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Impact Of Supply Air Flow Rate On System Performance And Space Comfort Of Residential Air Conditioning Systems

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Impact of Supply Air Flow Rate on System Performance and Space Comfort of Residential Air Conditioning Systems

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Biographical Sketch

Fouad Hasan Z. Al Rifaie was born in Baghdad, Iraq. In 1967, he earned the Bachelor Degree in Mechanical Engineering from Baghdad University Iraq. He earned a Master Degree in Mechanical Engineering from The Technical University of Denmark at 1981. He worked in that time as supervisor representative of Iraqi government for Iraqi ships building in Denmark. Also he worked in many higher engineering positions in Baghdad such as commander in chief for many mechanical & construction of public and privet sector companies, teaching in many Iraqi governmental Engineering Institutes and position a Dean of Iraqi Naval Academy. During 2000-2006, he established a private Company named Ibn-Khaldun Company doing engineering consulting & contracting. Many projects had been performed such as health care center, schools, power plants, water supply, sewerage and many engineering projects and private houses. In August 2011, he started his Master program at North Carolina Agricultural and Technical State University.

Dedication

To Department Chair, Supervisor, and All Professors of Civil, Architectural and Environmental Engineering Department……

To my parents…..

My wife Nihal Al Raees……..

My family……

And to all who support and believe in me…….

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List of Abbreviations

AC	Air Conditioning
AHU	Air-Handling Unit
ADS	Air Distribution System
Amp	Ampere
ASHP	Air Source Heat Pump
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
BA Cnet	Building Automation (Controller System)
BAS	Building Automation System
BEopt	Building Energy Optimization
Btu	British Thermal Units, a measuring of Energy or Heat Flow
CFM (cfm)	Cubic Feet per Minute
CO_2-DCV	Carbon Dioxide base Demand Control Ventilation
COP	Coefficient of Performance
DB	Dry Bulb Temperature
DCV	Demand Control Ventilation
DDC	Direct Digital Circuit
DDW	Detailed Data Wizard
DOAS	Dedicated Outdoor Air System
EER	Energy Efficiency Rating
GSHP	Ground Source Heat Pump
H	Enthalpy Btu
h	Enthalpy Btu / Lb

hr	Hour
HVAC	Heating, Ventilation, and Air Conditioning
IAQ	Indoor Air Quality
KW	Kilo Watt
Lab	Laboratory
MWh/yr	Mega Watt Hour per Year
SDW	Schematic Design Wizard
EdHP	Educational Heat Pump
VAV	Variable Air Volume
PID	Proportional Integral Derivative
SEER	Seasonal Energy Efficiency Rating
WB	Wet Bulb Temperature

Abstract

The amount of air flow rate supplied to the space by the residential air conditioning system can have significant effect on the whole system performance and on the thermal comfort level. This study investigates the impact of air flow rates when they are higher or lower than the current practices or manufacture's specified values on system performance and on space comfort. In addition, this work examines the performance operation data such as (1) the annual and hourly energy consumption, (2) the system operation hours, (3) the temperature of air supplied to the space, (4) humidity ratio of the space, (5) system efficiency, and (6) the amount of sensible / latent load that can be added / removed from the space. Experiments are conducted on the air conditioning heat pump located at the HVAC laboratory of the Civil, Architectural & Environmental Engineering (CAEE) department in order to identify experimentally the fan performance and define the fan performance curve and efficiency. Simulations are also done to investigate the whole system performance of air conditioning systems and associated thermal comfort with different USA locations based on ASHRAE climate zones. The model is developed by the energy simulation software eQuest, using the characteristics of the tested lab system for a 1600 ft^2 typical residential house conditioned by 3-ton residential air conditioning heat pump system. The simulation model is tested on various air flow rates, ranging from 900 cfm to 1400 cfm and considering 1200 cfm as baseline. The results show an increase in the fan energy and total annual energy consumption with the higher airflow rate supplied. A higher temperature of the airflow causes elevated humidity in space that can become an issue in terms of space comfort, especially in humid weather locations.

CHAPTER 1

Introduction

1.1 Energy Consumed for Residential and Commercial Buildings in United States

The energy consumed in the United States can be contributed mostly to the usage within buildings. Almost 40% of the total amount of energy consumed in the United States is nearly shared equally between residential and commercial buildings (Stephens et al, 2011). The space conditioning needs of the building can drastically affect the total energy usage. Central space conditioning is the most abundantly used space conditioning in the United States, with over 60% of existing buildings and 90% of new constructed buildings having them. This central space conditioning method is critical to the comfort levels in building, due to the fact that Americans spend approximately 90% of their life time inside buildings (Stephens et al, 2011).

Focusing on the residential buildings across the United States, the previous studies reported a poor performance of space conditioning in terms of energy use. Space conditioning consumes large amount of energy use causing a consumption concerns. Central space conditioning systems are at least 17% lower than its rated performance (Rodriguez et al, 1996). Performance of the space conditioning can be affected by the amount of the air supplied by the system.

Space conditioning system in residential buildings can be window air conditioner, split air conditioner, packaged air conditioner or central air conditioner system. These systems are mainly reliant on the proper airflow across the evaporator coil. However, problem with split system exists because of that the desired airflow rates are rarely inspected to see if they are consistent with what is needed for an efficient system (Parker et al, 1997).

The performance of conventional split system residential air conditioning (AC) depends on adequacy of the airflow across the evaporator coil. Sufficient air flow rate is necessary to achieve a proper balance between sensible and latent capacity. Typical target air flow rate are approximately between 350-450 cubic feet per minute per ton (cfm/ton) of cooling capacity. The Air Conditioning Contractors of America (ACCA) is the association of HVAC (heating, ventilation, air conditioning, and building performance) contractors. ACCA recommended selecting the cooling equipment by using the recommendation in (Manual S) based on its stated sensible and latent performance, designing ducts to accommodate the necessary air flow by using (Manual D) and adjusting air handler fan speed to match loads. However, a problem with this approach is that contractors seldom check in the field to determine if design flow rates corresponding with what are achieved (Rutkowski, 1996).

ACCA recommendations are important to improve the performance of conventional residential air conditioning system. Also these recommendations insure an adequate evaporator air flow to the rated values (often 400 cfm/ton), which can reduce the average residential cooling energy use by approximately 10%. This will require better duct installation, properly sizing duct system and return grills. Moreover, proper duct design reduces system static pressure, which in turn reduces the fan energy use by half and improves overall system Energy Efficiency Rating (EER) by 12%. Another reason to emphasis on designing low flow resistance duct design is the tendency in modern residential AC systems of adding air filter to the system. One of the studies reported that substitution of pleated "high efficiency" filters typically reduce system air flow by 5% (Parker et al, 1997). Over sizing of AC and Heat pump system capacity should be avoided to provide adequate dehumidification through long duty cycle run-times

1.2 Heating, Ventilation, and Air Conditioning (HVAC) System

The typical central air-conditioning system for residential and commercial buildings in the United States is an air-handling unit (AHU). An AHU consists of a fan, heating and cooling coils and filters connected to the air distributing system. The cooling coils along with the refrigerant lines are connected to the outdoor condenser/compressor unit to the system cycle on and off to meet thermostat demands for space conditioning (Stephens et al, 2011). The ductwork has a large amount of airflow going through its system thus leaks within this system can lead to increases in energy consumption, higher in energy use and air infiltration. The combination of the AHU and the ductwork is responsible for maintaining the thermal comfort of a building. Complications within the systems airflow and pressure drop along with occupant influences can affect how well a system performs compared to what is expected (Rodriguez et al, 1996).

Indeed (HVAC) system is a technology that provides thermal control in buildings; ASHRAE Standards define it as a system with four objectives that can control (1) temperature, (2) humidity, (3) air circulation, and (4) air quality. Properly designed and maintained HVAC system can provide comfortable indoor environment year round, which considers a key factor for a successful buildings performance (Al Raees, 2013).

Most of HVAC systems use variable speed drive (VFD) to control fan instead of methods such as inlet guide vane and outlet damper. The fan speed is controlled by using feedback control from the supply duct static pressure sensor located inside the duct. A fan energy saving can be realized if the duct static pressure set point is reset in such a way that at least one of the VAV boxes remains open (Nassif, 2010). Several standards exist for testing the energy performance of systems at standardized laboratory conditions (AHRI Standard 210/240), as well as actual field performance of duct systems (ASHRAE Standard 152, ASTM E1554).

HVAC System is responsible for maintaining thermal control in the building. Several studies have found that the actual field performance of HVAC systems is different from laboratory performance or design conditions, in terms of system capacity, airflow, and refrigerant change, which can have major implication for energy consumption.

However, many of the individual parameters combine to affect energy and indoor air quality (IAQ) in complex ways. For example, various airflow rates through the AHU affect system operation time (duty cycle), system fan power, cooling capacity, temperature and humidity differences within s and AHUs. Conversely, airflow rates and plenum operating pressures are directly related, and airflow rates are affected by pressure drops across filters, heating and cooling coils in the AHU. Pressures operation also affects duct leakage rates, which influence both energy and IAQ. Finally, occupant thermostat settings affect many parameters too such as fractional operation times, recirculation rates, cooling capacity, and temperature and relative humidity. Figure 1 shows a schematic of typical central air-conditioning system for residential or light-commercial building (Stephens et al, 2011).

Although some of the above parameters have been described in the literature, there are still gaps in researchers knowledge of how interactions of many of these system operational parameters affecting energy use and IAQ in existing buildings. The current state of knowledge of the parameters were explored by (Stephens et al, 2011), in the context of four main system components.

1. AHUs,
2. Outdoor units
3. Ducts
4. Occupant influences and overall performance.

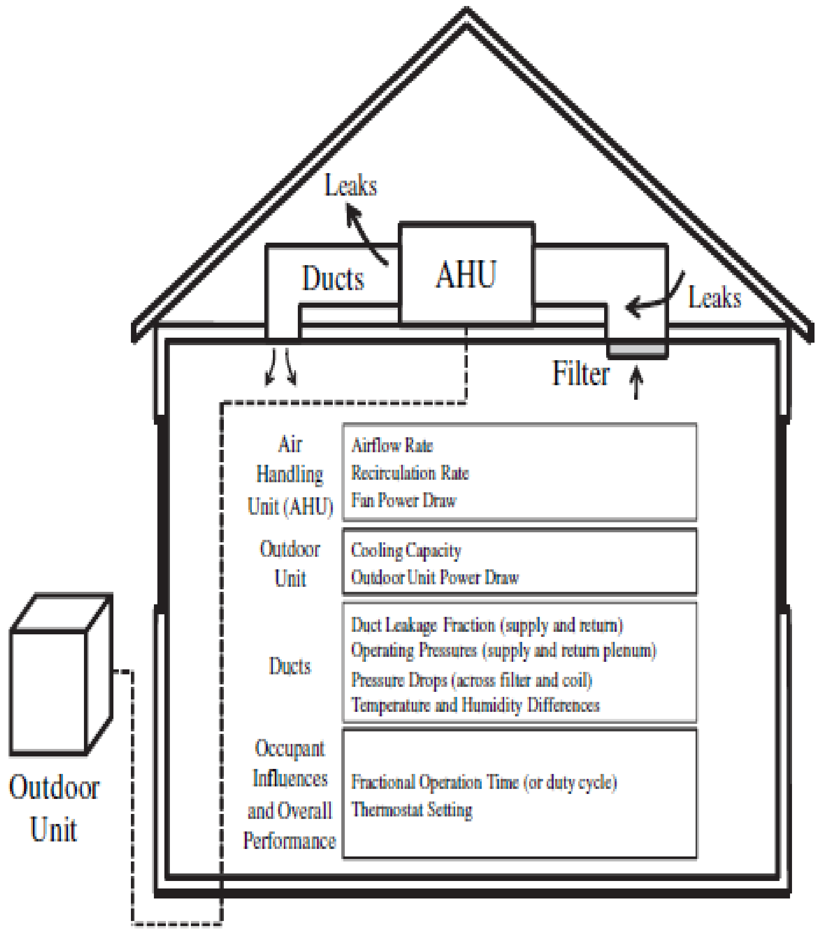


Figure 1. Residential/light-commercial central air-conditioning system (Stephens et al, 2011)

1.3 Significance of the Study

1.3.1 Problem Statement

Air flow rate (cfm) in the HVAC system may be supplied less or more than the required rate due to many problems. These problems can be classified as design phase, construction phase or even by the users such as:

- Improper duct design such as under sized of the return ducts or the grills. Also the long distribution system can cause a high static pressure in the ducts which affects the air flow:
- Improper duct design of the fan of the system or choosing the wrong fan speed mode.

