OPTIMIZATION OF VAV AHU TERMINAL BOX MINIMUM AIRFLOW

A Thesis

by

WEI WANG

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

August 2011

Major Subject: Mechanical Engineering

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Approved by:

Chair of Committee,	Michael Pate
Committee Members,	Warren Heffington
	Jorge L. Alvarado
Head of Department,	Jerald A. Caton

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ABSTRACT

Optimization of VAV Air Handling Unit (AHU) Terminal Box Minimum Airflow.

(August 2011)

Wei Wang, B.S., Jilin University

Chair of Advisory Committee: Dr. Michael Pate

Determining the optimal terminal box airflow is a complex process which is influenced by various factors, such as weather condition, supply air temperature, primary air fraction and internal load. A guideline for determination of a cost efficient minimum airflow setpoint for variable air volume (VAV) terminal box units is drawn in this research. The most efficient optimal minimum airflow setpoint should not be a fix setting, but should be changing with zone load and ventilation requirement.

A fixed minimum airflow is used in conventional control strategies. The terminal box minimum airflow required is not a constant since the supply air temperature, fresh air fraction and zone load are different. It is important to set up the minimum airflow to ensure indoor air quality (IAQ) and thermal comfort and to minimize energy consumption.

Analysis has been carried out to compare how the supply air temperature, fresh air fraction and zone load affect the minimum airflow setting of an exterior zone. And 30% of design airflow is not always a good number, and may cause comfort issue or ventilation problem. If the minimum airflow is set higher than required, terminal boxes

will have significantly simultaneous heating and cooling, and consume more fan power in the AHUs. If the minimum airflow is set lower than required, IAQ will be a concern. Energy saving ratio study is conducted to estimate the energy saving benefit by implementing an optimized minimum airflow.

DEDICATION

To my parents

ACKNOWLEDGEMENTS

I would like to thank my committee chair, Dr. Michael Pate, and my committee members, Dr. Warren Heffington, Dr. Jorge L. Alvarado and Dr. Wei Zhan, for their guidance and support throughout the course of this research. Thanks to all of my professors here at Texas A&M, and all of my past professors at the Jilin University, whose time, teachings and guidance have made this opportunity possible.

Thanks also go to Song Deng and Qiang Chen for providing the opportunity to join the Energy Systems Laboratory and develop my engineering skills. I also want to extend my gratitude to Chen Xu, Dave Dong and other colleagues in Vision Building Energy Efficiency, LLC. , who were always available for discussion and suggestion.

Finally, thanks to my family and friends for their support and encouragement.

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1. INTRODUCTION

Variable air volume (VAV) air-handling units (AHUs) are the widely used in commercial buildings in the U.S.A. Terminal boxes are the critical component of the VAV systems. The minimum airflow of terminal boxes has a great impact on comfort, indoor air quality (IAQ) and energy cost.

Usually a fixed minimum airflow is used in conventional control strategies. The constant setpoint is selected to satisfy the basic ventilation requirements, but this control sequence can cause occupant comfort issue or waste excessive energy. If the minimum airflow is set higher than required, terminal boxes will have significantly simultaneous heating and cooling, and consume more fan power in the AHUs. If the minimum airflow is set lower than required, indoor air quality (IAQ) will be a concern. The terminal box minimum airflow required is not a constant since the zone load, supply air temperature and fresh air fraction of the supply air are different. It is important to set up the minimum airflow to ensure IAQ and thermal comfort and to minimize energy consumption.

This thesis documents a study to optimize the minimum airflow setting for VAV AHU terminal boxes and analyzes the corresponding savings ratio under different operation scenarios.

This thesis follows the style of *ASHRAE Transactions.*

The study was divided into five main sections. The first section introduces the research background and objective. Literature review is documented after the introduction. Terminal box structure and working principles were reviewed; methods to determine terminal box minimum airflow were compared. The third section describes the methodology used in this study to determine the minimum airflow and the fourth section develops an analysis of the impact of the key influencing factors: supply air temperature, primary outdoor air fraction, and occupancy (zone load). Also, related energy savings ratios for various operation scenarios are investigated in this section. The last section is a conclusion of the whole study.

2. LITERATURE REVIEW

2.1 Introduction

This section gives an overview of the major topics affecting the study of single duct VAV terminal box minimum airflow reset. This literature review covers the following areas: terminal boxes and control strategies, ventilation requirements and determining the minimum airflow.

2.2 Terminal Box

VAV terminal units are available in many configurations, all of which control the space temperature by varying the volume of cool supply air from the air handler to match the actual cooling load. VAV terminal units are fitted with automatic controls that are either pressure-dependent or pressure-independent (ASHRAE. 2008).

2.2.1 Single Duct VAV Pressure Dependent Terminal Box (with Reheat, and without Reheat)

"Successful early VAV product research and development in the United States led to the production of energy efficient terminal boxes in the 1960s and the genesis of the system as we know it today"(Smeed, 2007). But it was not until the creation of a ceiling diffuser capable of maintaining satisfactory room air distribution at low air volume flow rates in the 1960s, that a variety of VAV systems became rapidly popularized (Chen, 1995).Pressure-dependent units control damper position in response to room temperature, and flow may increase and decrease as the main duct pressure varies. Figure 2-1 shows the schematic of SDVAV terminal box without reheat.

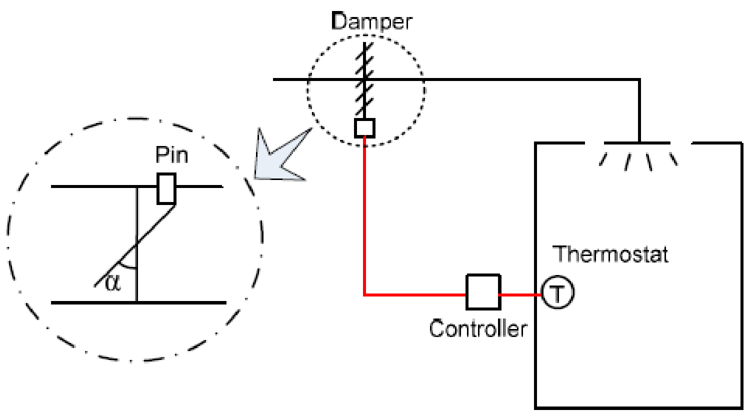


Figure 2-1 Schematic diagram of single-duct variable volume pressure dependent terminal box without reheat

2.2.2 Single Duct VAV Pressure Independent Terminal Box with Reheat

Researcher explained pressure independent controls are a necessity for reheat type terminal boxes. But velocity pressure is so low that it is not enough for reliable operation of the velocity controller when the unit is in reheat mode (Avery 1989). Stein (2005) talked about the importance of zone level control when manufacturing VAV boxes. The first decision concerns which type of box to use includes cooling only, reheat, series fan-powered, and parallel fan-powered boxes.

Pressure-independent units' measure actual supply airflow and control flow in response to room temperature. Figure 2-2 shows the schematic of SDVAV terminal box with reheat.

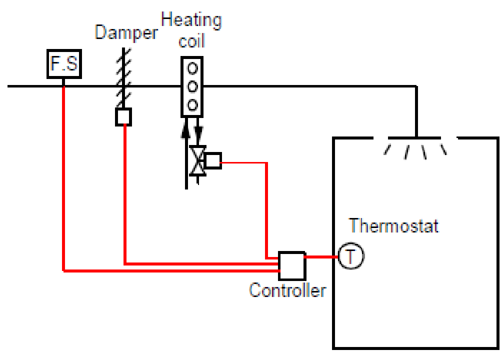


Figure 2-2 Schematic diagram of single-duct variable volume pressure dependent terminal box with reheat

2.2.3 Dual Duct VAV box

Dual duct systems became popular in the 1950s and 1960s in the states when energy price was not much a concern. The energy usage was not an important factor in designing HVAC systems until the energy crisis in the 1970s. The dual duct systems had inherent advantages that made them popular at that time. They provided good air circulation and excellent humidity. They required little or no maintenance. Pipe corrosion or leakage problem through occupied area can be prevented because only conditioned air is introduced to zones.

2.3 Minimum Airflow Requirements

According to the standard control sequence of the Single Duct VAV terminal box, the airflow requires a minimum limit, which should determine the fixed minimum airflow setpoint. Two categories are used to determine the terminal box minimum airflow setpoint. One is the principle method based on the standard, and another category is the common practice methods.

2.3.1 Based on ASHRAE Standard

For VAV reheat terminal boxes that serve exterior zones, the minimum airflow setpoint is typically selected by the maximum value of the following (ASHRAE. 2004):

- The airflow required by the room design heating load, or
- The minimum required for ventilation.

Problems come out when plan layouts change, and most of the time it is not practical to calculate the value for each terminal box with different supply air temperature, primary air fractions and zone load conditions. So, it is difficult to determine an accurate fixed minimum airflow set point with the principle method.

2.3.2 Engineering Practice

Zone thermostatic controls shall be capable of operating in sequence the supply of heating and cooling energy to the zone. Such controls shall prevent reheating, recooling, mixing or simultaneously supplying air that has been previously mechanically heated and air that has been previously cooled, either by mechanical cooling or by economizer; and other simultaneous operation of heating and cooling system to the same zone.

ASHRAE Standard 90.1 has three exceptions that certain level of simultaneous heating and cooling is allowed. According to this, the terminal box minimum airflow rate can be determined by the greatest of: (1) 30% of the maximum rate of airflow and 300 cfm; (2) the volume of outdoor air required to meet the ventilation requirements; (3) 0.4 cfm/ ft2 of air-conditioned floor area within the zone.

This common practice can be seen in many facility, new design or existing building. The building load is dynamic, which means a fixed setpoint most of the time is not optimized. And the values are erroneous, if they are not adjusted to the real situation of the room. (Liwerant, 2008).

2.4 Current Advance Methods

The conventional fixed minimum airflow setpoint is not optimized as the system is dynamic, causing either energy waste or comfort issue. Researchers and engineers have introduced some methods and try to solve the problem.

2.4.1 Fan-Powered Terminal Box

Fan powered terminal box has a major advantage that it can provide high air circulation during low cooling/heating load scenario by using a small and integral fan in the terminal box. There are two different types of fan-powered terminal boxes, series fan-powered terminal box and the parallel fan-powered terminal box presented in Figure 2-3.

