) for a given outside air temperature as mentioned below

$$Heating\ Energy\ Consumption\ (Btu/hr)$$
$$= 1.08 * Air\ flow\ through\ the\ coil\ *\ Temperature\ Rise\ Across\ the\ coil$$

$$\text{Heating Energy Consumption (Btu/hr)} = 1.08 * V_{TOT} * \Delta T \qquad \textit{Eq. E.3.2}$$

$$Total\ Heating\ Energy\ Consumption\ (MMBtu)$$
$$= \frac{Heating\ Energy\ Consumption\ (\frac{Btu}{hr}) *\ No.of\ Hrs}{10^6} \qquad \textit{Eq. E.3.3}$$

Similarly the total cooling energy for each temperature is calculated in Columns 5 and 8 (C5 and C8) for Rows 40-63 (R40-R63) based on the total airflow and the discharge air temperature setpoint (from Rows 3 and 6) and the preheat air temperature calculated in Columns 3 and 6 (C3 and C6)

$$Cooling\ Energy\ Consumption\ \left(\frac{Btu}{hr}\right) = 1.08 * V_{TOT} * (T_{DA} - T_{PH}) \quad Eq.\ E.3.4$$

$$Total\ Cooling\ Energy\ Consumption\ (MMBtu) = \frac{Cooling\ Energy\ Consumption\ (\frac{Btu}{hr}) * No. of\ Hrs}{10^6} \quad Eq.\ E.3.5$$

	C1	C2	C3
R1	**Given Input Conditions**	**Value**	**Units**
R2	Design Airflow	7,800	cfm
R3	% of time Steam Valve Fails	10%	
R4	Discharge Air Temperature Setpoint	55	°F

	C1	C2	C3
R6	**Variables**	**Description**	**Units**
R7	$\mathbf{T_{OA}}$	Outside Air Temperature	°F
R8	$\mathbf{T_{DA}}$	Discharge Air Temperature Setpoint	°F
R9	$\mathbf{\Delta T}$	Temperature Rise across Preheat Coil	
R10	$\mathbf{T_{PH}}$	Temperature Exiting Preheat Coil	°F
R11	$\mathbf{V_{TOT}}$	Design Airflow for the AHU	cfm

	C1	C2
R28	**Formulae**	
R29	$T_{PH} = T_{OA} + \Delta T$	Eq. E.3.1
R30	Heating Energy Consumption (Btu/hr) = 1.08 * V_{TOT}* ΔT	Eq. E.3.2
R31	Total Heating Energy Consumption (MMBtu) = Heating Energy Consumption * No. of Hrs /10^6	Eq. E.3.3

R32	Cooling Energy Consumption (Btu/hr) = 1.08 * V_{TOT} * (T_{DA} -T_{PH})	Eq. E.3.4
R28	Total Cooling Energy Consumption (MMBtu) = Cooling Energy Consumption * No. of Hrs /10^6	Eq. E.3.5

	C1	C2	C3	C4	C5	C6	C7	C8
	Bin Data		**5 psig Steam**			**15 psig Steam**		
R39	**Average OAT**	**Number of Hrs.**	**Average DAT**	**Heating Energy Consumption (MMBtu)**	**Cooling Energy Consumption (MMBtu)**	**Average DAT**	**Heating Energy Consumption (MMBtu)**	**Cooling Energy Consumption (MMBtu)**
R40	-	-	-	-	-	-	-	-
R41	20	0	61	0	(0.0)	65	0	0.0
R42	24	1	64	0	(0.0)	69	1	0.2
R43	28	11	67	4	0.3	72	4	1.6
R44	32	22	71	7	1.3	75	8	3.8
R45	36	22	74	7	2.0	78	8	4.3
R46	40	20	77	6	2.5	82	7	4.5
R47	44	39	80	12	6.3	85	14	9.9
R48	48	49	83	15	9.5	88	16	13.6
R49	52	60	87	17	13.6	91	20	18.2
R50	56	57	90	16	14.9	94	18	19.0
R51	60	50	93	14	14.6	98	16	17.8
R52	64	63	96	17	20.7	101	19	24.3
R53	68	106	100	28	38.4	104	32	43.7
R54	72	102	103	26	40.4	107	30	44.9
R55	76	75	106	19	32.3	110	22	35.0
R56	80	64	109	16	29.6	114	18	31.5
R57	84	50	112	12	24.8	117	14	26.0
R58	88	49	116	11	26.2	120	13	27.0
R59	92	23	119	5	13.2	123	6	13.4
R60	96	8	122	2	4.8	126	2	4.9
R61	100	1	125	0	0.8	130	0	0.8
R62	104	2	128	0	1.1	133	0	1.1
R63	108	0	132	-	-	136	-	-
R64	Total			**236**	**297**		**269**	**345**