- Dirty filters or fouled filter.
- Close one of the registers by users.
- Inhibited old age system.

Any of the above mentioned actions can cause change in the flow rate of the air which in turn, degradation in system performance and space thermal comfort besides the energy consumption (Parker et al, 1997). This research is focusing on how various air flow rate supplied by the HVAC system can affect system performance and the space thermal comfort.

A laboratory air conditioning unit is selected for this study. The unit represents a three ton heat pump split system. The system has been designed to control and to evaluate the air movement and energy transfer occurring in all parts of the system and through the whole process of air conditioning.

Heat transfer processes are occurred due to the effect of refrigerant cycle between the compressor, condenser and evaporator. The heat transfer process is responsible for adjusting the outdoor air to the desirable condition (temperature and humidity) inside the building. While the fan and the distribution system is responsible for supplying and recalculating the modified air between the space and system.

A lower or higher air flow may be expected to cause several problems in the real application of HVAC system in general and heat pumps in specific. The main goal of this study is to investigate the performance of HVAC system under different air flow rates. To achieve this goal several objectives are considered:

- Effect of air flow on daily and annual energy use.
- Effect of pressure across the fan on the fan energy use.
- Daily system operation hours.

- Temperature of air leaving cooling coil.
- Relative humidity (Humidity Ratio) of the space that affect occupant thermal comfort.
- Determine the sensible and latent load that removed by system for various air flow rate.
- Investigate the impact of extreme weathers on the specified air flow rate (cfm) per ton of cooling.

Four locations with different climate zones are selected for this objective. To achieve the above objectives, a building model is developed using the whole building energy simulation software. The developed building model reflects a house can be served with the tested heat pump system. The model is run with various air flow rate to analyze the annual/hourly energy usage and the other parameters discussed above.

1.3.2 Thesis Statement

Air flow rate in residential air conditioning system can be less or more than that specified by manufactures. This in turn could affect the performance of the residential air conditioning system and thus the thermal comfort. This study involves lab testing and simulations. The aim of the experiments is to measure and develop fan performance curve for 3-ton residential heat pump system located in HAVC Lab. The simulations carried for typical residential home to investigate performance of its HVAC system. Different locations were selected to identify the effect of climate on the performance of the air conditioning system when air flow rate are differed than the manufacturers' specified.

There is a lack of data available related to the effect of maintenance on the performance of residential air conditioning heat pump system. This is especially true when reduced evaporator airflow is considered. More studies thus are needed to evaluate the widespread effect of these problems on the overall performance of the air conditioner and on the energy usage and peak demand as well as penalties created by these effects (Parker et al, 1997).

The purpose of this research is to experimentally investigate the performance of the residential air conditioning heat pump systems to quantify the effects of various air flow rate (cfm) on the performance of the system and on the energy use.

The study also demonstrates the effect of supply air flow rate on the pressure difference across the fan. Variables considered in this experimental research are:

1. Supply air flow rate (cfm).
2. Pressure difference across the fan (in W.g).
3. Different location of ASHRE climate zone represented in different cities of US.

The variables used to describe the performance of the air conditioners are air quantity (cfm), pressure required in W.g and power kW. In reality, not only these variables influence the performance air conditioning, but there are several other factors that may affect the performance such as improper unit installation, wrong selecting of the fan speed, and lack of maintenance and age of the system. A model simulation by eQuest software is created to investigate the parameters that affect the residential air conditioning heat pump system. The 1600 ft^2 house is first modeled by the Building Energy Optimization (Beopt) software and then exported to the whole building simulation software eQuest for further performance analysis.

1.3.3 Objectives

To conduct experiment and simulation investigation to determine the effect of various supply air flow rate (cfm) of the residential air conditioning heat pump (HVAC) systems on:

- System performance
- Energy Usage

- Space thermal comfort

The objective is done by analyzing:

1. Effect of pressure across the fan on fan energy use.
2. Daily system operation hours.
3. Temperature of air supplied to the space.
4. Humidity ratio of the space.
5. The amount of sensible/latent loads removed from the space.

1.3.4 Methodology

The following methodology is used:

- Experimental studies are conducted to construct the performance fan curve of the system. The relationships of the pressure across the fan (in W.g) versus the supplied air flow rate (cfm) are developed.
- A model of the house in Beopt is created and exported to eQuest software to perform the simulation by eQuest with various supplied airflow rates and different pressures across the fan. The simulations are to the annual and hourly cooling energy use. An experiment is performed to develop the fitting fan performance curve and it is used in the simulations.
- Investigate the impact of the airflow rate (cfm) supplied by analyzing the simulation's output of the model that is used in tested the air flows with different pressures across the fan. The location of building model was changed in the eQuest software to test the effects of different weathers regarding the different locations of ASHRE climate zone.
- Experiments were done in four different locations with different climate zones weathers. Output results of each parameter were analyzed and discussed separately in details.

1.3.5 Thesis Organization

The thesis is divided into five chapters to investigate the various effects of the supplied design air flow rates for the typical residential air conditioning heat pump. This work investigates and discusses the performance of a residential air conditioning system with a constant air volume and constant fan speed when operated with different design air flow rates. The discussions are in terms of the parameters which mainly affect system performance, energy consumed and space thermal comfort.

Chapter 1 introduces the objective of this research and the significant of the study. Chapter 2 provides a literature review of the works done on this topic. In chapter 3, the methodology will be discussed, including model simulating by BEopt and eQuest programming which is created to investigate and analyze all parameters affecting the performance of the residential air conditioning heat pump system. Chapter 4 discusses the results of the calculations obtained for different parameters and investigates their effect on the performance of the system. Finally, in the Chapter 5, the discussion and conclusions are provided, including recommendation for further work.

CHAPTER 2

Literature Review

Adequate air flow across the evaporator coil to achieve a balance between sensible heat transfer and moisture removal are so significant due to the strong relating with the performance of a conventional split system residential air conditioner (AC). In the United States, people spend 80-90 percent of their life time inside buildings thermal comfort and indoor air quality (IAQ) for the occupants inside buildings has big influence on the health and productivity of human (Howell et al, 2005). A significant impact on the quality of life is due to the air-conditioning of indoor environment. Several authors reported that the performance of a conventional split system residential air conditioner is dependent on the amount airflow rate (cfm) supplied to achieve the human thermal comfort.

Amount of coil air flow supplied must be adjusted to approach the design flow rates correspond with what is achieved. Too high, air moisture removal is compromised and fan power may be increased as well as if flow is too low, sensible cooling is reduced with degradation of cooling system energy efficiency ratio (EER). Very low air flow may lead to evaporator coil icing, refrigerant flood back and eventual compressor failure. A typical airflow rates supplied for residential split systems are often 350 - 450 cubic feet per minute (cfm) per ton of cooling capacity as manufacturer recommended. Supplied airflow might be of 425 - 450 cfm per ton through a dry coil usually will be needed to achieve 400 cfm/ton (664 L/S per KW) when the AC is operating with a wet coil (Parker et al, 1997).

2.1 Energy Consumption for Space Conditioning

The consumption of energy for space conditioning is mentioned to be 20% of the total energy use in the residential sector and the average performance of residential air-conditioners is at least 17% below rated performance (Rodriguez et al, 1996). Another report mentioned that for an overview of almost 9000 residential air-conditioners and over 4000 light-commercial air-conditioners in California, the researcher mentioned that the majority of residential and commercial systems had rated capacities of 8.8 to10.6 KW and 15.8 to 17.6 KW, respectively The manufacturer recommended to use as more than 5% from correct charge of refrigerant for the reason that over half of the systems had either too much or too little refrigerant charge (Stephens et al, 2011).

Relatively a large fraction of total building energy consumption, are due to the HVAC systems and associated equipment, and high percent portion of which is due to fan operation. Improper operation and installation of economizer dampers can cause high energy consumption in fans. The potential high pressure drops through those dampers and associated high total pressures that should be developed by supply and/or return fans are the mainly affected reason. So a proper strategy to operate optimally the economizer dampers should be conducted with minimum fan energy use (Nassif, 2010).

2.1.1 Heat Energy Transfer

Energy is one of the most basic requirements and a fundamental element for all economic systems. The subsequent increase in energy prices due to the global shortage in low cost energy generation resources has initiated energy conservation studies. Air conditioning laboratory unit has been designed to demonstrate and to evaluate the energy transfer occurring in all the process which required. A heat energy transfer process from hot surface being cooled as (warm air return) to the cooling substance as (cold refrigerant) flowing through the evaporator coil tubes which this process defined as evaporation cycle.

Heat energy transfer processes are due to the effect of refrigerant cycle which occurs during the difference between the entering and leaving conditions in the evaporator coil, hot surface being cooled as (warm air return) to the cooling substance as (cold refrigerant) flowing through the evaporator coil tubes which this process defined as evaporation cycle. Regarding to the Fourier equation in heat transfer, rate of heat flow in the evaporator coil is:

$$Q = U * A * \Delta T \qquad (2.1)$$

Where:

Q = is the heat flow in evaporator coil in Btu/hr

U = Over all heat transfer coefficient Btu /hr ft^2 ^{o}F (including coil wall resistance, air side film and refrigerant side film)

A = Total surface area of heat transfer throw it in ft^2

Δ T = Temperature difference between the air flow across the evaporator in ^{o}F

This equation is determining the controlling of Q to be regulated Δ T, supposed the condition to be a steady state condition.

The effect of refrigerant is the difference between the entering and leaving conditions in the evaporator coil. The value of enthalpy h (Btu/lb) should be included any additional heat observed by the refrigerant as it reaches 100% evaporation and the refrigerant (R-22) would has the evaporator as a super-heated vapor and the measuring unit of the refrigerant rate is Ton where:

1 Ton of cooling = 12,000 Btu / hr which calculated (Siegel et al, 2000), by melting on ton of ice during 24 hour and the determination of this value is as:

1 Ton = 2000 lb / day x 144 Btu / lb = 288, 000 Btu / day

= 288,000 Btu / day / (24hr/day) = 12,000 Btu / hr

Where: 144 Btu / lb are the latent heat of fusion for ice.