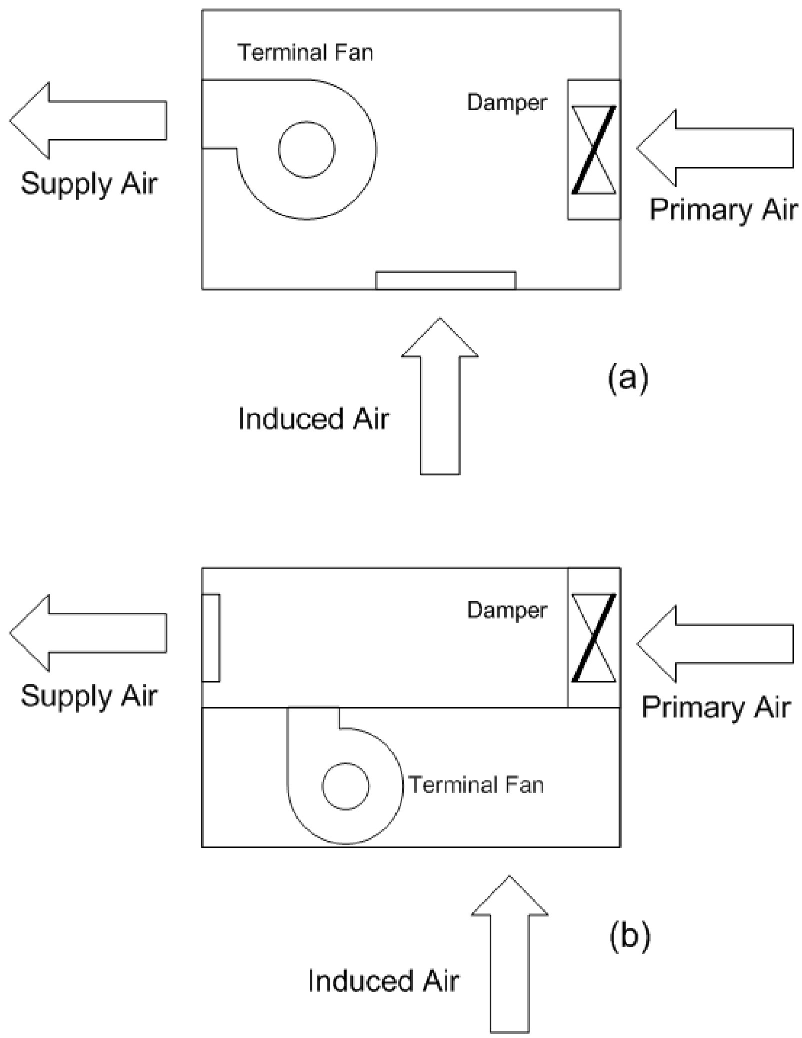


Figure 2-3 Generic VAV fan powered terminal unit design (a) series (b) parallel

Fan powered terminal boxes save reheating energy by using plenum return air, circulating air from over-ventilated zone and shutting off the main AHUs. However, fan powered box solved the problem, it has disadvantages. It's more complicated with an individual fan in each box, and malfunction of fans will cause major comfort issue in the zone. Fan noise is also a concern, which needs extra sound proofing resolution. It requires high initial investment or retrofit costs and extra maintenance.

2.4.2 VAV Adjustable Thermal Diffuser

A new product, VAV adjustable thermal diffuser is developed by some manufactures. It is a self-contained unit, includes airflow inducer, solid center panel, heating slave, warm-up sensor, disc, actuators, and VAV heating and cooling set point adjustments. Figure presents a schematic diagram of the VAV Adjustable Thermal Diffuser. Room air is induced past the thermal elements which sense and adjust air flow based upon space requirements. The diffuser shall vary supply air volume to provide both VAV heating and VAV cooling. They are thermally powered by using two room temperature sensing elements, and two supply air sensing elements. One room sensing element shall sense room temperature and vary the support air when cooling. The other room sensing element shall sense room temperature and vary the supply air when heating. The changeover between the heating and the cooling modes is determined by the supply air temperature.

Figure 2-4 presents a schematic diagram of the VAV Adjustable Thermal Diffuser. The benefits of VAV adjustable thermal diffuser are obvious. It is a self-contained, and no electrical or pneumatic connections are required. It has low cost installation and operation. It can maintain a high velocity even at the low load condition. It has separate set points for VAV cooling and heating, and easy access to adjust the setpoints. (Cranes, 2005)

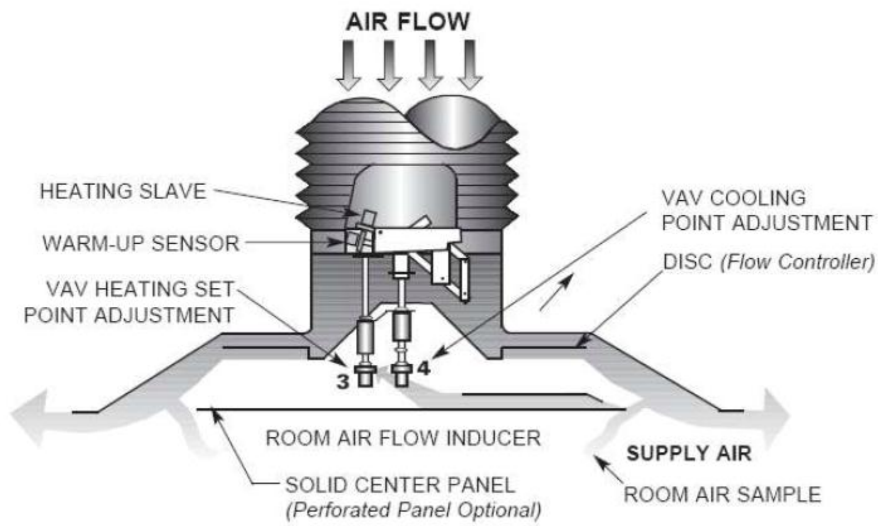


Figure 2-4 VAV adjustable thermal diffuser (Crane SFPV diffuser)

2.4.3 CO2 -Based Demand-Controlled Ventilation (DCV)

Though previous developed methods can solve the air circulation issue, they don't actually meet the fresh air requirement. Demand-Controlled Ventilation (DCV) was researched to control the boxes and solve the issue. CO2-Based DCV refers to the control of using carbon dioxide concentrations as an indicator of room occupant ventilation rate. Generally, the higher the physical exertion an activity entails, the greater the amount of carbon dioxide the body will produce. Lots of research has been done on the control strategies using CO2 based Demand-Controlled Ventilation, and it's a promising solution to the problem.

2.5 Summary

In this section, terminal box structure and working principles were reviewed; methods to determine terminal box minimum airflow were compared. It's easier to use the common engineering practice to determine the minimum airflow. But to make the minimum flow sufficient for variable zone load conditions and ventilation requirements, it's common that minimum setpoint is higher than enough which causes excessive energy cost. It's recommended that new construction to use advanced methods to make the system perform more efficiently. And to those existing buildings, renovation and upgrade sometimes are not doable due to the high capital cost. So it would be helpful to find the lowest minimum setpoint based on zone load and ventilation requirement.

3. DETERMINE MINIMUM AIRFLOW

This section is intended to describe the method used in this study to obtain the optimal setting and compare the energy consumption driven by changing minimum outside airflow ratios. EQuest Version 3-64 program was used as the main simulation program. In the research, the optimal terminal box minimum airflow setpoint is determined based on heating load and ventilation requirement. The changes of the heating/cooling energy and fan power consumption under different minimum outside airflow ratios are also compared.

3.1 Minimum Airflow Based on Room Heating Load

Room heating load is affected by outside air condition, solar gains, building materials, internal loads, and some more factors. In the research, room load is assumed to have a linear relationship with outside air temperature (ASHRAE. 2005).

The minimum airflow based on room heating load should meet the building heating load requirement, which can be calculated by Equation 3-1,

$$V_{\min\_h} = \frac{Q_h}{c \cdot \rho \cdot (T_s - T_{room})} \tag{3-1}$$

where,

$V_{\min\_h}$ − Air volumetric flow rate for heating load, ft3/min

Q_h − Room heating load, Btu/hr

c − Specific heat of air, Btu/lbm °F

ρ − Standard air density, lbm/ft3

T_s − Discharge air temperature, °F

T_r − Room air temperature, °F

With a simplified relationship between outside air temperature and room load (BTU/hr), the exterior zone of a clinic (out-patient) is simulated in the following part.

3.2 Building Load Simulation

In order to evaluate the changes of energy consumption driven by minimum airflow setting, a dynamic building simulation model was developed using the eQuest software.

3.2.1 Introduction to eQuest

eQuest is a sophisticated, yet easy to use analysis tool that provides professional-level results with an affordable level of effort. It combines schematic and design development building creation wizards, an energy efficiency measure (EEM) wizard and a graphical results display module with a complete up-to-date DOE-2 (version 2.2) building energy use simulation program. (U.S. Department of Energy. 2010)

3.2.2 Location and Weather File

The simulated building is located in Newport News, VA, (37.132^0,-76.493^0). Real weather files for Newport News, VA were used for estimate the building load. The average daily outdoor dry bulb temperature of 2009 ranges from 21°F to 87°F. The annual average temperature is 61.4°F. Figure 3-1 below is monthly average OAT for Newport News in 2009.

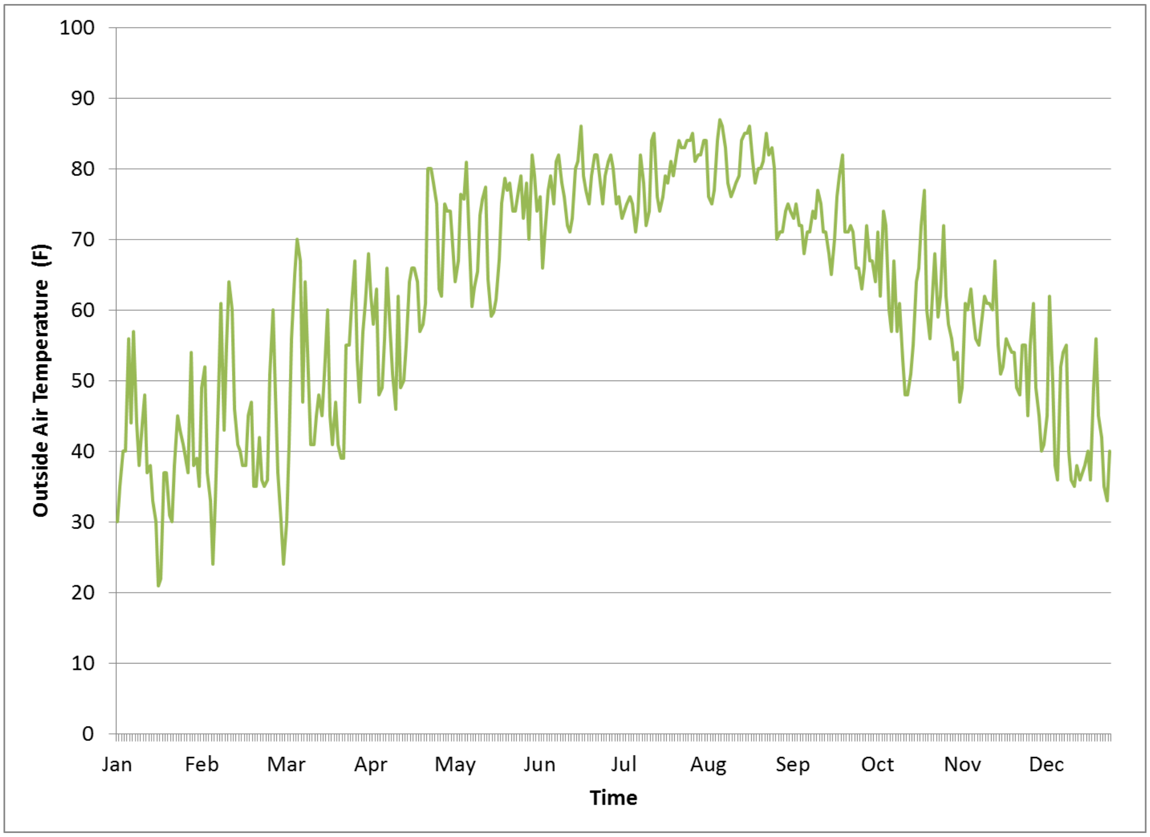
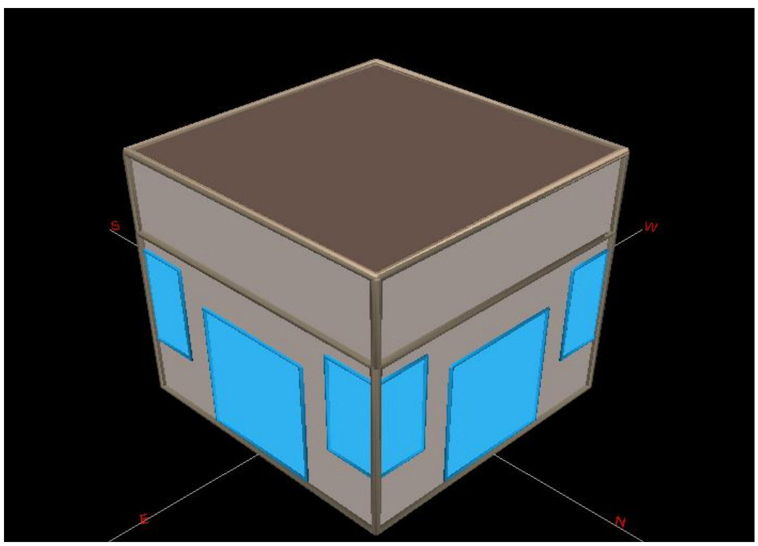