E.4 INPUT FILE FOR PROCEDURE TO ESTIMATE STEAM PRESSURE IN A COIL AND ENERGY SAVINGS FOR IMPROVED CONTROL SEQUENCE 6.2.2

This input file is used to estimate the steam pressure in a coil and the corresponding energy savings for the improved control sequence involving a modulating valve. The input conditions for determining the steam pressure are provided in the Section "Given Input Conditions" in the spreadsheet (Rows 1-8). The valve positions and the corresponding temperature rise are known and given in Rows 15-19.

The air flow temperature is converted into Celsius scale and the preheat temperature is calculated. This scale conversion is done to facilitate the estimation of log mean temperature differential.

$$T_C = 5/9 * (T_F - 32) \qquad \textit{Eq. E.4.1}$$

$$T_{PH} = T_{EAT} + \Delta T \qquad \textit{Eq. E.4.2}$$

For a given outside air temperature at various steam valve position the log mean temperature difference (ΔT_{LMTD}) in Column 2 (C2) for Rows 52 -65 (R52-R65) is then calculated by equating the heat transfer from the coil to the heat gained by the air stream making use of the relation given in Row 38 (R38).

$$1.08*V_{TOT} * (T_{LAT} - T_{EAT}) = U * A * \Delta T_{LMTD} \qquad \textit{Eq. E.4.3}$$

Based on the calculated log mean temperature differential, the corresponding temperature of steam in the coil is then calculated in Column 5 (C5) for Rows 52-65 (R52-R65) by using the below mentioned relation.

$$\Delta TLMTD = (T_{LAT} - T_{EAT})/ln\{(T_{SP} - T_{LAT})/(T_{SP} - T_{EAT})\} \qquad \textit{Eq. E.4.5}$$

The steam pressure is computed from the steam tables using this saturation temperature value.

The total heating energy consumption for each temperature bin is calculated in Columns 5, 9 and 13 (C5, C9 & C13) for Rows 37-56 (R37-R56) using the total airflow through the coil (V_{TOT}) and the temperature rise across the coil (ΔT) for a given outside air temperature as mentioned below

$$Heating\ Energy\ Consumption\ (Btu/hr)$$

$$= 1.08 * Air\ flow\ through\ the\ coil * Temperature\ Rise\ Across\ the\ coil$$

$$Heating\ Energy\ Consumption\ (Btu/hr) = 1.08 * \mathrm{V_{TOT}} * \Delta\mathrm{T} \qquad Eq.\ E.4.6$$

$$Total\ Heating\ Energy\ Consumption\ (MMBtu)$$

$$= \frac{Heating\ Energy\ Consumption\ (\frac{Btu}{hr}) * No.of\ Hrs}{10^6} \qquad Eq.\ E.4.7$$

Similarly the total cooling energy for each temperature is calculated in Column 5 and 8 (C5 and C8) and for Rows 40-63 (R40-R63) based on the total airflow and the discharge air temperature setpoint (from Rows 3 and 6) and the preheat air temperature calculated in Columns 3 and 6 (C3 and C6) .

$$Cooling\ Energy\ Consumption\ \left(\frac{Btu}{hr}\right) = 1.08 * V_{TOT} * (T_{DA} - T_{PH}) \qquad Eq.\ E.4.8$$

$$Total\ Cooling\ Energy\ Consumption\ (MMBtu)$$

$$= \frac{Cooling\ Energy\ Consumption\ (\frac{Btu}{hr}) * No.of\ Hrs}{10^6} \qquad Eq.\ E.4.9$$