2.1.2 Heat Pumps

A definition of Heat Pumps that heat energy flows naturally from high to low temperature. Heat pump is a mechanical device that takes heat at a lower temperature and pumps it to a higher temperature, such as the refrigerators is a cooling cycle which it contains a heat pump that takes heat from its interior and heats a coil at the back of the refrigerator. The back of the refrigerator is much warmer than the interior of the home, and the interior is cooler, so the cycle produce that, the energy cost of doing, is the electricity to run a compressor. In air conditioner a similar operation is happen that the heat pump takes the heat from a coil in ductwork, which cools the air that flows around it, which is cooling the home and pumps it to an outdoor unit, which is hot, causing the heat rejected to the outdoors.

Heat pump refrigeration system consists of compressor, evaporator, expansion valve, and refrigerant in two cooper coils for the indoor and outdoor ambient, it transfers heat in and out. Figure 2 shows a diagram of Standard Air Conditioner Air-to-Air heat pump system for heating and cooling cycle (Angel et al, 2009). When a building is in heating mode, the liquid refrigerant on the outside coils extracts heat from the air that evaporates into gas; whereas, the indoor coils release heat from the refrigerant that is condensed back into a liquid. During cooling, the expansion valve can change the direction of the refrigerant which called a refrigerant cycle (Al Raees, 2013).

In a refrigerant cycle, heat moves from one ambient with low temperature to another ambient with high temperature to heat or cool the building/space. An outdoor ambient can be air, water or the ground, while the indoor ambient is the air inside the building. A heat pump can be air source, water source or geothermal according to its associated ambient. This study considers a standard air conditioner An Air-to-Air heat pump unit.

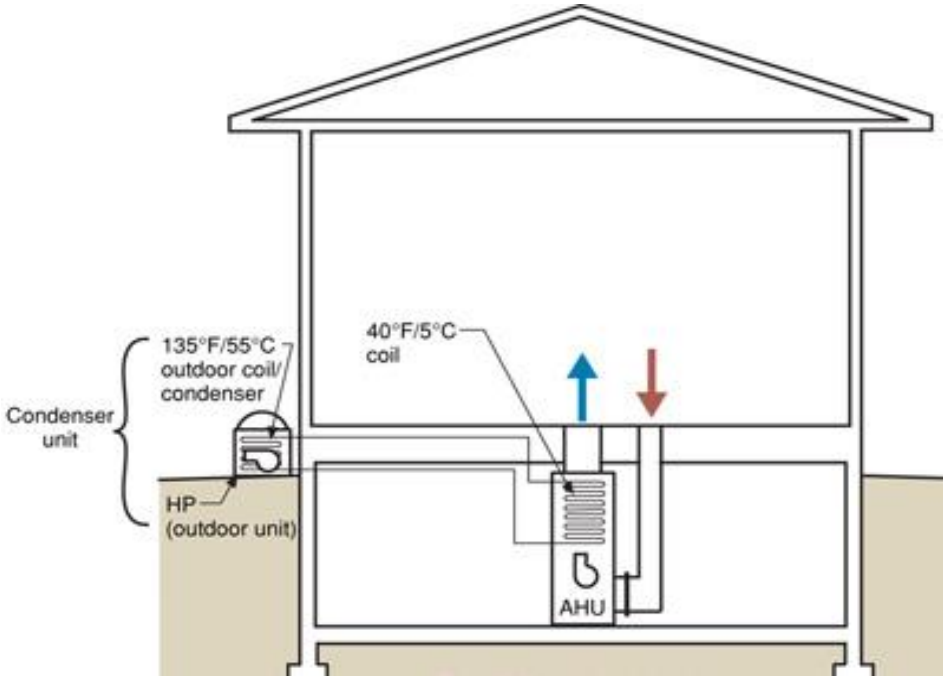


Figure 2. Standard Air Conditioner Air-to-Air Heat Pumps (Rodriguez et al, 1996).

Same fundamental concepts of the heat pump are for refrigerator heat pump cycle which has the same components except that there is no fan to plow air. Cooling operation occurs through refrigerant inside cooper coils in the indoor. A heat exchanging between the cold cooper coils and the indoor medium and just the cooling of indoor medium happen during the conduction and convection of the heat flow rate (Rodriguez, 1995).

2.1.3 Ground Source Heat Pumps

A Ground Source Heat Pump (GSHP) is another form of space conditioning for a building. It is one many heat pump systems available. This is believed to be a highly efficient method of conditioning spaces within the building. There is a doubt however in the field performance in comparison what was initially expected. Climates with moderate ground temperatures and buildings with approximately equal annual heating and cooling loads can cause unexpected results when using this type of space conditioning (Stephens et al, 2011).

Either collection heat from the ground and pumps it to a coil inside the ductwork to provide air heating, or collection heat from the same coil in the ductwork (thereby cooling the air) and rejects it to the ground, this process known as ground source heat pump.
Another phase of ground source heat pump, that in some systems uses a loop of tubing in a radiant floor or ceiling instead the heat collected - rejected to a coil in ductwork. Figure 3 shows a schematic cycle of a ground source heat pump (Straube, 2009).

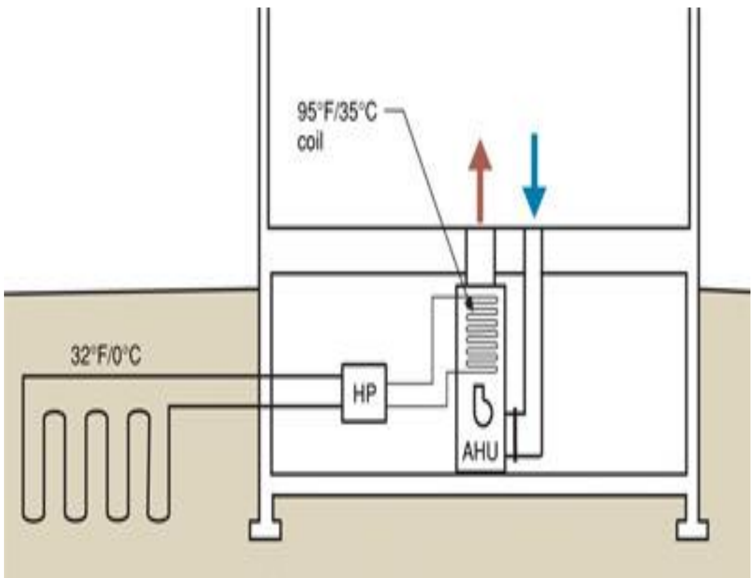


Figure 3. A schematic of a ground source heat pump with heating mode (Building Science)

2.2 Sequence of Operation

The building occupant are sequenced the operation by them self manually. The heating or cooling mode selection are manually entered in Thermostat set point at the "auto" setting, heating or cooling system operation cycles on-off as the thermostat set point is satisfied. The disable heating / cooling operation by selecting the "off" control position is doing by the occupant also. The sequence of operation is a, Standard Air Conditioner-An air-to-air heat pump that pumps heat from the interior to the exterior.

The sequence operation flows in air conditioner is a heat pump that takes heat from a coil in ductwork (which cools the air that passes over it, thereby cooling the home) and pumps it to an outdoor unit (which is hot, so that heat is released to the outdoors). These are familiar and well understood technologies (Straube, 2009).

2.2.1 Heat Pump Cycle

A type of HVAC system that uses refrigerant cycle is Heat pump system. Heat moves from one ambient with low temperature to another ambient with high temperature to heat or cool the building/space this called a refrigerant cycle. An outdoor ambient can be air, water or the ground, while the indoor ambient is the air inside the building. A heat pump can be air source, water source or geothermal according to its associated ambient. This study considers the first type air source heat pump.

The components of the Heat pump refrigeration system are the compressor, evaporator, expansion valve, and refrigerant in two cooper coils for the indoor and outdoor ambient, it transfers heat in and out. Figure 4 shows a diagram of basic heat pump cycles for heating and cooling cycle. When a building is in heating mode, the liquid refrigerant on the outside coils extracts heat from the air that evaporates into gas; whereas, the indoor coils release heat from the refrigerant that is condensed back into a liquid. During cooling, the expansion valve can change the direction of the refrigerant (Al Raees, 2013).

The performance of an air-conditioning system is airflow and recirculation rates are a part dependent on the airflow rate through the system. An important parameter in IAQ models is the recirculation rate (the HVAC volumetric airflow rate divided by the volume of space that a system serves), those that assess pollutant removal technologies, because the product of in-duct air cleaner efficiency and recirculation rate can be directly compared to other loss mechanisms including air exchange and deposition loss. Recirculation rates are a function of system airflow rates, house volume, and fractional operation times (i.e., duty cycles).

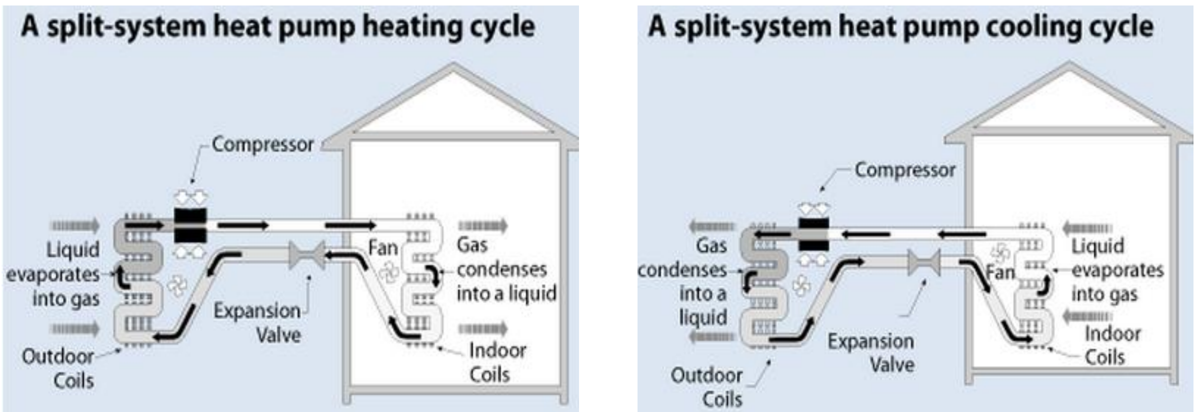


Figure 4. Basic heat pump cycles for heating and cooling methods (Al Raees, 2013)

HVAC designers design the condition of air provided by HVAC system to the conditioned space depend on ASHRAE 62.1-2010 ventilation for acceptable indoor air quality and ASHRAE 55 for thermal comfort concerned the considerations that:

- Property of mixed air; the portion of fresh air from outdoor dampers to return air from return duct should meet specifications; through determining the amount of ventilation specified based on occupancy number and activity type in the conditioned space .
- Temperature, humidity and velocity of supply air is determined according to thermal comfort specification ASHRAE 55 with energy cost considerations (Al Raees, 2013)

2.2.2 Fan Power

In the air handling unite (AHU), the power draws often exceed standard assumptions referring to some studies which have shown that AHU power draws often exceed standard assumptions for air-conditioner rating test procedures and that residential AHU fans regularly consume more energy annually than a typical refrigerator. HVAC system and associated equipment consume a relatively large fraction of total building energy consumption, a significant portion of this energy consumption is belong to fan operation, when increasing concern about the total energy consumption in building (Nassif, 2010).