Figure 3-1 Outside air temperature for Newport News 2009

3.2.3 Building Envelope Information

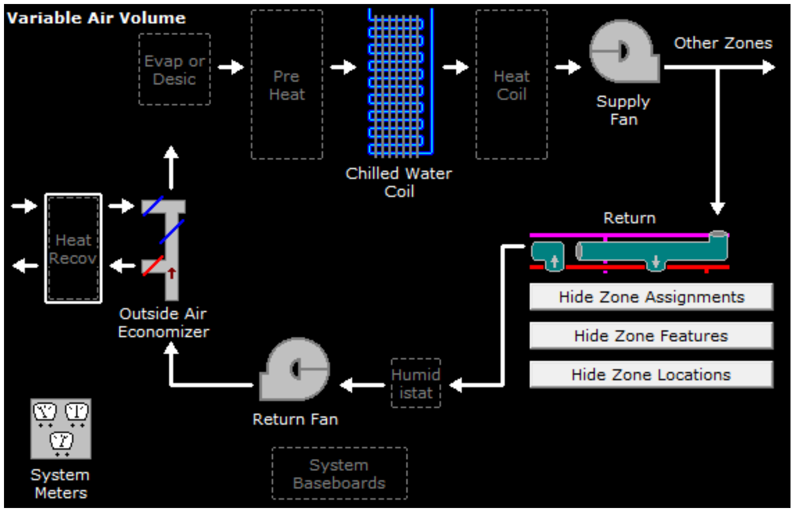
The building used as a simulation model is a one-story Health, Medical Clinic building, with only one exterior conditioned zone. It includes a return plenum with a 169 ft2 floor area, 1521 ft3 volume.

The heating and cooling system consists of one SDVAV systems with terminal reheat boxes serving the conditioned space. The AHU supply fan modulates the fan speed based on the static pressure setpoint. The cooling valve in the cooling coil modulates to maintain SAT at the setpoint. Dampers and heating valves in the terminal box adjust the air flow and heating energy to maintain the room temperature setpoint. Figure 3-2 displays the simulated one room building, which makes the whole zone exterior zone, and the HVAC system.



(a) Building envolop

Figure 3-2 Schematic diagram of building model for simulation



(b) HVAC system

Figure 3-2 Continued

Table 3-1 and Table 3-2 display the detailed input data for building wall materials, and operation assumptions.

Table 3-1 Detail of constructions

CONSTRUCTION NAME	U-VALUE (BTU/HR-SQFT-F)	SURFACE ABSORPTANCE	SURFACE ROUGHNESS INDEX
Exterior Wall Construction	0.084	0.6	1
Roof Construction	0.046	0.6	1
Ceiling Construction	0.514	0.7	3
Interior Wall Construction	0.402	0.7	3
Interior Floor Construction	0.941	0.7	3
UFCons (G.1.U2)	0.244	0.7	3

Table 3-2 Simulation input data of internal gain

Item	Input Data
People	115 W/person
	200 ft2/person
Lighting	1.4 W/ft2
Electrical Equipment	1.5 W/ft2

Figure 3-3 to Figure 3-5 presents the hourly lighting schedule, hourly equipment schedule and hourly occupancy schedule.

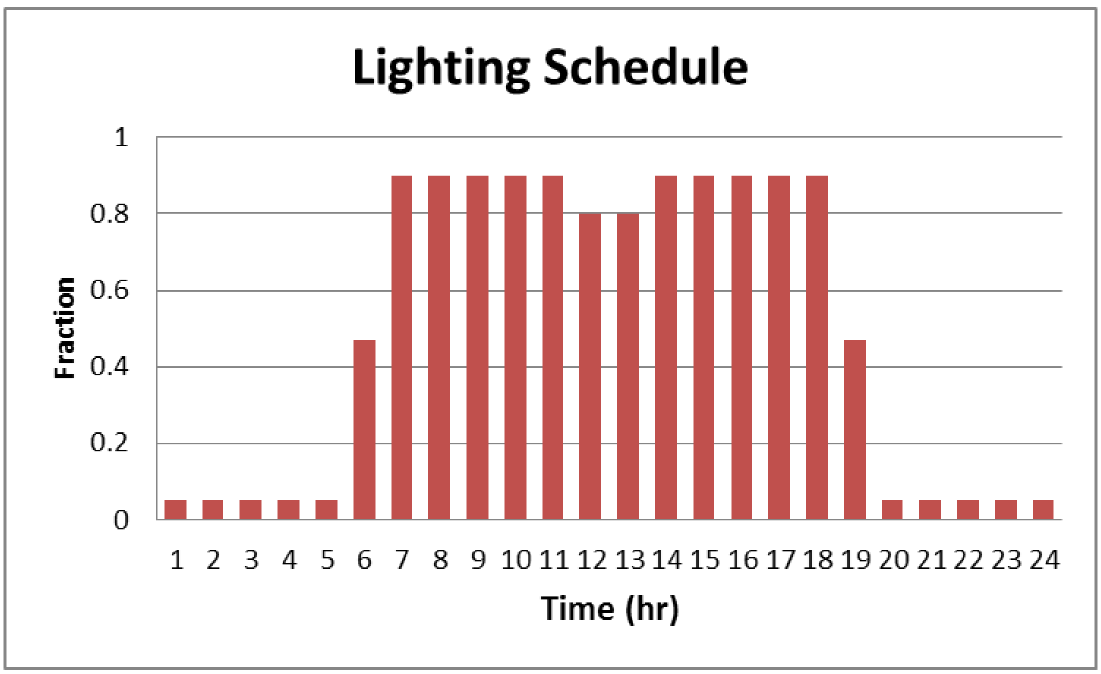


Figure 3-3 Hourly lighting schedule

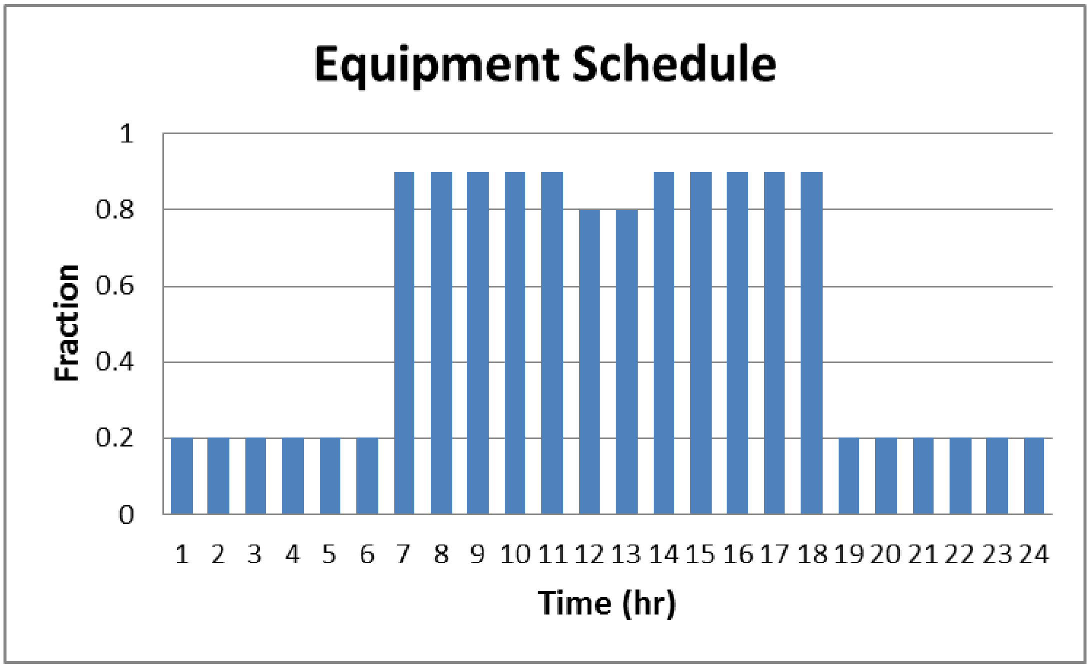


Figure 3-4 Hourly equipment schedule

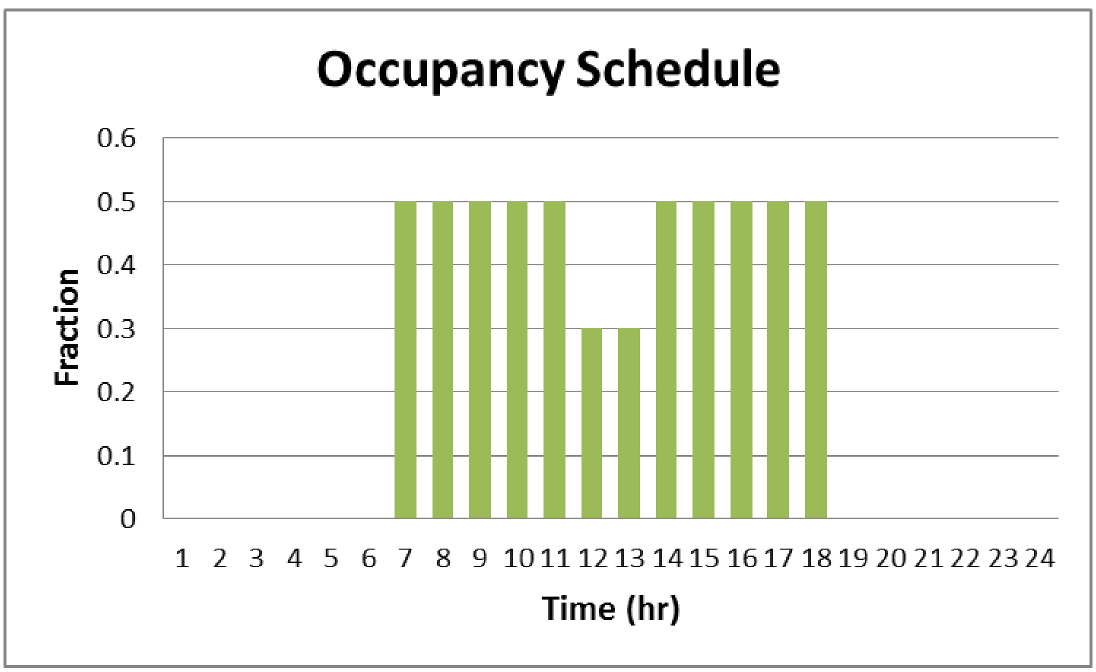


Figure 3-5 Hourly occupancy schedule

3.2.4 Building Envelope Load and Airflow

The simulated zone is an exterior zone, and the room heating load is linear to the outside air temperature. Figure 3-6 displays the room load changes with outside air temperature under different occupancy in a typical exterior clinic room. Figure 3-7 displays required minimum airflow based on design heating load, when zone is 50% occupied and SAT=120°F.