	C1	C2	C3
R1	**Given Condition**	**Value**	**Units**
R2	Design OA Flow	11,030	cfm
R3	UA	5,937	
R4	Entering Air Temp (T_{EAT})	32.5	°F
R5	T_{PHS}	45	°F
R6	T_{DAS}	48	
R7	Coil Pressure for fully open valve	10.7	psig
R8	T_{SP} @ 10.7psig	240.9	°F

	C1	C2
R14	**Valve Position**	**Temperature Rise (°F)**
R15	30	5.0
R16	40	8.0
R17	60	21.0
R18	70	28.0
R19	100	50.4

	C1	C2	C3
R21	**Variables**	**Description**	**Units**
R22	T_F	Temperature in Fahrenheit	°F
R23	T_C	Temperature in Celsius	°F
R24	T_{EAT}	Entering Air Temp	°F
R25	T_{LAT}	Leaving Air Temp	°F
R26	V_{TOT}	Design Airflow rate	cfm
R27	T_{PHS}	Preheat Temp Setpoint	°F
R28	T_{DAS}	Discharge Air Temp Setpoint	°F
R29	ΔT	Temperature Rise Across Coil	°F
R30	T_{PH}	Preheat Temp	°F
R31	T_{DA}	Discharge Air Temp	°F
R32	ΔT_{LMTD}	Log Mean Temperature Differential	°F

	C1	C2	C3
R21	**Variables**	**Description**	**Units**
R33	T_{SP}	Saturated Steam Temp	°F

R35	**Formulae**	
R36	$T_C = 5/9 * (T_F - 32)$	Eq. E.4.1
R37	$T_{PH} = T_{EAT} + \Delta T$	Eq. E.4.2
R38	Heat Gained by Air Stream = Heat Transferred by Steam Coil	
R39	$1.08*V_{TOT} * (T_{LAT} - T_{EAT}) = U * A * \Delta T_{LMTD}$	Eq. E.4.3
R40	$1.08*V_{TOT} * (T_{LAT} - T_{EAT})/\{U * A\} = \Delta T_{LMTD}$	Eq. E.4.4
R41	$\Delta T_{LMTD} = (T_{LAT}-T_{EAT})/\ln\{(T_{SP} -T_{LAT})/(T_{SP}-T_{EAT})\}$	Eq. E.4.5
R42	$\exp \wedge\{(T_{LAT} - T_{EAT})/\Delta T_{LMTD}\} = (T_{SP} - T_{LAT})/(T_{SP} - T_{EAT})$	
R43	Let $X= \exp \wedge\{(T_{LAT} - T_{EAT})/\Delta T_{LMTD} \}$	
R44	$T_{LAT} - X *T_{EAT} = T_{SP} * (1-X)$	
R45	$(T_{LAT} - X *T_{EAT})/ (1-X) = T_{SP}$	
R46	Heating Energy Consumption (Btu/hr) = 1.08 * V_{TOT} * ΔT	Eq. E.4.6
R47	Total Heating Energy Consumption (MMBtu) = Heating Energy Consumption (Btu/hr) * No. of Hrs/10^6	Eq. E.4.7
R48	Cooling Energy Consumption (Btu/hr) = 1.08 * V_{TOT} * $(T_{PH}-T_{DAS})$	Eq. E.4.8
R49	Total Cooling Energy Consumption (MMBtu) = Cooling Energy Consumption (Btu/hr) * No. of Hrs/10^6	Eq. E.4.9

	C1	C2	C3	C4	C5
R51	Valve Position	LMTD	X	Steam Temperature (°C)	Steam Temperature (°F)
R52	60	56	1.25	62	144.1
R53	70	69	1.27	77	171.1
R54	80	81	1.28	91	196.1
R55	90	91	1.29	104	219.3
R56	91	93	1.29	105	221.5
R57	92	94	1.30	107	223.7
R58	93	95	1.30	108	225.9

R59	94	96	1.30	109	228.1
R60	95	97	1.30	110	230.3
R61	96	98	1.30	111	232.4
R62	97	99	1.30	113	234.6
R63	98	100	1.30	114	236.7
R64	99	101	1.31	115	238.8
R65	100	102	1.31	116	240.9