The air flow produced by an air handler against the duct system's frictional air flow resistance is governed by the indoor unit's fan performance characteristics. A system pressure versus flow relationship in which air flow increases as the external air flow resistance is reduced are described by the blower fan curve as shown in Figure 5 (Parker et al, 1997). "Characterized by a resistance versus flow relationship are by duct system's performance; in general when a more air is forced through the duct, the duct resistance in the system increases rapidly." If test and balance data on duct air flow and external static pressure is available for a single point, the entire duct resistance curve is readily derived as in equation (2.2), (Rodriguez, 1995).

$$R_n = P_1 \left(\frac{cfmn}{cfm1}\right)^2 \qquad (2.2)$$

Where:

R_n = Duct resistance inches of water column (IWC or Pa) at flow "n"

P_1 = External static pressure at test point (IWC)

cfm_1 = cfm flow at test point

cfm_n = cfm at flow "n"

The plotted curves of Influence of fan performance and duct flow resistance on system operating point as shown in Figure 5. The operating point is important since it represents the only operating condition obtained when a single blower is mated to a given duct system, where the fan curve intersects with the duct system resistance curve. Often there are three or more speeds available to the blowers; there are a corresponding number of operating points. So fan provides the air flow rate required in order to cover the thermal and ventilation loads and adequate pressure to meet all pressure drops in air ducts and dampers. Under low load in a VAV system, when the fan air flow rate is decreased, the fan speed can be controlled by way of using a variable speed drive. To control the fan speed in this regards, the supply duct static pressure sensor is normally used (Nassif, 2007).

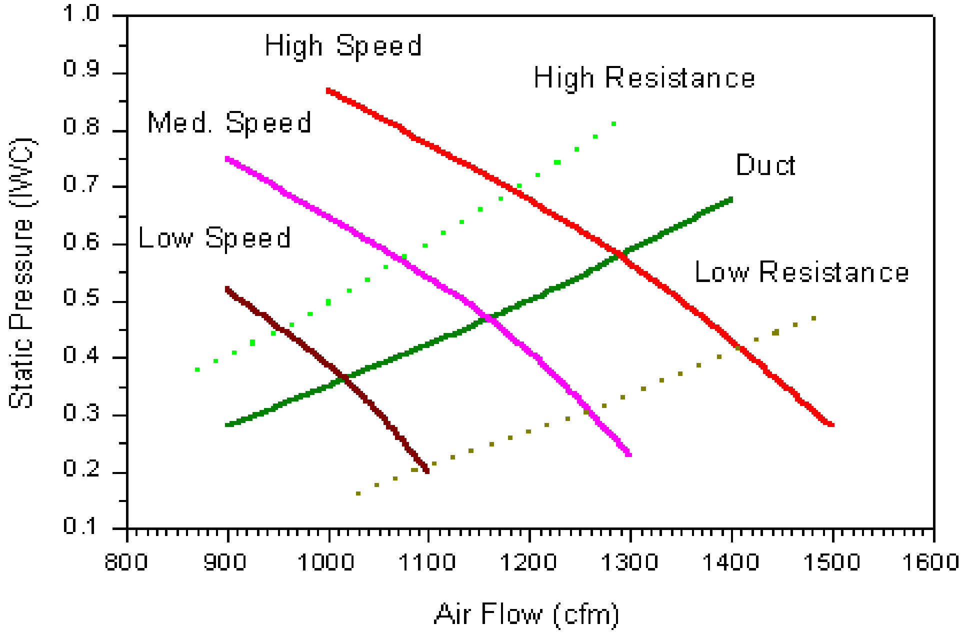


Figure 5. Fan performance and duct flow resistance on system operating (Parker et al, 1997).

Air side performance is also strongly influenced by the duct system flow resistance as illustrated. However, the resistance of the duct system may be affected by unintentional aspects of the duct system installation and operation. Improper installation with site compromises to duct system design constricted or pinched and collapsed ducts (ACCA. 1995, Manual J). The available fan static pressure may be concerned pressure drop such as coils, dampers, filters, grills, resistance heaters. On the other hand, homeowners can reduce air flow by adding high efficiency filters, or allowing filters and/or coils to become soiled. They can also close off supply registers in an attempt to zone spaces or control room temperature distribution.

2.3 Power and Air System Energy

A system with a strategy required slightly less fan power as concluded (Nassif and Moujaes, 2007), they mentioned depending on other author, that also they proposed a new strategy in which only the discharge and recirculation dampers were linked while the outdoor damper always remained full open. They found that the strategy prevents air from entering the system through the discharge damper. However, the effects of the proposed strategy on fan energy use were not investigated.

Referring to the first consideration in energy design effective systems due to the heat capacity equation, it becomes evident that if the designer can do anything to reduce the load, which are a power term (KBtuS/h of cooling), the system and machinery will be smaller, thus will cost less and use less energy when operated.

The fan power equations (Howell et al, 2005) are:

$P_{fan} = (Q * \Delta p_t) / (6350 * \eta_f)$... (2-3)

Where:

P_{fan} = fan power, hp

Q = air circulation rate, cfm

Δp_t = fan total pressure rise, inW.g.

η_f = fan efficiency, decimal

The equation shows that reducing the flow rate or the system pressure or increasing the fan efficiency will reduce the power. The flow rate can be reduced by:

(1) Reducing the load in Equation (2-3).

(2) By making a careful load analysis that does not include excessive or hidden uncertainty or "safety factors," or

(3) By increasing the temperature difference between the supply air and room temperature. The pressure loss requirement of the conditioner and the distribution system equals the fan pressure, term that can be controlled by the system designer. Friction losses in the distribution system can be expresses by the Darcy-Weisbach equation,

$\Delta h_f = C_f * (L/D) * (\rho V^2/2g_c)$.. (2-4)

Where:

Δh_f = friction in loss, in W.g.

f = friction factor, dimensionless

L = length of duct, ft

D = diameter of duct, ft

V = velocity of air in duct, ft/s

g_c = units conversion constant, lb_m. ft/s^2. lb_f

C_f = unit conversion factor, 0.1923, (inW.g ft^2)/lb_f

ρ = density of air, lb_m /ft^3

Then the methods are to reduce the fan pressure requirement (fluid head loss) are:

- Limit the length of duct runs to the minimum possible
- Increase the diameter (or equivalent diameter)
- Reduce the velocity
- Reduce the roughness of interior surface, which reduces the friction factor

Moreover, in most systems, the duct fittings create a significant amount of the pressure losses (Howell et al, 2005). In both the supply and return ductwork, should be designed for minimum pressure losses for all fittings, which are a function of the fitting construction and the velocity head to ensure continued operation with low pressure drops, any device such as a damper, coil, turning vanes, etc., which could result in a blockage to airflow, should be provided with an inspection and access port. An additional consideration regarding the pressure is the temperature rise across the fan. The temperature rise is expressed by the equation (Howell et al, 2005).

$$\Delta t_f = (0.371 * \Delta p_f) / \eta_f \quad \text{(2-5)}$$

Where:

Δt_f = temperature rise across fan, ºF

Δp_f = total pressure rise across fan, in W.g.

η_f = fan efficiency, decimal

If the fan efficiency supposed to be 74%, Equation (2-5) shows that, the temperature rise in ºF would equal one-half the pressure would rise in inW.g. For example, a 74% efficiency fan producing rise 4 in W.g pressure would raise the air temperature 2 ºF. In most systems, this temperature rise becomes part of the cooling load, thus requiring the use of yet more energy.

Fans should always be selected at the maximum efficiency point on their curves regarding to the fan efficiencies, and it is highly recommended that a designer always use a fan curve when selecting a fan so that the anticipated range of operation can be analyzed.

The fan energy equation is the power equation multiplied by the hours of operation and with the appropriate terms for motor efficiency and conversion of horsepower to kilowatts, that to divide 6350 by 0.746 which is the conversion from Hp to KW, to get the number 8512 as:

$$q_{fan} = (Q * \Delta p_t * \theta) / (8512 * \eta_f * \eta_m) \quad \text{(2-6)}$$

Where:

q_{fan} = fan energy, KWh.

Q = air circulation rate, cfm.

Δp_t = total fan pressure, in W.g.

θ = time of operation, hours.

η_f = fan efficiency.

η_m = motor efficiency.

The hours of operation and the motor efficiency are the only two variables that have been incorporated in this equation that were not included in the power equation (2.3). Accommodation should always be made to maintain an unoccupied building under controlled conditions in selecting and designing a system, while shutting down the major energy consuming devices or operating them at a low energy consumption idle mode. Always recommended for fan or pump drives, regarding to motor efficiencies, the use of high-efficiency motors should be selected.

(Howell et al, 2005), listed another form of the fan energy equation for a fixed or given system (i.e., the system curve is constant) is:

$p_{fan} = Q^3 / (6350 * \eta_f * C_s^2)$…….. (2-7)

$q_{fan} = (Q^3 * \theta) / (8512 * \eta_f * \eta_m * C_s^2)$... (2-8)

Where: C_s is the system constant in $ft^3/(\min(\text{in W.g.})^{0.5})$).

This means, if the air delivery rate can be reduced for a given system, the energy consumption is reduced as the cube of the flow rate (e.g. 20% flow reduction equates to 49% energy reduction, 50% flow reduction equates to 87.5% energy reduction, etc.) regarding to equation (2.7). This characteristic explains one of the major benefits of using a variable flow VAV system instead of a variable Δt system (Howell et al, 2005).

2.3.1 Energy Efficiency

Sources of energy can be electrical, heating, lighting, mechanical, chemical, solar, and even nuclear energy. In buildings can be found many fields such as in intelligent building management systems, lighting, heating, cooling, refrigerant cycle, and ventilating systems, all of them are related with energy efficiency (Al Raees, 2010).

Moreover, energy efficiency and energy conservation in buildings should not compromise a comfortable environment for occupants. The study explores methods of improving the efficiency of heating, ventilation, and air-conditioning (HVAC) systems, and the savings can be obtained while assuring IAQ and thermal comfort is reached by applying the ventilation control strategy and economizer. Another measurement is the efficiency of a heat pump itself is to report the amount of energy that is pumped relative to the amount that must be added to do the pumping. This ratio is called the Coefficient of Performance (COP), where COP = quantity of heat delivered / energy required by pump. COP is considered the best way to measure the efficiency of a heat pump itself. For example if a COP of about 3.5 for a typical efficient air conditioner this means it can remove heat at a rate of about 3.5 KW while consuming about 1 KW of electrical energy, (Building Science).