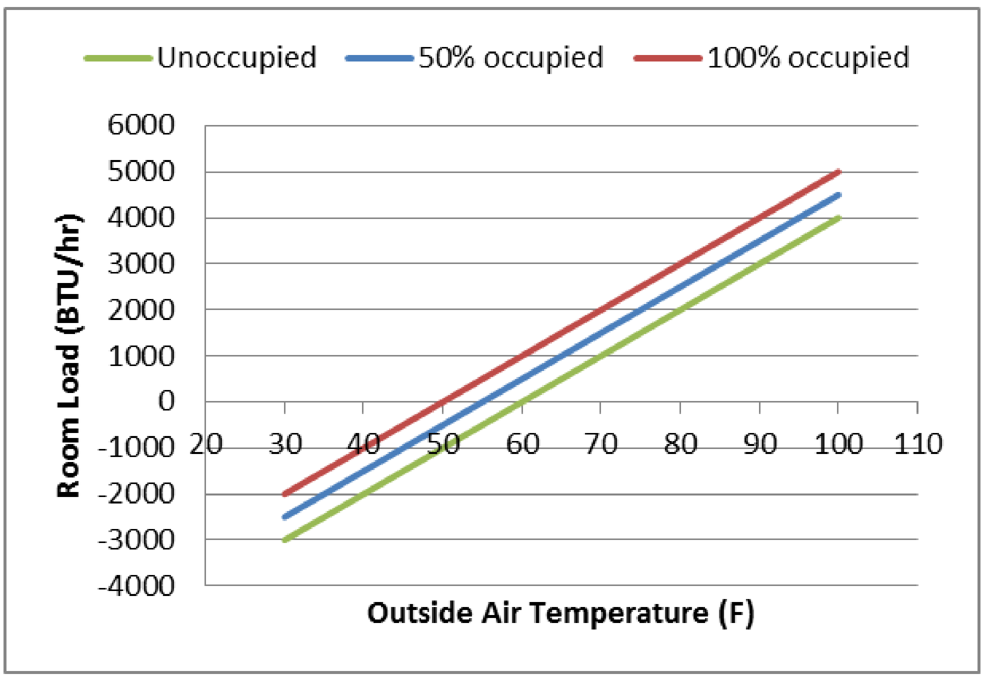


Figure 3-6 Room load changes with different occupancy in a typical exterior clinic room

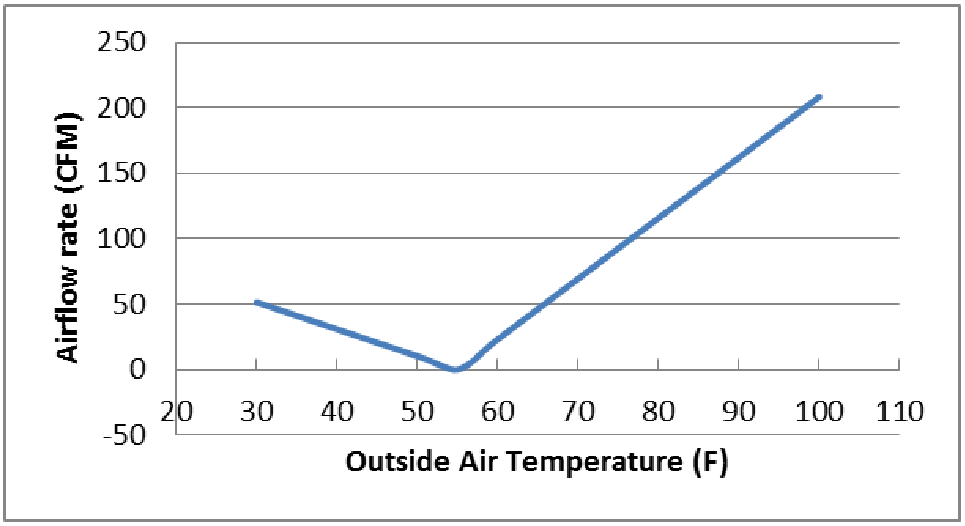


Figure 3-7 Required minimum airflow based on design heating load, when zone is 50% occupied and SAT=120°F

3.3 Minimum Airflow Based on Ventilation Requirement

Before 2001, ASHRAE standard determines the required ventilation rates based on either the number of occupants in the zone (cfm/person) or the floor area of the zone (cfm/ft2). From 2004, Standard 62.1-2004 prescribed new minimum breathing zone ventilation rates and a new calculation procedure to find the minimum intake airflow needed for different ventilation systems.

$$V_{bz} = R_p * P_z + R_a * A_z \tag{3-2}$$

where,

V_{bz} – breathing zone outdoor air flow, ft3/min

R_p – outdoor airflow rate required per person

P_z – zone population, person

$\mathrm{R_a}$ – outdoor airflow rate required per unit area

A_z − zone floor area

Zone outdoor airflow can be calculated using Equation 3-3,

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{3-3}$$

where,

V_{oz} − zone outdoor air flow, ft3/min

E_z − zone air distribution effectiveness

The zone air distribution effectiveness (Ez) shall be determined using Table 3-3 (Table 6-2, ASHRAE standard 62.1-2004)

Table 3-3 Zone air distribution effectiveness E_z

Air Distribution Configuration	E_z
Ceiling supply of cool air	1.0
Ceiling supply of warm air and floor return	1.0
Ceiling supply of warm air 15F (8C)or more above space temperature and ceiling return	0.8
Ceiling supply of warm air less than 15F (8C) above space temperature and ceiling return provided that the 150 fpm (0.8 m/s) supply air jet reaches to within 4.5 ft (1.4m) of floor level.	1.0
Floor supply of cool air and ceiling return provided that the 150 fpm (0.8 m/s) supply air jet reaches 4.5 ft (1.4m) or more above the floor.	1.0
Floor supply of cool air and ceiling return provided low velocity displacement ventilation achieves unidirectional flow and thermal stratification.	1.2
Floor supply of warm air and floor return	1.0
Floor supply of warm air and ceiling return	0.7
Makeup supply drawn in on the opposite side of the room from the exhaust and/or return	0.8
Makeup supply drawn in near to the entrance and/or return location	0.5

The minimum rate of airflow can be calculated using Equation 3-4,

$$Z_{pz} = Min\left(Max\left(\frac{T_R - T_S}{T_R - T_{oa}}, minoa\right), 1\right) \tag{3-4}$$

where,

Z_{pz} − primary outdoor air fraction, %

T_R − return air temperature, °F

T_S − outside air temperature, °F

T_{oa} − supply air temperature, °F

Zone primary airflow can be calculated using Equation 3-5,

$$V_{pz} = \frac{V_{oz}}{Z_{pz}} \tag{3-5}$$

where,

V_{pz} − the zone primary airflow, including outdoor air and recirculated air

The outdoor air intake V_{ot} (for AHU) can be calculated as Equation 3-6 ~ 3-8,

$$V_{ou} = D\sum\nolimits_{all\ zones} R_p * P_z + \sum\nolimits_{all\ zones} R_a * A_z \tag{3-6}$$

$$D = \frac{P_S}{\sum_{all\ zones} P_z} \tag{3-7}$$

$$V_{ot} = \frac{V_{ou}}{E_v} \tag{3-8}$$

V_{ou} − uncorrected outdoor air intake, ft3/min

D − Occupant Diversity

P_s − The design system population

E_v − system ventilation efficiency

V_{ot} − outdoor air intake, ft3/min

Figure 3-8 displays the zone minimum airflow based on ventilation requirement.

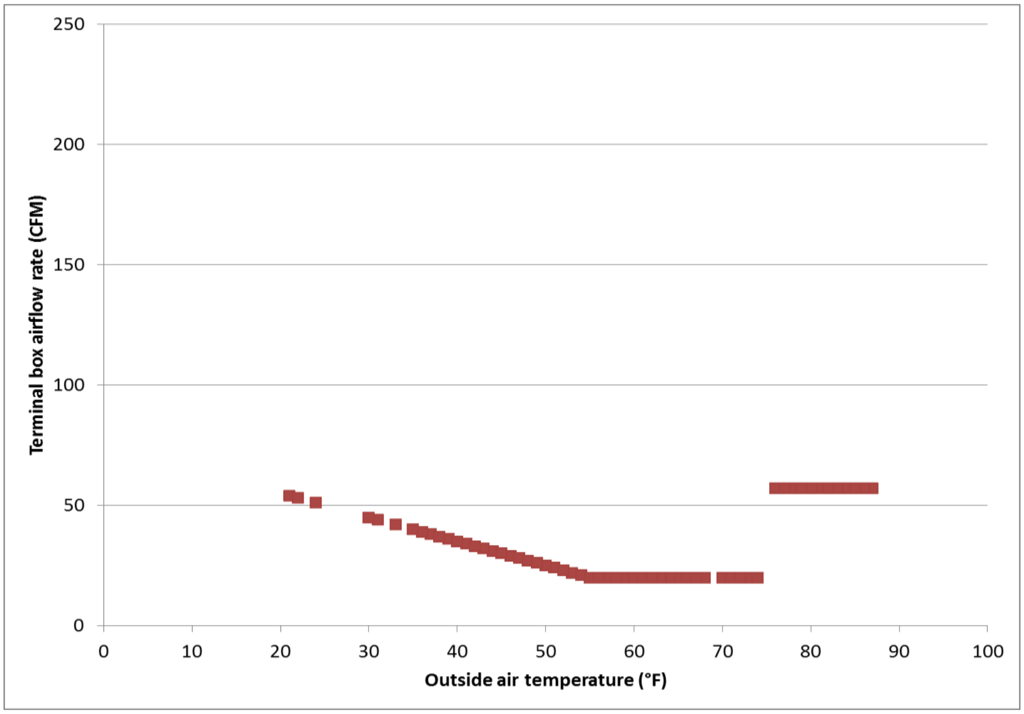


Figure 3-8 The zone minimum airflow based on ventilation requirement

3.4 Determined Minimum Airflow

The determined terminal box minimum airflow should be set at no less than the highest airflow required by the design heating load or ventilation requirement. Figure 3-9 and Figure 3-10 display the conventional fix minimum airflow setting and optimized minimum airflow setting under fully occupied and unoccupied conditions, with 30% primary air fraction. When the outside air temperature is low, the heating load required by the zone determines the minimum airflow setting. As the temperature goes up, the airflow based on the ventilation requirement becomes greater than that of the heating load. Once the outside air temperature goes up to certain level, the cooling requires more airflow, which can bring enough fresh air into the zone to meet the ventilation requirement. And once again, the airflow is determined by the zone load. It is obvious that fixed minimum airflow causes excessive airflow during wild weather.