	C1		C2	C3	C4 C5	C6		C7
					Original Control			
	Dry Bulb Temp. Bin Data		**# of hours**	**Average Temp (F)**	**Temperature Rise (F)**	**Heating Energy Consumption (MMBtu)**	**Total Cooling (MMBtu)**	**Total Energy Consumption (MMBtu)**
R37	5	10	0	7.5	-	0	0	0
R38	10	15	2	12.5	50.4	1	0	1
R39	15	20	38	17.5	50.4	22	8	31
R40	20	25	120	22.5	50.4	70	34	104
R41	25	30	169	27.5	50.4	99	57	156
R42	30	35	388	32.5	50.4	226	155	381
R43	35	40	621	37.5	7.5	55	0	55
R44	40	45	624	42.5	2.5	19	0	19
R45	45	50	633	47.5	0	0	0	0
R46	50	55	801	52.5	0	0	43	43
R47	55	60	631	57.5	0	0	71	71
R48	60	65	990	62.5	0	0	171	171
R49	65	70	841	67.5	0	0	195	195
R50	70	75	789	72.5	0	0	230	230
R51	75	80	915	77.5	0	0	322	322
R52	80	85	687	82.5	0	0	282	282
R53	85	90	373	87.5	0	0	176	176
R54	90	95	101	92.5	0	0	54	54
R55	95	100	37	97.5	0	0	22	22
R56	**Total**		**8760**	**-**	**-**	**493**	**1820**	**2313**

	C1		C2	C3	C8	C9	C10	C11
	Dry Bulb Temp. Bin Data		**# of hours**	**Average Temp (F)**	**Improved Control**			
					Temperature Rise (F)	**Heating Total (MMBtu)**	**Cooling Energy (MMBtu)**	**Total Energy (MMBtu)**
R37	5	10	0	7.5	-	0	0	0
R38	10	15	2	12.5	42.3	1	0	2
R39	15	20	38	17.5	42.3	23	9	32
R40	20	25	120	22.5	42.3	72	36	108
R41	25	30	169	27.5	42.3	101	60	162
R42	30	35	388	32.5	42.3	233	161	394
R43	35	40	621	37.5	7.5	55	0	55
R44	40	45	624	42.5	2.5	19	0	19
R45	45	50	633	47.5	0	0	0	0
R46	50	55	801	52.5	0	0	43	43
R47	55	60	631	57.5	0	0	71	71
R48	60	65	990	62.5	0	0	171	171
R49	65	70	841	67.5	0	0	195	195
R50	70	75	789	72.5	0	0	230	230
R51	75	80	915	77.5	0	0	322	322
R52	80	85	687	82.5	0	0	282	282
R53	85	90	373	87.5	0	0	176	176
R54	90	95	101	92.5	0	0	54	54
R55	95	100	37	97.5	0	0	22	22
R56	**Total**		**8760**	**-**		**435**	**1763**	**2198**

					C12	C13	C14	C15
	Dry Bulb Temp. Bin Data		# of hours	Average Temp (F)	Heating Hot Water Control			
					Temperature Rise (F)	Heating Total (MMBtu)	Cooling Energy (MMBtu)	Total Energy (MMBtu)
R37	5	10	0	7.5	-	0	0	0
R38	10	15	2	12.5	32.5	1	0	1
R39	15	20	38	17.5	27.5	12	0	12
R40	20	25	120	22.5	22.5	32	0	32
R41	25	30	169	27.5	17.5	35	0	35
R42	30	35	388	32.5	12.5	58	0	58
R43	35	40	621	37.5	7.5	55	0	55
R44	40	45	624	42.5	2.5	19	0	19
R45	45	50	633	47.5	0	0	0	0
R46	50	55	801	52.5	0	0	43	43
R47	55	60	631	57.5	0	0	71	71
R48	60	65	990	62.5	0	0	171	171
R49	65	70	841	67.5	0	0	195	195
R50	70	75	789	72.5	0	0	230	230
R51	75	80	915	77.5	0	0	322	322
R52	80	85	687	82.5	0	0	282	282
R53	85	90	373	87.5	0	0	176	176
R54	90	95	101	92.5	0	0	54	54
R55	95	100	37	97.5	0	0	22	22
R56	**Total**		**8760**	**-**		**212**	**1566**	**1778**

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