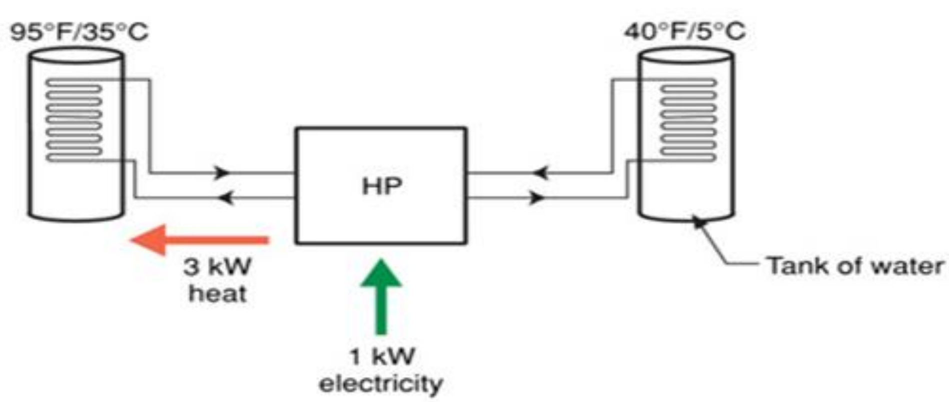


Figure 6. Heat pump coefficient of performance

Input for efficient heat pumps increases as the height that must be pumped decreases. For example, to cool a home a typical conditioner must “pump” heat from a cold “coil” (the finned, radiator-like device hidden in the ductwork that picks up heat from the air flowing over it) temperature of about 50 ^{0}F (10 ^{0}C) to a hot outdoor "condenser" temperature of about 135 ^{0}F (60 ^{0}C) a "height" of 85 ^{0}F (50 ^{0}C). If the same air conditioner only cooled the coil to 60 ^{0}F, the COP would be higher and the heat pump (air conditioner) would be more efficient. If the air outdoor was cool, say 70 ^{0}F, the outdoor "condenser" unit sitting next to the house would be more easily able to reject the heat to the outdoor air and so the outdoor coil might only need to be at 100 ^{0}F. In this scenario, the "height to pump" would be even less, at 50 ^{0}F, and the COP (efficiency) would be even higher".

2.3.2 Energy Efficiency Rating (EER)

It is the numerical ratio of the cooling provided (in Btu per hour) divided by the electricity required (in KW). Air conditioning systems are rated based on how much energy they need to provide cooling under specific standard test conditions (a set height to pump), the rating is obtained by (EER). Since this rating measure used Imperial units divided by metric units, it seems a strange measurement, but it is easy to convert from EER to COP by dividing by 3.412. For example an air conditioner with an EER=12 would have a COP of about 12/3.412 or 3.5. Because not all hours demanding air-conditioning are equally hot, the Seasonal Energy Efficiency Rating (SEER) was developed to consider times when cooling is required but the outdoor temperatures are not as high.

For the standard SEER test that the conditions are a rather unrealistic 80 ^{0}F indoors and 82 ^{0}F outdoors: hence the temperature "lift" is small and so advertised SEER rating can often be artificially high (models with SEER = 24 or a COP of 5.5) are now available.

The system COP of a heat pump in a cold climate set in heating mode are rated at COP = 4+ can easily drop to COP = 3. A system COP of 3 for a heat pump in heating mode would be considered good in cold climates (cold soil) by the experience, even with very efficient heat pump equipment and well-designed and installed pumps. COP values of as high as 4 to field heating mode are possible in warmer climates (warmer soil) and with the best design and best equipment. In cooling mode in mixed and cool climates, summer-time system COP values tend to be higher because the ground temperature in summer (perhaps 60 ^{0}F) are close to the desired air conditioning coil temperature (40 ^{0}F), whereas during winter, the heating coil temperature (at say 100 ^{0}F) is far from the winter ground temperature (say 35 ^{0}F). The electrical energy to run the pumps, fans and compressor of the whole system is useful heat in the winter (the inefficiency in the motors results in heating, which is the whole purpose) and increases the cooling load in summer, this means all of the inefficiency results in heat, which then has to be removed by the heat pump (Al Raees, 2013)

The following equations were used how to calculate SEER and COP (Leonel, 2004):

$$KW = \sqrt{3} \times E \times I \times PF = 1.37 \text{ x} \times E \times I \times PF \quad \ldots\ldots (2.9)$$

Where:

$1.37 = \sqrt{3}$

E = Actual voltage in Volts

I = Actual Current in Amps

PF = Power factor = (0.9 assumed)

$$kW = 34 \quad 13 \times \mathrm{Btu/Hr} \quad \ldots\ldots (2.10)$$

$$\text{Total Heat (Btu/Hr)} = 4.5 \times CFM \times \Delta H \quad \ldots\ldots (2.11)$$

Where:

H = Enthalpy of air at saturation (Btu/LB of dry air) calculated from the Psychometric Chart by using the air dry bulb and wet blub temperatures.

H_1 = Supply air enthalpy

H_2 = Return air enthalpy

$\Delta H = H_1 - H_2$

CFM = AHU air flow volume

4.5 = Conversion constant

$$\text{SEER} = \frac{\text{Btu/Hr REfrigeration Effect}}{\text{Input power in watts}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2.12)$$

Where:

SEER is the single-phase energy efficiency rating

$$\text{COP} = \frac{\text{KW Re frigeration Effect}}{\text{KW Input}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2.13)$$

Where:

COP is the coefficient of performance

$$\text{KW/ TON} = \frac{\text{KW Input}}{\text{Tons Re frigeration Effect}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2.14)$$

Where:

1 TON refrigeration = 12,000 Btu/Hr.

These values indicate that the small HVAC system operates very efficiently, which provides a high SEER of 24.5 and a kW/TON of 0.525 when operating at 48 Hz (80%), and when operating at 60 Hz (100%), it provides a SEER of 18.72 and kW/ TON of 0.64 (Leonel, 2004).

2.4. Impact of Outdoor Temperature on Residential Air Conditioning System

An inversely relation between the performance of residential and commercial air conditioners at high outdoor temperatures. This is important to the electric utilities in the desert south west, where outdoor temperatures can reach 110°F (43.3°C) on hot summer days (ASHRAE Fundamentals, 1993). The thermal load on buildings are affected and lead to increase as outdoor air temperatures increases, while the efficiency and capacity of air conditioners decrease with increasing outdoor temperatures. So when electrical demand is usually greatest, occurring that the air conditioners have their worst performance.

During the months of June through September (Northern Hemisphere), the design temperatures represent values that have been equaled or exceeded by 1%, 2.5%, and 5% of total hours. A standard energy efficiency value, called the seasonal energy efficiency ratio (SEER) also, that air conditioners and heat pumps are rated with, which based on a set of tests specified in the Air Conditioning and Refrigeration Institute (ARI) test procedures (ARI, 1989).

To produce better performance over the whole cooling season for an air conditioner should has a higher value of SEER. A several installation and maintenance factors are affected the performance of an air conditioner. Quantitative data to properly analyze is needed to properly analyze the effect of these factors on the high temperature performance of air conditioners. As example of the affecting of these factors that air leakage into the return duct from a hot attic space reduces the capacity and efficiency of the unit and this causes the unit to run longer and use more energy. The relevant literature summarizes the data available on the effects of maintenance items on the air conditioner performance are the three maintenance items such as; improper charging, return air leakage, and reduced evaporator airflow.

Most manufacturers recommended measured airflow rates were outside of the range for almost every system; actual measured cooling capacity were less than two thirds of rated cooling capacities on averages; hourly fractional operating time increased approximately 6% for every 1℃ increase in indoor-outdoor temperature difference (Stephens et al, 2011).

2.4.1 Impact of Low Evaporator Air Flows

Energy-efficiency is implicated in low evaporator air flows also. (Palani et al, 1992) test shows a cooling capacity decreased linearly until about 50% evaporator air flow, where it decreased much faster. For 90% air flow reduction, the cooling capacity decreased by 76%. Reductions in EER were similar to reductions in cooling capacity. The EER decreased linearly for decreasing evaporator airflow, then after 50% airflow, the decrease became non-linear and the drop was greater. The main conclusion was that to maintain sufficient cooling, at least 50% of the rated evaporator air flow was needed. Table 1 is a Summary of Evaporator Airflow Results. (Parker et al, 1997) mentioned that when simulation and test bench data produced by the study which suggest that a 25% reduction in air flow from 400 to 300 cfm/ton can reduce typical AC system EER by approximately 4%.

Table 1

Summary of evaporator airflow results (Palani, et al, 1992)

Reduction in Evaporator Air flow	Percent of Variation		
	Capacity (Btu/h)	Power (W)	EER (Btu/Wh)
25% Reduction	-7.51	-3.54	-4.20
50% Reduction	-14.71	-8.93	-6.51
75% Reduction	-44.71	-15.43	-34.63
90% Reduction	-76.11	-17.22	-71.14

However, sensible EER, which controls cooling system energy use under thermostat control, is reduced by about 10%. Reduction to evaporator air flow below 200 cfm per ton can lead to coil icing and greatly shorten compressor life.

They conclude that improving evaporator air flow to rated values (often 400 cfm/ton in residential air conditioning systems has the potential to reduce average residential cooling energy use by approximately 10%. This will be best accomplished in new installations by properly sizing duct systems and return grills to reduce duct system static pressure. It was also shown to aim at reducing system static pressure has the potential to reduce fan energy by half and to improve overall system EER by 12% could achieve that when select a proper duct design. It appears that should be avoided to provide adequate dehumidification through long duty cycle run-times during over sizing of AC system capacity.

Another reason to provide emphasis to low flow resistance duct design is the tendency in modern residential AC systems to add increased air filtration. Measurements within the project showed that substitution of pleated "high efficiency" filters typically reduce system air flow by 5%. In existing installations, constrictions which increase return or supply duct pressure drop should be addressed and fan speeds set according to measured return air flow. In instances where existing air flow is deficient, installation of a larger indoor unit or modification to the duct system may be necessary. They conclude that most installation-related problems can be avoided by standard test and balance of the air side residential AC systems. Maintenance issues, such as encouraging filter changes and coil cleaning may also be useful to improve field performance (Long et al, 2011). Under or overcharging a system can have a significant effect on performance as shown in Figure 7 (Parker et al, 1997).

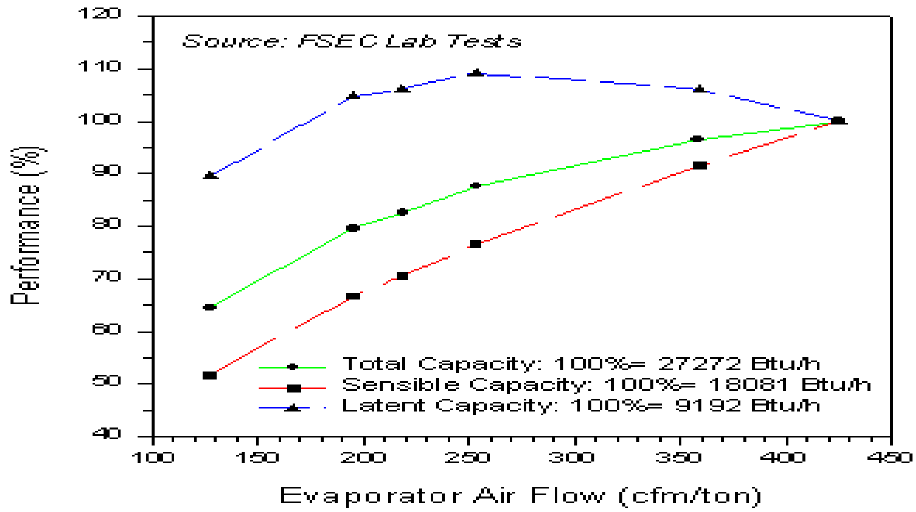


Figure 7. Impact of reduce airflow on cooling capacity (Parker et al, 1997)

Another study reported the effect of evaporator airflow on the frost/defrost performance of a heat pump (Palani et al, 1992). Low airflow degraded orifice performance by lowering the condensing temperature and reducing the inlet pressure. Low airflow also lowered the system's heating capacity compared to the base case. The major effect of low airflow on system performance seemed to be the accelerated decrease in evaporating temperature. Low airflow lowered the evaporating temperature an average of 40% from base case conditions, which caused an acceleration in frost formation. These studies have demonstrated the detrimental effect on air conditioning performance caused by the installation items to be studied: improper charging of the system, reduced evaporator airflow and leakage into return air ducts from hot attic spaces.