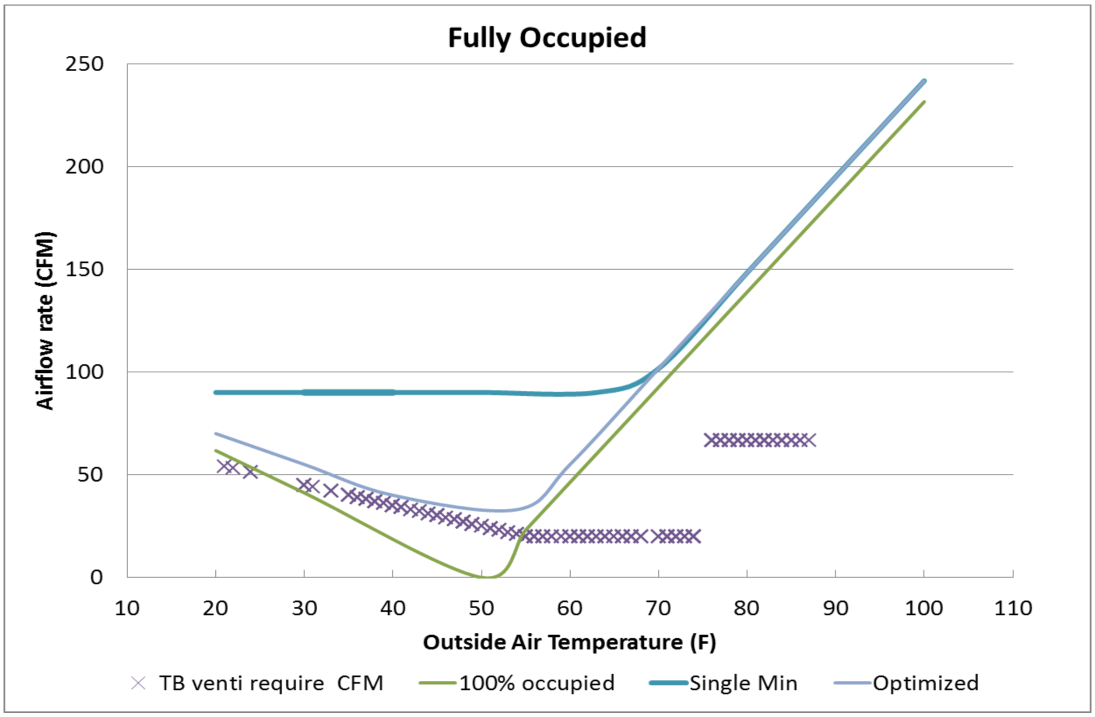


Figure 3-9 Determined minimum airflow when zone is fully occupied

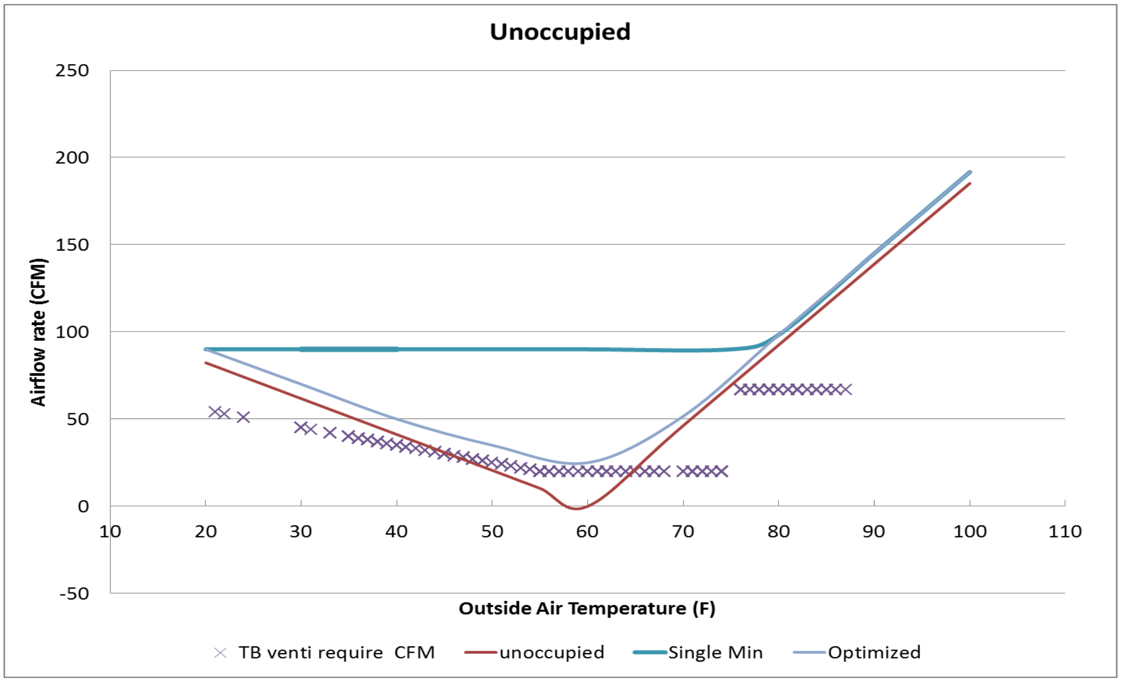


Figure 3-10 Determined minimum airflow when zone is unoccupied

4. DIFFERENT OPERATION CONDITIONS AND SAVING ANALYSIS

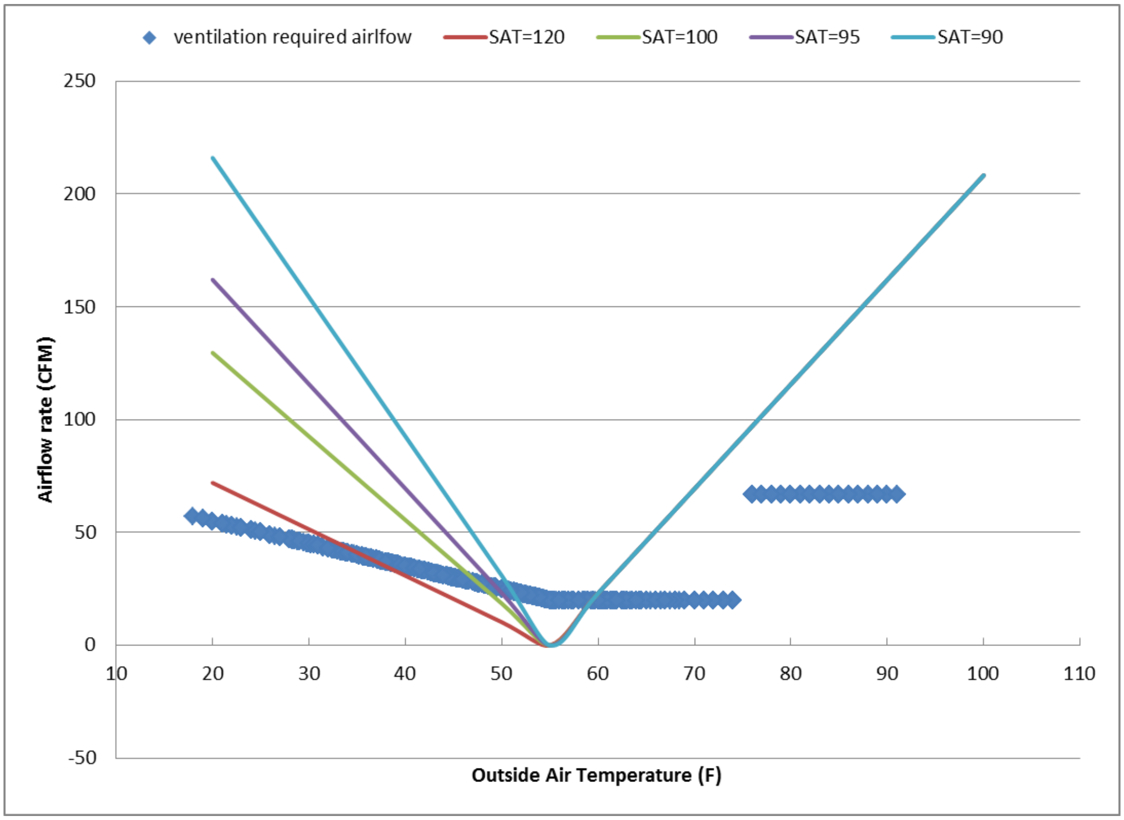
4.1 Different Operating Comparison

4.1.1 Influence of Supply Air Temperature

This section analyzes the changes of the optimal minimum airflow driven by changing SAT reset. To analyze this, the SAT has been changed from 120°F to 100°F, 95°F and 90°F while all the other input parameters were kept the same. Zone is 50% occupied, with 30% outside air in the AHU.

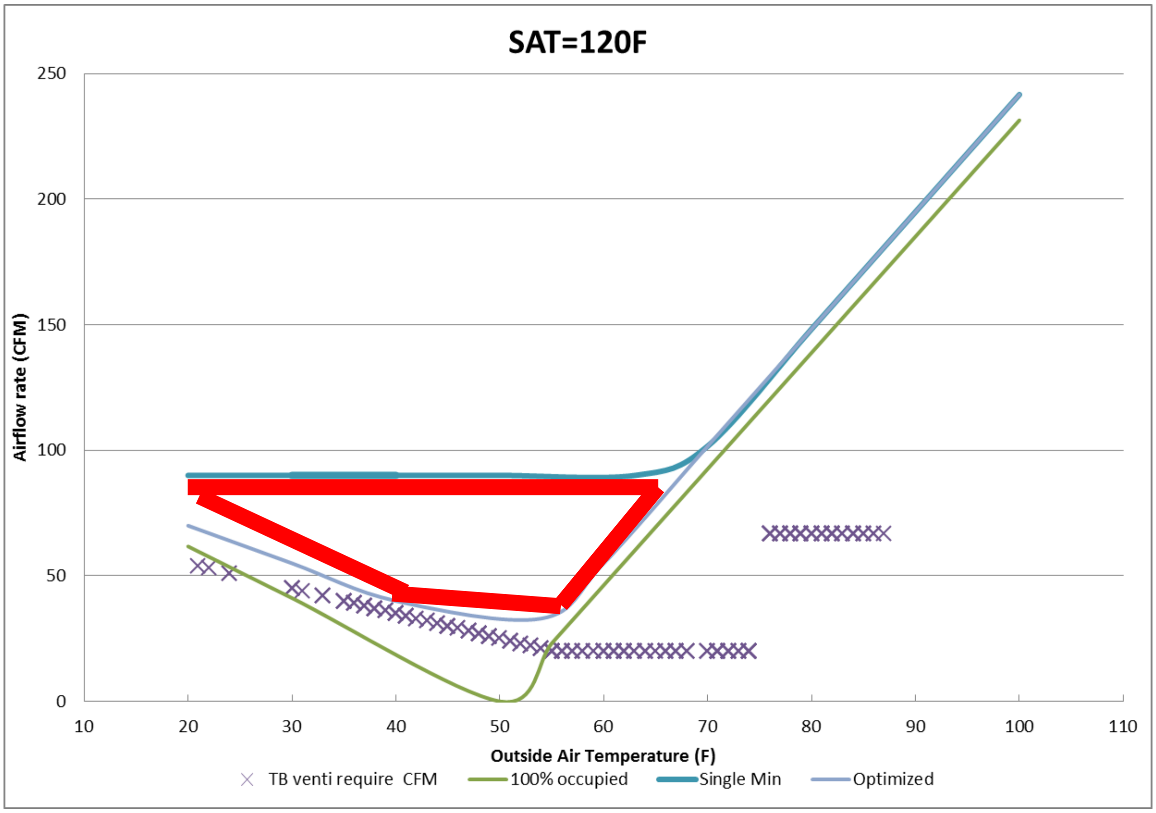
Figure 4-1 displays the optimized minimum airflow for different supply air temperatures. As the AHU supply air temperature changes, the heating capacity of the air handling unit changes, which requires different amounts of minimum airflow from the terminal boxes. When the supply air temperature is low at 90°F, the terminal box will require more aiflow, which not only cause the conventional fixed minimum ariflow setting significant high wasting energy at partial load, and cost more fan power energy in the AHU at the same time.

From the figures, it is reasonable to conclude that reset the supply air temperature from 100°F to 120°F will lead to significant saving. So, it’s important to find the proper supply air temperature reset schedule to maximize energy efficiency.