2.4.2 Past Laboratory Test of Impact of Low Air Flow

To expand on previous work, a test bench was established to evaluate the impact of reduced evaporator air flow in a controlled environment. The testing took place in a laboratory for testing residential air conditioners with cooling capacities up to 3.5 tons (12.32 KW).

Past Laboratory test result of reduced evaporator airflow rate supplied is negatively effects on AC performance (Parker et al, 1997).

2.5 Correlation Between HVAC Systems and Electrical Energy Use

Approximately 20% of the total electrical energy use in the residential sector (HVAC) applications attributed for heating and air conditioning systems (ASHRAE, 1991), but in this time followed the newest ASHRAE and design conditions should implement ASHRAE 62.1-2010 ventilation for acceptable indoor air quality and AHRAE 55 for thermal comfort (ASHRAE, 2013). ASHRAE 55 standards meet the thermal comfort level so people are comfortable enough while inside buildings (Taleghani et al, 2013).

In the summer season electrical demand for electric utilities is usually greater when outdoor temperatures are higher. For this demand, two important contributors to residences are electric heat pumps and air conditioners. With increasing outdoor temperatures both have lower capacity and efficiency. The reason is attributed to electric utilities are interested in reducing peak summer electrical demand; they have begun to focus attention on the performance of residential air conditioners at high outdoor temperatures.

If the system is not installed properly, the most efficient system will not perform as expected. Very important in the determination of the performance of these units at high outdoor temperatures are the installation and maintenance items such as:

1. Improper amount of charge in the system,
2. Reduced evaporator airflow and air leakage in the return air duct from a hot attic space.

2.6 Improving HVAC system performance

About half of total energy usages in buildings are consumed by (HVAC) systems. So the improving performance of HVAC system is essential to reduce the building‘s energy usages; and this can be done through system design, construction, equipment efficiency, scheduling, ventilation control strategies and so on. The highest potential in reducing energy usages could be achieved depending on operating HVAC and ventilation systems based on demand response is identified (Bel et al, 2009). Moreover, to achieve the high performance of the system that might be very important with the proper installation and regular maintenance. To avoid the problems could be lead to degradation in system performance and thermal comfort in the space; it is very necessary following some stapes:

- Avoid the improper amount of refrigerant charge to the system.
- Do not reduced or use more evaporator airflow
- Avoid the air leakage in the return air duct from a hot attic

2.7 Performance Assessment

Several recurring factors were found to account for the inadequate flows:

- Return ducts and return grills were often undersized
- Fans were set to medium rather than high speed for cooling operation
- Filters and cooling coils were dirty with high flow resistance
- Duct system static pressures were elevated due to circuitous runs, pinched ducts etc.
- Larger outdoor units were installed without changing the indoor unit.
- Devices had been added which increased system static pressures.

(Rodriguez, 1995).

CHAPTER 3

Experimental and Simulation Procedures

3.1 Introduction

The amount of air flow that specified by the manufactures is required to achieve a high level of performance in an air conditioning system. However, the actual amount of air flow could be lower or higher than the specified amount due to several design and operating factors. This chapter discusses these factors and presents a detailed description of the testing facility and testing procedure, as well as the experimental results. The chapter also includes the simulation procedure used in this study.

3.2 Testing Facility

The experiments are done in the heating, ventilating, and air conditioning HVAC lab which consists an Education Heat Pump Lab. It is located at North Carolina Agricultural and Technical State University. The lab is 920 ft^2 (40ft x23ft); with a ceiling height of 9 ft. The space represents one thermal zone. The lab has typical walls. One of the two long sides is exposed to an air conditioned corridor, and the other side is exposed to the outdoors with windows which covering 60% of its net area.

There is a three tons splits heat pump system already installed and It is assumed that the unit supplies 400 cfm per ton of cooling which is considered as the base line of our experiments. The condenser/compressor unit shown in Figure 8 is located outside the lab on the roof of the main building. The other main components of the system such as: air handling unit AHU, air distribution system, digital panel, and other components are installed inside the lab as shown in Figure 9. The air conditioning heat pump system with cooling capacity of 3 ton should provide heating/cooling capacity of 36000 Btu/hr (10.552KW). The cooling part is vapor compression system and electrically-driven; while the heating part is reversed cycle vapor (heat pump) with a supplemented electric heat resistance in the air handling unit. The thermostat, which provides on-off control of the system, is located in the entry hall of the laboratory.

Figure 8. The condenser/compressor unit of typical residential split-system heat pump.

The heat pump is connected to a complete Air Distribution System (ADS) including five diffusers and one Direct Digital Circuit (DDC) with independent-pressure Variable Air Volume (VAV) box. The components of the unit consist of a main duct with a rectangular cross section; this has a long right and left branches with six main diffusers for cooling air distribution. The blower fan is inside the main duct which is to blow the flow air through the cooling or heating. All components are connected to the condenser/compressor and some control logics to form the heat pump system. Figures 9 shows the heat pump system inside the Civil Engineering Department HVAC lab.

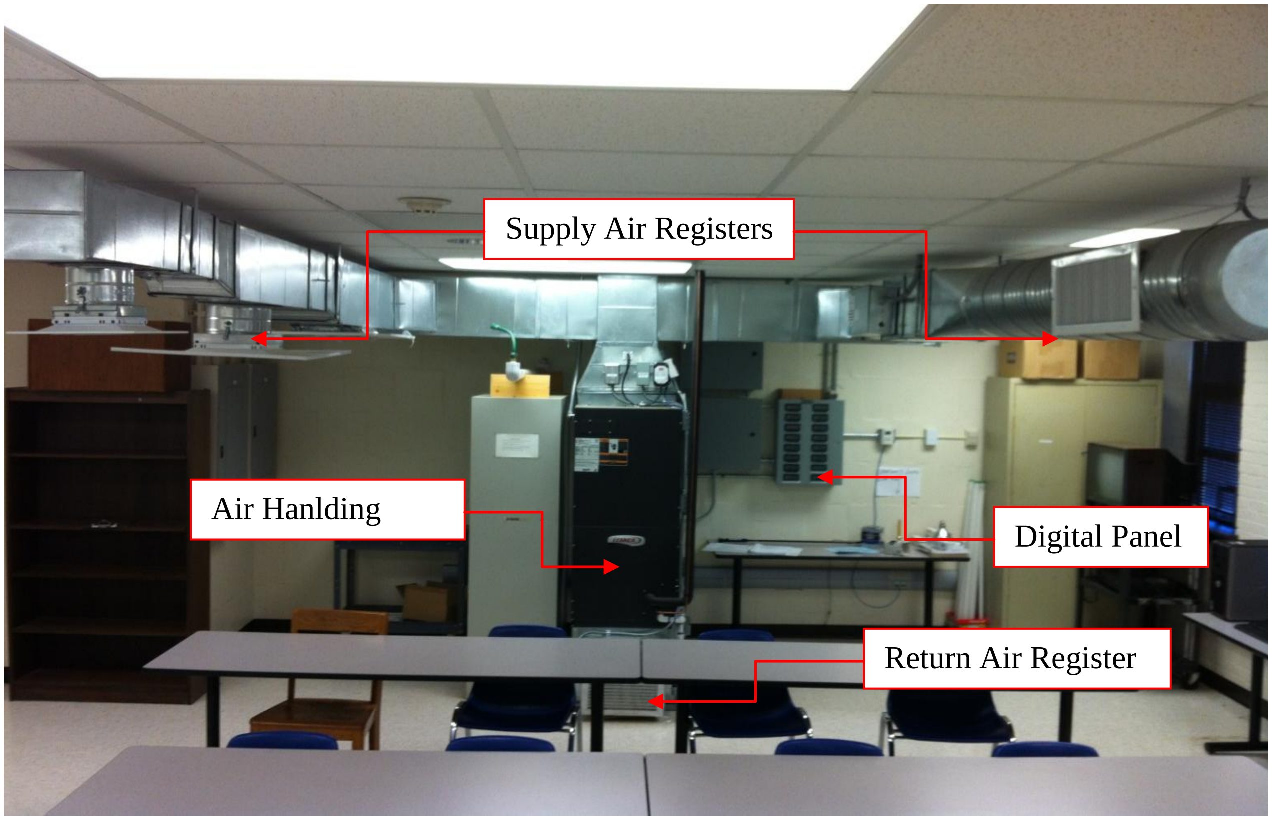


Figure 9. Heat pump system inside the Civil Engineering Department HVAC lab

The main controller is attached to the split system central heat pump unit. This controller is controlling the major system components such dampers, fan, expansion valve, and compressor. Another controllers are attached to the VAV box which is located in one of the branches of air distribution system ADS. The airflow rate in the VAV box is controlled by two PID cascade loops; where the first one determines the airflow rate set point, which is based on the error between the actual space temperature and its set point. The second loop controls the VAV damper to main the supply airflow rate set point. Figures 10 (a) shows the Digital Panel and (b) show the Digital Panel connected to the controllers. Figure 11 shows the thermostat attached to the AHU of the system.

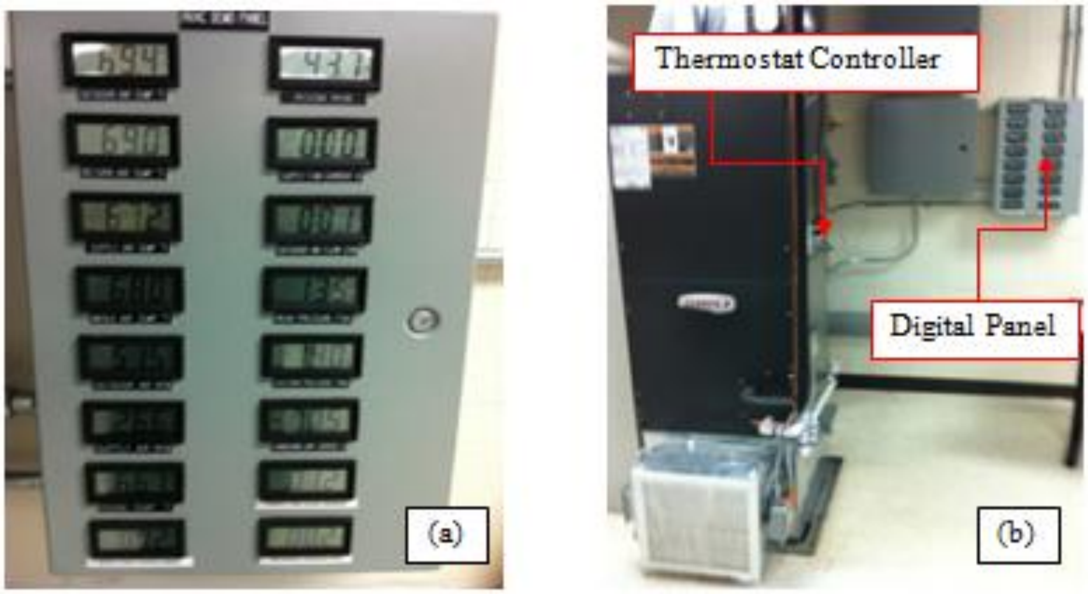


Figure 10. (a) Digital Panel, (b) Digital Panel connected to the system controllers

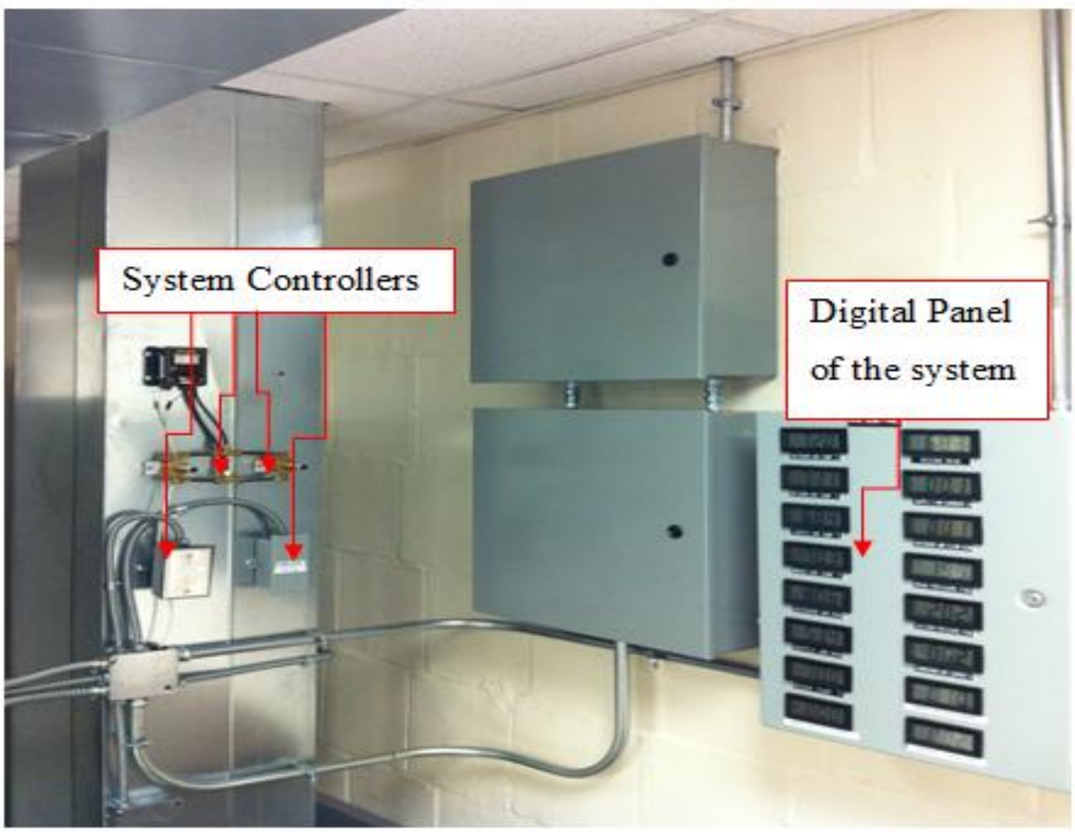


Figure 11 The thermostat attached to the AHU of the system

The Education heat Pump control is tied to campus-wide Alerton BAS as shown in Figure 13 (Education Demo is at top right tab). This configuration gives us the opportunity to perform the lab tests on Ed HP. The system is controlled by Alerton Building Automation System (BAS) as shown in Figure 12.

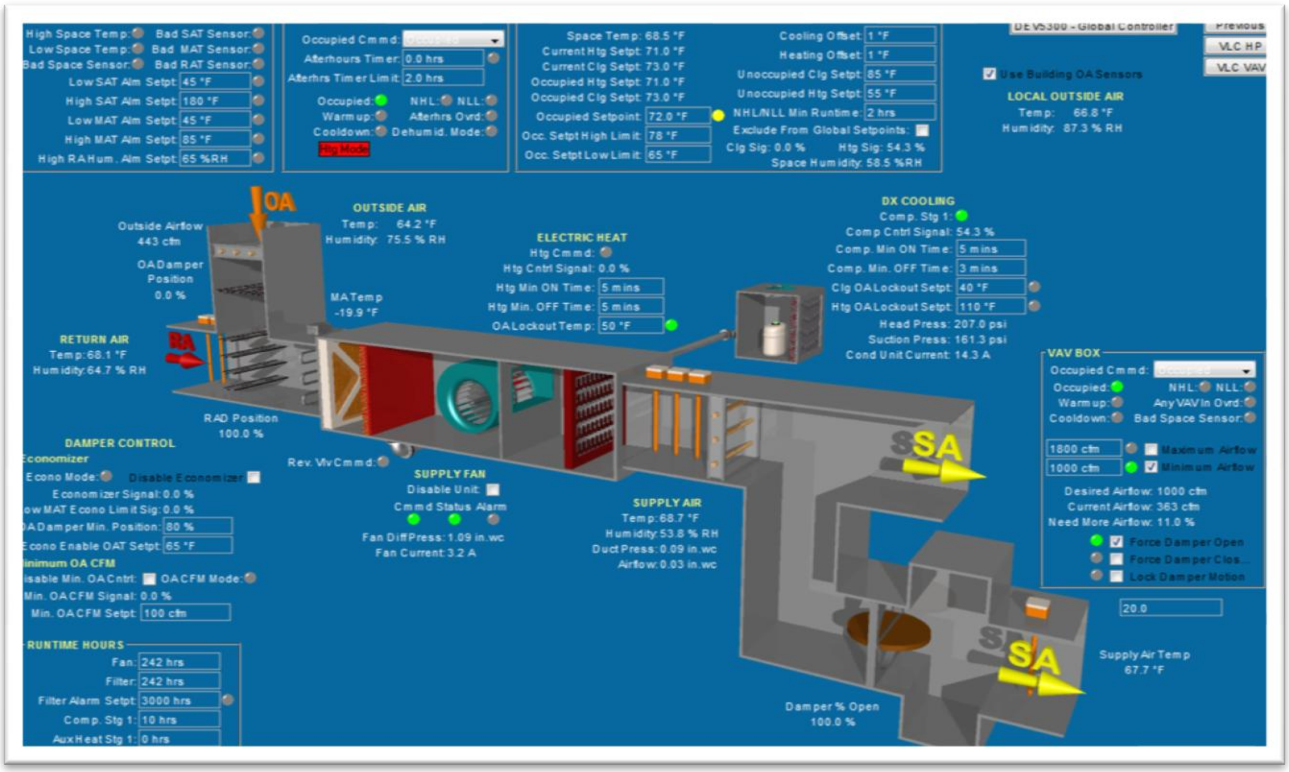


Figure 12. Screenshot of the (Alerton BAS) for the central unit heat pump

3.3 Test Procedure

Figure 13 shows a schematic diagram of the heat pump located in HVAC Lab. Airflow rate through the heat pump is varied by varying the outside air damper and while keeping the return air damper fully closed. The damper position changes from 10% to 100% and the corresponding airflow rates and static pressure across the fan are measured. Table 2 shows the measured data during the test. The fan current is also measured during the test to determine the fan efficiency with various airflow rates.

The relationship between the air flow rate supplied cfm and the pressure difference across the fan of the system is developed through the experiment for this purpose. The fan electrical current (Amp) is measured. Then the fan hours powers (Hp) are calculated by using equation (3.4), and the fan efficiency is calculated with equation (3.3).

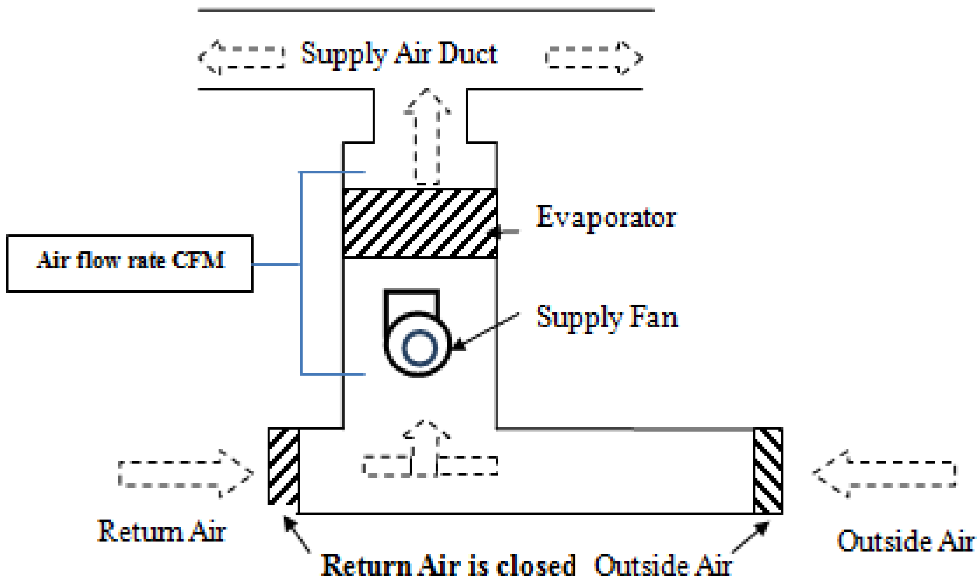


Figure 13. A schematic of the split-system heat pump located in the lab

Table 2

Measured and calculated data from experiments

Test #	Damper Opening %	Air Flow (cfm)	ΔP (inW.g)	Fan Current (Amp)	Calculated Fan Power (KW)	Calculated Fan efficiency
1	10	460	1.43	3.5	0.735	0.1905
2	30	485	1.42	3.8	0.798	0.1837
3	50	690	1.32	4.8	1.008	0.1923
4	75	900	1.2	6.1	1.281	0.1794

3.4 Fan Performance Curve

The fan performance curve is developed from the experimental data and expressed by fitting fan curve which is represented by the following quadratic equation:

$\Delta P = a_2 * Q^2 + a_1 * Q + a_0$ …………………………………………………….. (3.1)

Regarding to this equation, the plotted fitting fan curve obtained from the experimental data in Excel which is given by the following equation:

$\Delta P = -2^{-0.07} * Q^2 - 0.0003 * Q + 1.7302$ ………………….………………....... (3.2)

Where:

ΔP: is the pressure difference across the fan Inch Water Gage (inW.g)

Q: is airflow rate supplied (cfm)

a_0, a_1, and a_2: are constant factors that are determined by fitting the measured data, and

$a_0 = 1.7302$

$a_1 = -0.0003$

$a_2 = (-2^{-0.07}) = -0.90533$

Using the equation 3.2, the airflow rates supplied and the associated fan pressure used in the simulation are shown in Table 3.

Table 3

Airflow rate and associated pressure difference used in the simulation study

Supplied Airflow (cfm)	Pressure Difference Across Fan (in W.g)
500	1.5302
650	1.4507
800	1.3622
950	1.2647
1100	1.1582
1250	1.0427
1400	0.9182

Equation (3.2) refers that ΔP is a function of Q and inversely proportional with it. (This means when ΔP decreases, Q increases as in Table 3)

Figure 14 shows the plotted fan curve obtained from the experimental data.