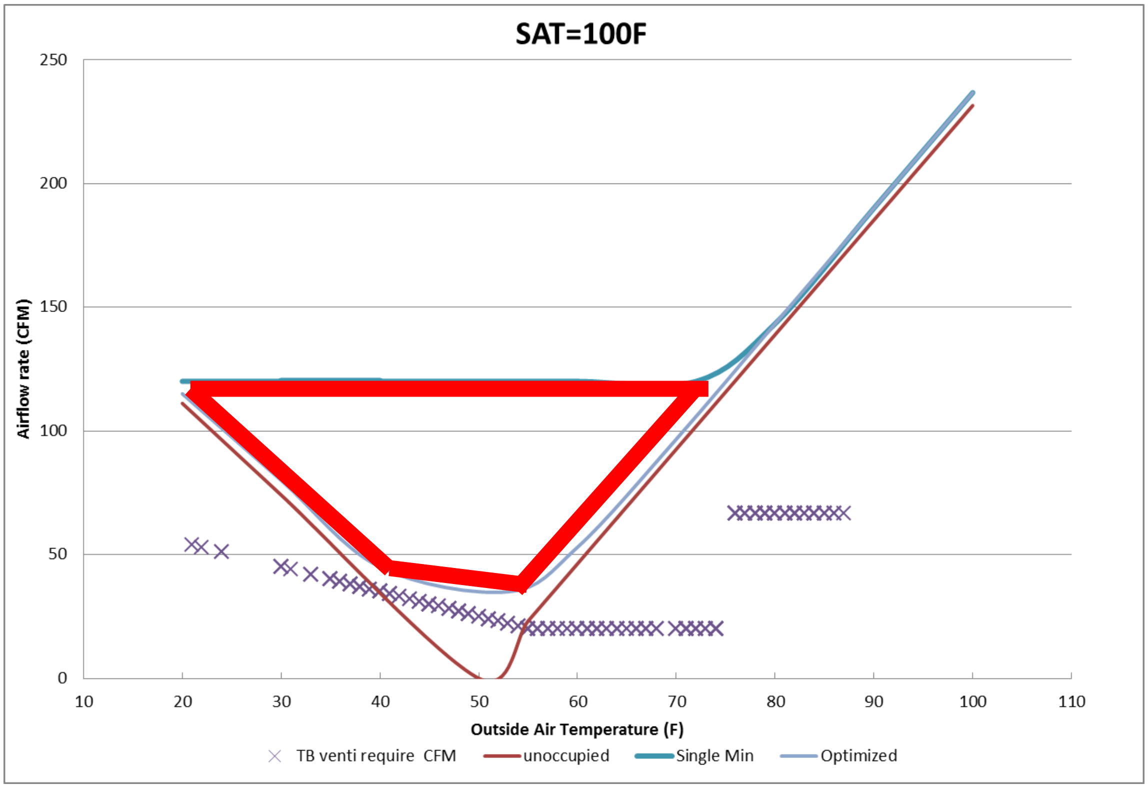
(a) Supply air temperature varies

Figure 4-1 Optimized minimum airflow for different supply air temperature



(b) When supply air temperature is 120F

Figure 4-1 Continued



(c) When supply air temperature is 100F

Figure 4-1 Continued

4.1.2 Influence of Primary Outdoor Air Fraction

This section analyzes the changes of the optimal minimum airflow driven by changing primary air fraction. To analyze this, the primary outdoor air fraction has been changed from 50% to 40%, 30% and 20% while all the other input parameters were kept the same. Zone is 50% occupied, and supply air temperature is 120°F.

In Figure 4-2, when the AHU primary air fraction is set too low at 20%, the system will have potential ventilation problem even if minimum airflow satisfies zone cooling load at hotter weather around 80°F. When the zone is unoccupied, the cooling load will be even lower which make the potential ventilation problem even worse. So, it's important to make sure room has enough fresh air under any load conditions, and check the outside air damper routinely in case damper is stuck at a low percentage position.

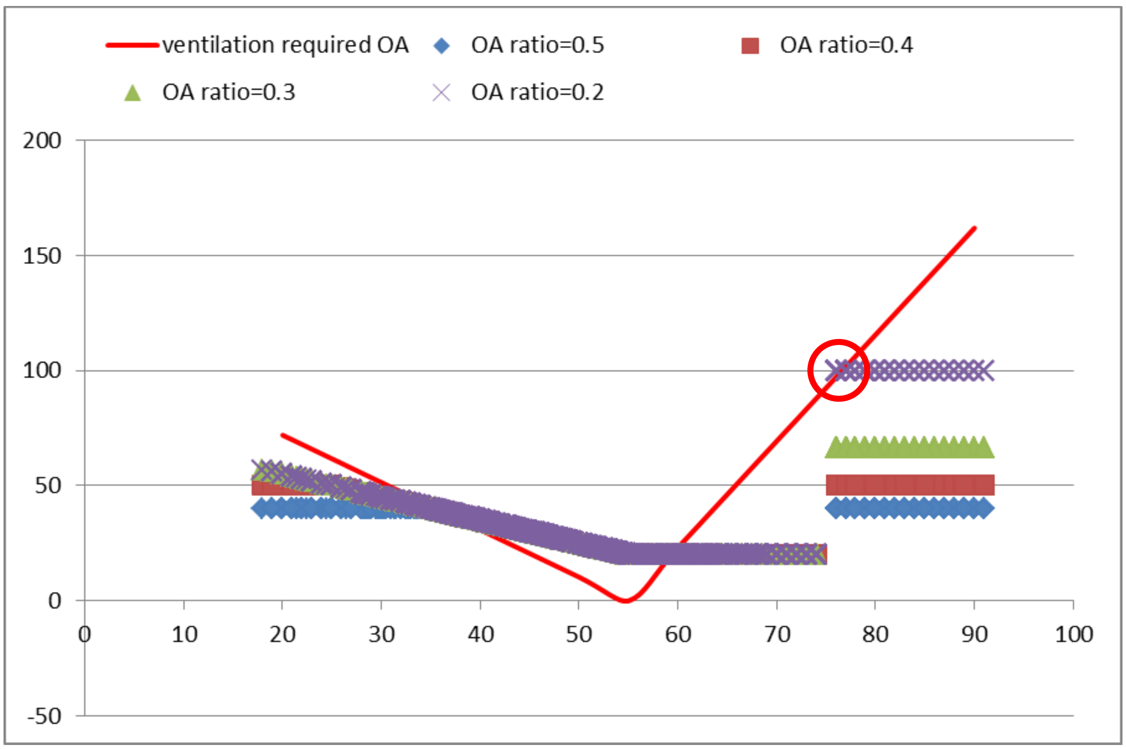
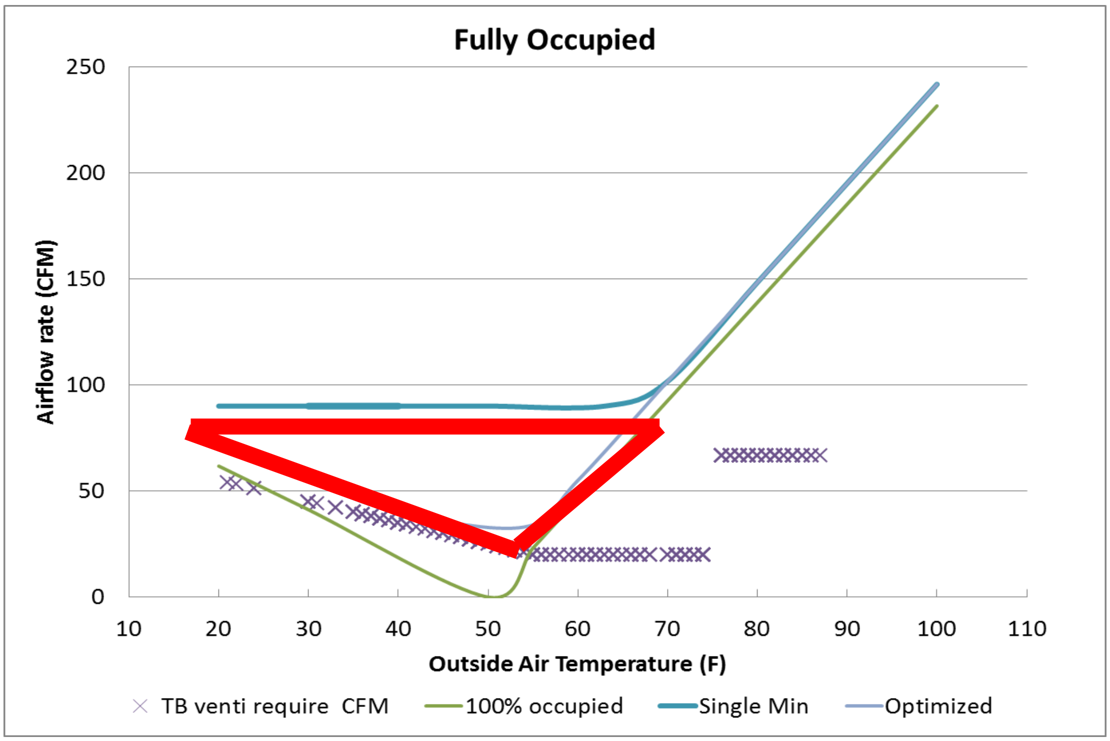


Figure 4-2 Optimized minimum airflow for different primary air fraction

4.1.3 Influence of Zone Load

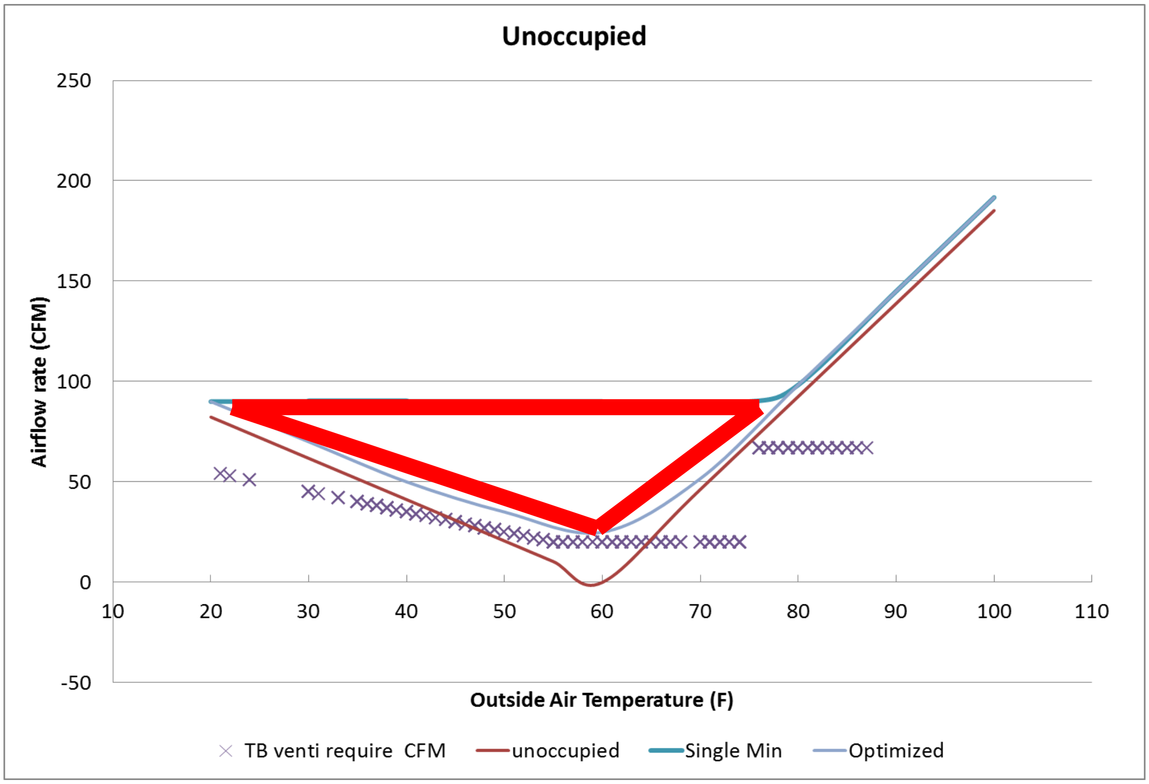
This section analyzes the changes of the optimal minimum airflow driven by changing zone load. To analyze this, the zone has been changed from 100% occupied to unoccupied while all the other input parameters were kept the same. Zone is primary air fraction is 30%, and supply air temperature is 120°F.

During fully occupied conditions, the ventilation requirement serves as a more critical factor; during unoccupied conditions, minimum airflow is determined more by the zone load requirement.



(a) Fully occupied conditions

Figure 4-3 Optimized minimum airflow for different occupancy conditions



(b) Unoccupied conditions

Figure 4-3 Continued

4.2 Energy Analysis

To compare the rate of energy savings of fixed single minimum airflow (base case), and optimized minimum airflow (optimized case), numerical analysis was conducted. Analysis was based on 2009 daily weather data and a simplified steady state model.

The fixed single minimum airflow ratio α_{fix} can be calculated by Equation 4-1,

$$\alpha_f = \frac{CFM_{min,f}}{CFM_d} \tag{4-1}$$

where,

α_f − fixed minimum airflow ratio, %

$CFM_{min,f}$ − Fixed air volumetric flow rate, ft3/min

CFM_d − Design air volumetric flow rate, ft3/min

The heating load ratio β_h can be calculated by Equation 4-2,

$$\beta_h = \frac{Q_h}{Q_{h,d}} \tag{4-2}$$

where,

β_c − cooling load ratio

Q_c − Room cooling load, Btu/hr

$Q_{c,d}$ − Room design cooling load, Btu/hr

The cooling load ratio β_c can be calculated by Equation 4-3,

$$\beta_c = \frac{Q_c}{Q_{c,d}} \tag{4-3}$$

where,

β_c − cooling load ratio

Q_c − Room cooling load, Btu/hr

$Q_{c,d}$ − Room design cooling load, Btu/hr

The terminal box minimum airflow ratio for heating requirement heating can be calculated by Equation 4-4,

$$\alpha_h = \frac{CFM_h}{CFM_d} = \frac{CFM_{min,f}}{CFM_d} \cdot \beta_h = \alpha_f \cdot \beta_h \tag{4-4}$$

where,

α_h − minimum airflow ratio for heating requirement, %

$\mathrm{CFM_{min,f}}$ – Fixed air volumetric flow rate, ft3/min

$\mathrm{CFM_d}$ – Design air volumetric flow rate, ft3/min

The terminal box minimum airflow ratio for ventilation requirement α_{vent} can be calculated by Equation 4-5,

$$\alpha_{vent} = \frac{CFM_{vent}}{CFM_d} \tag{4-5}$$

where,

α_{vent} – minimum airflow ratio for ventilation requirement, %

$\mathrm{CFM_{vent}}$ – air volumetric flow rate for ventilation requirement, ft3/min

The variable minimum airflow ratio α_{min} can be calculated by Equation 4-6,

$$\alpha_{min} = Max(\alpha_h, \alpha_{vent}) \tag{4-6}$$

where,

α_{min} – optimized minimum airflow ratio , %

Energy consumption for base case

The cooling energy consumption can be calculated by Equation 4-7,

$$\begin{cases} E_c = m\Delta h = 60 \cdot CFM_d \cdot \rho \cdot \beta_c \cdot (h_{ma} - h_{sa}), & \beta_c > \alpha_f \\ E_c = m\Delta h = 60 \cdot CFM_d \cdot \rho \cdot \alpha_f \cdot (h_{ma} - h_{sa}), & \beta_c \le \alpha_f \end{cases} \tag{4-7}$$

where,

$\mathrm{E_c}$ – cooling energy consumption, Btu/hr

ρ – air density, lbm/ft3

$\mathrm{h_{ma}}$ – mixed air enthalpy, Btu/lbm

$\mathrm{h_{sa}}$ – supply air enthalpy, Btu/lbm

The heating energy consumption can be calculated by Equation 4-8 and 4-9,

$$E_h = cm\Delta T = 60 \cdot c_p \cdot CFM_d \cdot \rho \cdot \alpha_f \cdot (T_{da} - T_{sa}) \tag{4-8}$$

$$T_{da} = T_{ra} + \frac{\beta_h \cdot Q_{h,d}}{60 \cdot \rho \cdot c_p \cdot \alpha_f \cdot CFM_d} \tag{4-9}$$

where,

E_h – heating energy consumption, Btu/hr

c_p – specific heat of air, Btu/lbm°F

T_{da} – discharge air temperature, °F

T_{sa} – supply air temperature, °F

T_{ra} – return air temperature, °F

Fan power energy consumption can be calculated by Equation 4-10,

$$E_f = \frac{E_d \cdot \alpha_{base}^3}{\eta_f} \tag{4-10}$$

Energy consumption for optimized case,

The cooling energy consumption can be calculated by Equation 4-11,

$$\begin{cases} E_c = m\Delta h = 60 \cdot CFM_d \cdot \rho \cdot \beta_c \cdot (h_{ma} - h_{sa}), & \beta_c > \alpha_{min} \\ E_c = m\Delta h = 60 \cdot CFM_d \cdot \rho \cdot \alpha_{min} \cdot (h_{ma} - h_{sa}), & \beta_c \leq \alpha_{min} \end{cases} \tag{4-11}$$

The heating energy consumption can be calculated by Equation 4-12,

$$\begin{cases} E_h = cm\Delta T = 60 \cdot c_p \cdot CFM_d \cdot \rho \cdot \beta_h \cdot (T_{da} - T_{sa}), & \beta_h > \alpha_{min} \\ E_h = cm\Delta T = 60 \cdot c_p \cdot CFM_d \cdot \rho \cdot \alpha_{min} \cdot (T_{da} - T_{sa}), & \beta_h \leq \alpha_{min} \end{cases} \tag{4-12}$$

Fan power energy consumption can be calculated by Equation 4-13,

$$E_f = \frac{E_d \cdot \alpha_{opt}^3}{\eta_f} \tag{4-13}$$

Energy savings can be calculated,

The cooling energy saving can be calculated by Equation 4-14,

$$\Delta E_c = E_{c,base} - E_{c,opt} = \begin{cases} 0, & \beta_c > \alpha_f \\ 60 \cdot CFM_d \cdot \rho \cdot (\alpha_f - \alpha_{min}) \cdot (h_{ma} - h_{sa}), & \beta_c \leq \alpha_f \end{cases} \tag{4-14}$$

The heating energy saving can be calculated by Equation 4-15,

$$\Delta E_h = E_{h,base} - E_{h,opt} = \begin{cases} 0, & \beta_h > \alpha_f \\ 60 \cdot c_p \cdot CFM_d \cdot \rho \cdot (\alpha_f - \alpha_{min}) \cdot (T_{da} - T_{sa}), & \beta_h \leq \alpha_f \end{cases} \tag{4-15}$$

Fan power saving (neglect fan efficiency change) can be calculated by Equation 4-16,

$$\Delta E_f = E_{f,base} - E_{f,opt} = \frac{E_d \cdot \alpha_{base}^3 - E_d \cdot \alpha_{opt}^3}{\eta_f} \tag{4-16}$$

where,

ΔE_f – fan power, kW

E_d – design fan power, kW

α_{base} – base case supply fan airflow ratio, %

α_{opt} – optimized case supply fan airflow ratio, %

$\eta_{f,d}$ – fan efficiency

The energy saving ratios can be calculated with Equation 4-17 ~ 4-19,

$$\eta_c = \frac{E_{c,base} - E_{c,opt}}{E_{c,base}} = \begin{cases} 0, & \beta_c > \alpha_f \\ \frac{\alpha_f - \alpha_{min}}{\alpha_f}, & \beta_c \leq \alpha_f \end{cases} \tag{4-17}$$

$$\eta_h = \frac{E_{h,base} - E_{h,opt}}{E_{h,base}} = \begin{cases} 0, & \beta_c > \alpha_f \\ \frac{\alpha_f - \alpha_{min}}{\alpha_f}, & \beta_c \leq \alpha_f \end{cases} \tag{4-18}$$

$$\eta_f = \frac{E_{f,base} - E_{f,opt}}{E_{f,base}} = \begin{cases} 0, & \beta_c > \alpha_f \\ \frac{\alpha_f - \alpha_{min}}{\alpha_f}, & \beta_c \leq \alpha_f \end{cases} \tag{4-19}$$

where,

η_c − energy savings ratio for the cooling energy consumption, %

η_h − energy savings ratio for the heating energy consumption, %

η_f − energy savings ratio for the fan power consumption, %

Table 4-1 displays the changes of zone load as outside air temperature changes.

Table 4-1 Zone load based outside air temperature changes

Unoccupied		100% occupied	
q=-6000+100*OAT		q=-5000+100*OAT	
Outside Air Temperature (F)	Unoccupied Room Load (BTU/hr)	Outside Air Temperature (F)	Unoccupied Room Load (BTU/hr)
30	-3000	30	-2000
100	4000	100	5000

4.2.1 Supply Air Temperature Impact on Energy Saving Ratio

When primary air fraction is 30%, and room is fully occupied, the energy saving ratio is calculated. Table 4-2 and Figure 4-4 display the data of saving percentage between base and improved case for different occupied conditions. Part of the calculation results of the minimum airflow ratios, which were used to the calculate energy saving ratio for base and optimized case using the hourly weather data, is displayed in Appendix.

Table 4-2 Comparison data of saving percentage data between base and improved case for different occupied conditions

	SAT=120F	SAT=100F
Cooling energy	8%	14%
Reheating energy	13%	17%
Fan power	56%	62%

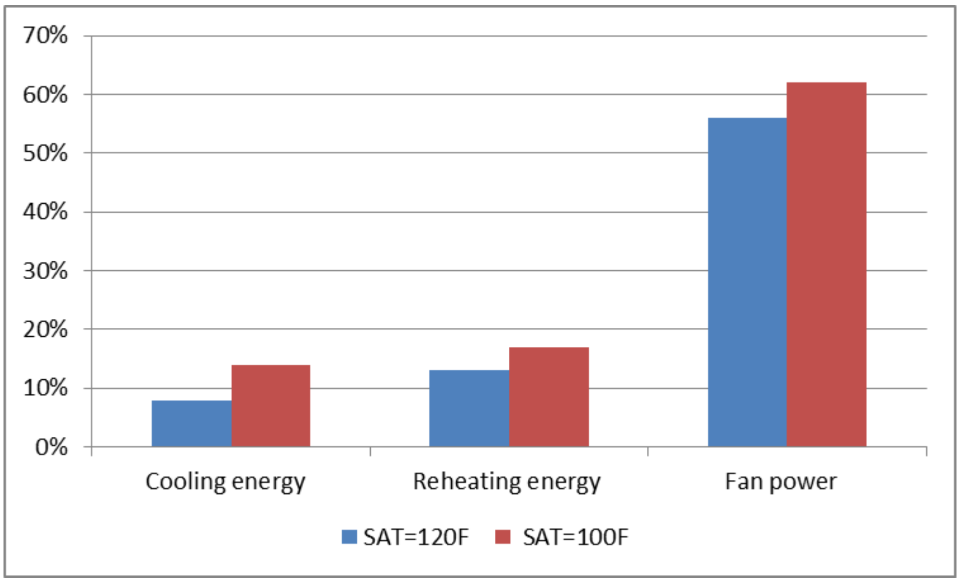


Figure 4-4 Saving percentage data between base and improved case for different occupied conditions

The numerical analysis was completed to determine the exterior zone minimum airflow requirement and estimate the overall potential energy savings ratio. According to this analysis, the calculated minimum airflow ratio was 14 to 48% of the design airflow.

Airflow varies a lot and can be lower than 30% of the design airflow. 30% percent minimum airflow may satisfy the fresh air requirement or load requirement after air is well mixed, but before that point comfort and ventilation be will sacrificed. Decrease the supply air temperature from 120F to 100F, and the cooling saving ratio increases from 8% to 14%, heating saving ratio from 13% to 17%. The fan can save up to half of its energy vary from 56% to 62% depending on supply air temperature.

4.2.2 Primary Air Fraction Impact On Energy Saving Ratio

As displayed in Figure 4-2, in most the temperature range, the optimized minimum airflow will not be affected by the outside air ratio a lot. The primary air fraction will larger than its minimum position, and it is determined by the fresh air requirement.

4.2.3 Zone Load Impact on Energy Saving Ratio

When supply air temperature is 120°F, primary air fraction is 30%, the energy saving ratio is calculated. Part of the calculation results of the minimum airflow ratios, which were used to the calculate energy saving ratio for base and optimized case using the hourly weather data, is displayed in Appendix. Table 4-3 displays the determined minimum airflow for different occupied conditions, and assumed conventional minimum airflow settings.

Table 4-3 The determined and assumed conventional minimum airflow for different occupied conditions

	Design airflow (CFM)	Minimum Airflow(CFM)		Minimum Airflow Ratio (%)	
		Base	Optimized	Base	Optimized
Fully Occupied	250	90	33~90	36%	22~36%
Unoccupied	250	90	25~90	36%	13~36%

Table 4-4 and Figure 4-5 display the data of saving percentage between base and improved case for different occupied conditions.

Table 4-4 Comparison data of saving percentage data between base and improved case for different occupied conditions

	Fully occupied condition	Unoccupied condition
Cooling energy	8%	24%
Reheating energy	13%	29%
Fan power	56%	66%

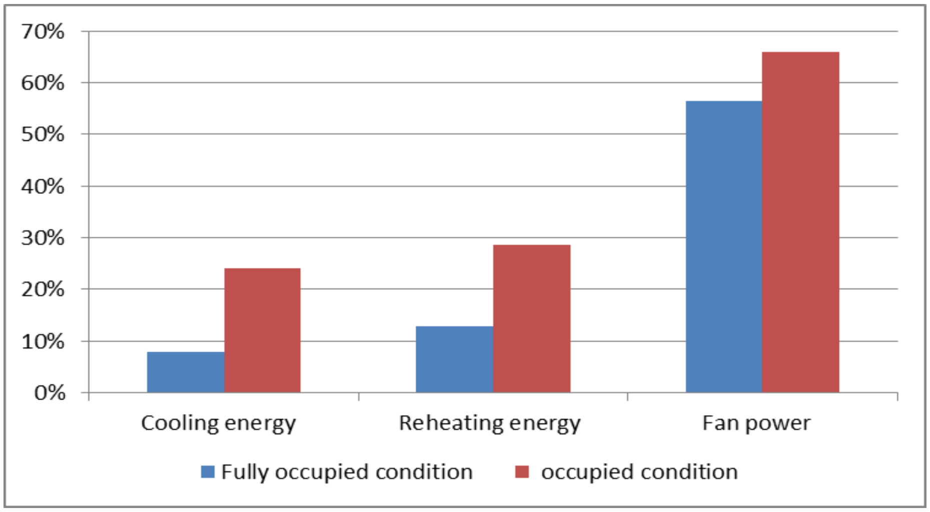


Figure 4-5 Saving percentage data between base and improved case for different occupied conditions

The numerical analysis was completed to determine the exterior zone minimum airflow requirement and estimate the overall potential energy savings ratio. According to this analysis, the calculated minimum airflow ratio was 13 to 36% of the design airflow. Airflow can be lower than 30% of the design airflow. The simultaneously cooling and heating saved was about 8% to 24% depending on the zone occupied condition. The fan power energy savings of the optimal minimum airflow can be much higher to more than 50% saving. During unoccupied period, saving ratios is usually higher than zone is fully occupied.

5. CONCLUSIONS AND CONTINUING WORK

5.1 Optimization of the Terminal Box Minimum Airflow Setpoint

Determining the optimal terminal box airflow is a complex process which is influenced by various factors, such as weather condition, supply air temperature, and internal load. A guideline for determination of a cost efficient minimum airflow setpoint for single duct VAV units is drawn in this research.

The most efficient optimal minimum airflow setpoint should not be a fix setting, but should be changing with zone load and ventilation requirement. When outside air temperature is around the temperature that the building has a neutral load, for the exterior zone, ventilation requirement determines the minimum airflow. As either zone heating/cooling load go beyond certain outside air temperature, the airflow required to maintain the temperature setpoint becomes higher than required by ventilation.

5.2 Influence of Different Operating Conditions

- Supply air temperature will determine the supply airflow minimum when heating load requires more airflow than ventilation requirement. Single fixed minimum airflow setting causes simultaneously heating and cooling, as well as fan power energy. Optimized minimum airflow can significantly reduce energy usage.

- Primary outside air fraction is usually not at its minimum position in mild weather condition. Higher primary air fraction will increase AHU energy cost. If primary air fraction is set too low, there will be potential ventilation issue.
- During unoccupied occupied conditions, the ventilation requirement serves as a more critical factor; during fully conditions, minimum airflow is determined more by the zone load requirement. Exterior zone load varies as outside air temperature changes. Occupancy also affects the zone interior load. During unoccupied period, it would be beneficial to set back the minimum airflow.

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APPENDIX

Sample calculation of energy saving ratio:

Table A-1 Calculation results of the minimum airflow ratios for cooling energy saving ratio

Date	Time	OAT (F)	Cooling			
			Fully occupied		Unoccupied	
			a_f	a_min	a_f	a_min
1/1/2009	00:55	28	36%	22%	36%	30%
1/1/2009	01:55	27	36%	22%	36%	30%
1/1/2009	02:55	27	36%	22%	36%	30%
1/1/2009	03:55	27	36%	22%	36%	30%
1/1/2009	04:55	27	36%	22%	36%	30%
1/1/2009	05:55	26	36%	22%	36%	30%
1/1/2009	06:55	25	36%	23%	36%	31%
1/1/2009	07:55	26	36%	22%	36%	30%
1/1/2009	08:55	27	36%	22%	36%	30%
1/1/2009	09:55	29	36%	21%	36%	29%
1/1/2009	10:55	31	36%	21%	36%	29%
1/1/2009	11:55	33	36%	20%	36%	28%
1/1/2009	12:55	34	36%	20%	36%	28%
1/1/2009	13:55	35	36%	20%	36%	28%
1/1/2009	14:55	35	36%	20%	36%	28%
1/1/2009	15:55	35	36%	20%	36%	28%
1/1/2009	16:55	33	36%	20%	36%	28%
1/1/2009	17:55	29	36%	21%	36%	29%
1/1/2009	18:55	26	36%	22%	36%	30%
1/1/2009	19:55	25	36%	23%	36%	31%
1/1/2009	20:55	25	36%	23%	36%	31%
1/1/2009	21:55	26	36%	22%	36%	30%
1/1/2009	22:55	30	36%	21%	36%	29%
1/1/2009	23:55	30	36%	21%	36%	29%
1/2/2009	00:55	30	36%	21%	36%	29%
1/2/2009	01:55	30	36%	21%	36%	29%
1/2/2009	02:55	30	36%	21%	36%	29%

Table A-2 Calculation results of the minimum airflow ratios for heating energy saving ratio

Date	Time	OAT	heating			
			Fully occupied		Unoccupied	
			a_f	a_min	a_f	a_min
1/1/2009	00:55	28	36%	22%	36%	30%
1/1/2009	01:55	27	36%	22%	36%	30%
1/1/2009	02:55	27	36%	22%	36%	30%
1/1/2009	03:55	27	36%	22%	36%	30%
1/1/2009	04:55	27	36%	22%	36%	30%
1/1/2009	05:55	26	36%	22%	36%	30%
1/1/2009	06:55	25	36%	23%	36%	31%
1/1/2009	07:55	26	36%	22%	36%	30%
1/1/2009	08:55	27	36%	22%	36%	30%
1/1/2009	09:55	29	36%	21%	36%	29%
1/1/2009	10:55	31	36%	21%	36%	29%
1/1/2009	11:55	33	36%	20%	36%	28%
1/1/2009	12:55	34	36%	20%	36%	28%
1/1/2009	13:55	35	36%	20%	36%	28%
1/1/2009	14:55	35	36%	20%	36%	28%
1/1/2009	15:55	35	36%	20%	36%	28%
1/1/2009	16:55	33	36%	20%	36%	28%
1/1/2009	17:55	29	36%	21%	36%	29%
1/1/2009	18:55	26	36%	22%	36%	30%
1/1/2009	19:55	25	36%	23%	36%	31%
1/1/2009	20:55	25	36%	23%	36%	31%
1/1/2009	21:55	26	36%	22%	36%	30%
1/1/2009	22:55	30	36%	21%	36%	29%
1/1/2009	23:55	30	36%	21%	36%	29%
1/2/2009	00:55	30	36%	21%	36%	29%
1/2/2009	01:55	30	36%	21%	36%	29%
1/2/2009	02:55	30	36%	21%	36%	29%
1/2/2009	03:55	31	36%	21%	36%	29%
1/2/2009	04:55	31	36%	21%	36%	29%
1/2/2009	05:55	32	36%	21%	36%	29%
1/2/2009	06:55	32	36%	21%	36%	29%
1/2/2009	07:55	33	36%	20%	36%	28%
1/2/2009	08:55	36	36%	19%	36%	27%

Table A-3 Calculation results of the minimum airflow ratios for fan power energy saving ratio

Date	Time	OAT (F)	Fan Power			
			Fully occupied		Unoccupied	
			a_f	a_min	a_f	a_min
1/1/2009	00:55	28	36%	22%	36%	30%
1/1/2009	01:55	27	36%	22%	36%	30%
1/1/2009	02:55	27	36%	22%	36%	30%
1/1/2009	03:55	27	36%	22%	36%	30%
1/1/2009	04:55	27	36%	22%	36%	30%
1/1/2009	05:55	26	36%	22%	36%	30%
1/1/2009	06:55	25	36%	23%	36%	31%
1/1/2009	07:55	26	36%	22%	36%	30%
1/1/2009	08:55	27	36%	22%	36%	30%
1/1/2009	09:55	29	36%	21%	36%	29%
1/1/2009	10:55	31	36%	21%	36%	29%
1/1/2009	11:55	33	36%	20%	36%	28%
1/1/2009	12:55	34	36%	20%	36%	28%
1/1/2009	13:55	35	36%	20%	36%	28%
1/1/2009	14:55	35	36%	20%	36%	28%
1/1/2009	15:55	35	36%	20%	36%	28%
1/1/2009	16:55	33	36%	20%	36%	28%
1/1/2009	17:55	29	36%	21%	36%	29%
1/1/2009	18:55	26	36%	22%	36%	30%
1/1/2009	19:55	25	36%	23%	36%	31%
1/1/2009	20:55	25	36%	23%	36%	31%
1/1/2009	21:55	26	36%	22%	36%	30%
1/1/2009	22:55	30	36%	21%	36%	29%
1/1/2009	23:55	30	36%	21%	36%	29%
1/2/2009	00:55	30	36%	21%	36%	29%
1/2/2009	01:55	30	36%	21%	36%	29%
1/2/2009	02:55	30	36%	21%	36%	29%
1/2/2009	03:55	31	36%	21%	36%	29%
1/2/2009	04:55	31	36%	21%	36%	29%
1/2/2009	05:55	32	36%	21%	36%	29%
1/2/2009	06:55	32	36%	21%	36%	29%
1/2/2009	07:55	33	36%	20%	36%	28%
1/2/2009	08:55	36	36%	19%	36%	27%

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