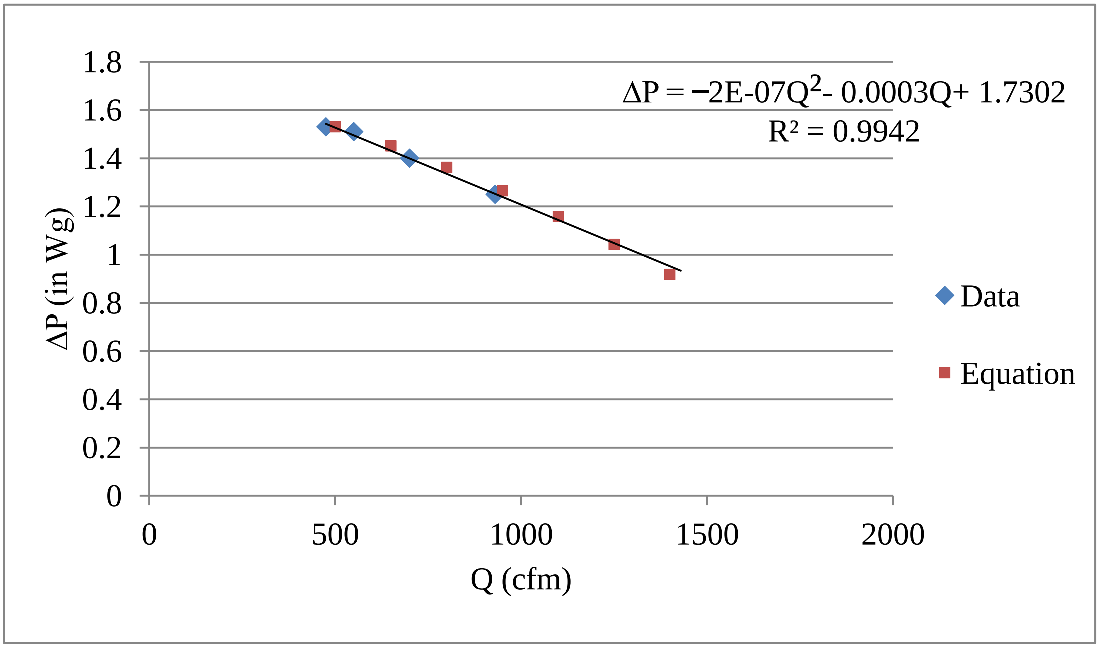


Figure 14. System performance fitting fan curve obtained from the experiment

3.4.1 Fan Efficiency

At each airflow rate and its associated pressure, the fan efficiency is also determined by the following equations:

η_f = (Q *Δp) / (6350 * P_{fan} *0.74)... (3.3)

Where:

Q = measured airflow rate supplied cfm

Δp = measured pressure difference across the fan in W.g

P_{fan} = fan power kW

The fan power is calculated using the following equation:

Pfan = Pf × I × V ……….. (3.4)

Where:

Pf: Power factor that is assumed to be 1

I: the current (Amp) that is measured during the test

V: the voltage that is assumed to be 210 (based on heat pump specifications)

P_{fan}: is the power of the fan in Hp and changed to kW (multiplied by 0.746)

Hp = 0.746 kW

η_f = fan efficiency

The calculated fan power and efficiency are as shown in Tables 3.

3.5 Energy Simulation

3.5.1 Creating the Building Simulation Model

A building model is created to explore how the different amount of air supplied by the system could affect the other parameters of the air conditioning system, which includes system's performance and space thermal comfort. Several types of software such as Building energy optimization (BEopt) and (eQuest) are capable of analyzing and evaluating building and the HVAC system's performance.

BEopt software is suitable for residential buildings, while eQuest can utilize for both residential and commercial buildings. The simulations are done by eQuest due to its features and the options of using the annual and hourly outputs. In this work the simulation by using BEopt 1.3 software which is done as shown in Figure 15. Then the model has been exported to eQuest software and customized as shown in Figure 16. The residential building model created in the software was a residential house with an area of 1600 ft^2. This square footage was chosen, corresponding to the three tons capacity heat pump system that is used in the HVAC Lab. The building model is for a typical constructed house that complies with the ASHRAE Standard

Construction Code. Assumption was made when changing the location of the house model. It is assumed that by choosing the code analysis, the code version and justification of ASHRAE Standard 90.1, the construction would be justified through the software based on the code for each location.

3.5.2 House Description

The house is selected with following specification:

Location: Greensboro, North Carolina

Area: 1600 ft^2, One Story

Construction: Typical construction compiles s with ASHRE Standard Construction Code

Walls: Bricks and wood (3/4 in fiber, BD sheathing, R- 2)

Building Type: Mixed use

Insulation: R-19

Interior Finish: Carpet (No Pad)

Roof Surface: Metal Frame > 24 in O.C

Door Type: Opaque

Window Size: 7ft x 3 ft

Glass Category: Double Clear ¼ in, ½ in, Air

Cooling and Heating: Heat Pump System

HVAC Size: Three Ton Split AC

Weather: As ASHRAE Climate Zone

Exposure: Earth contact

Code Analysis: LEED – AC (Appendix G)

Orientation: As shown in Figure 16.

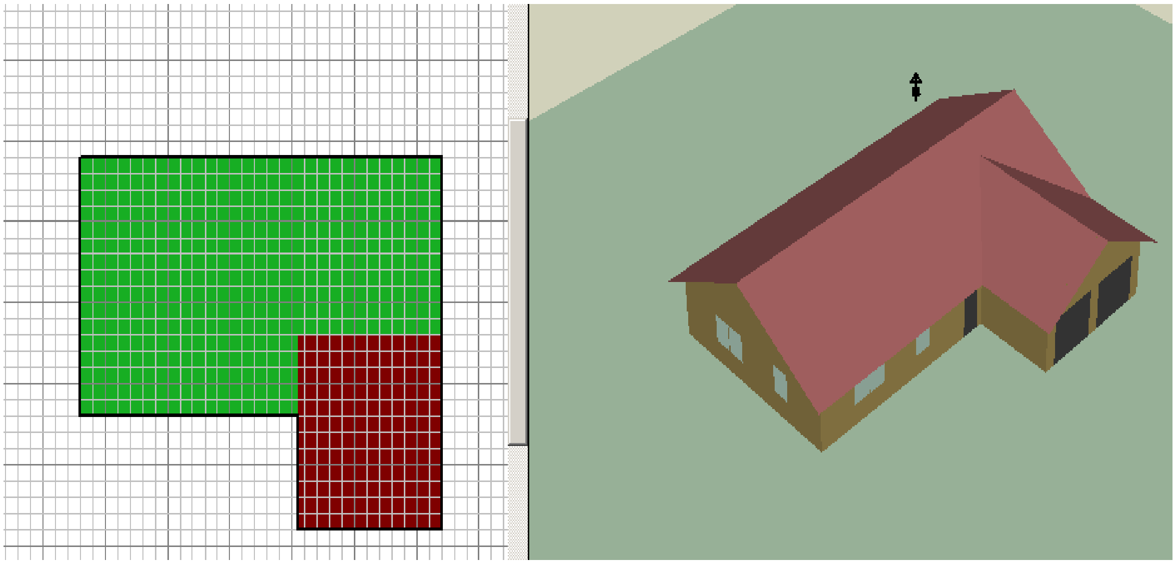


Figure 15. Three Dimensional of the house model by BEopt 1.3.

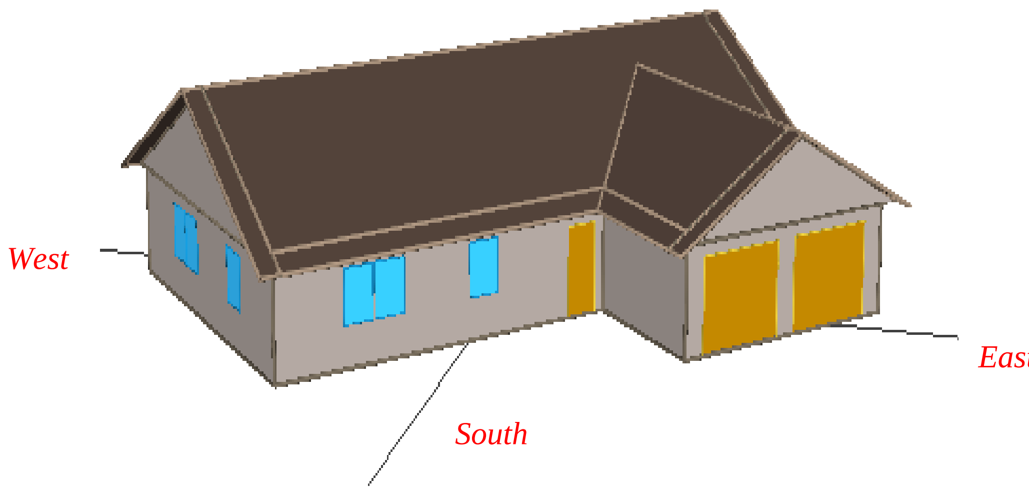


Figure 16. Three dimensions of the house simulation model by eQuest

Figure 17 shows the code analysis and the justifications in the project & site interface. The air conditioning heat pump model is adjusted in eQuest to imitate the tested system by customizing the system capacity and by using fan variables airflow charge (cfm) and pressure differences throw the fan; Figure 18 shows system components that used in eQuest.

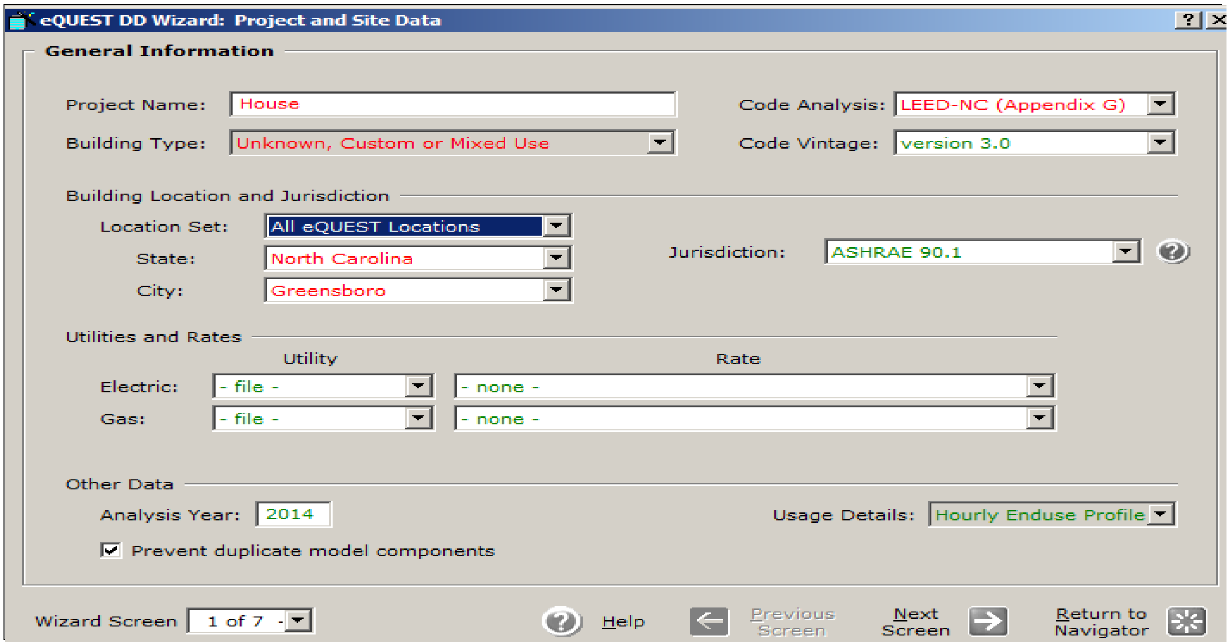


Figure 17. Code analysis and site data interface in the detailed data wizard eQuest

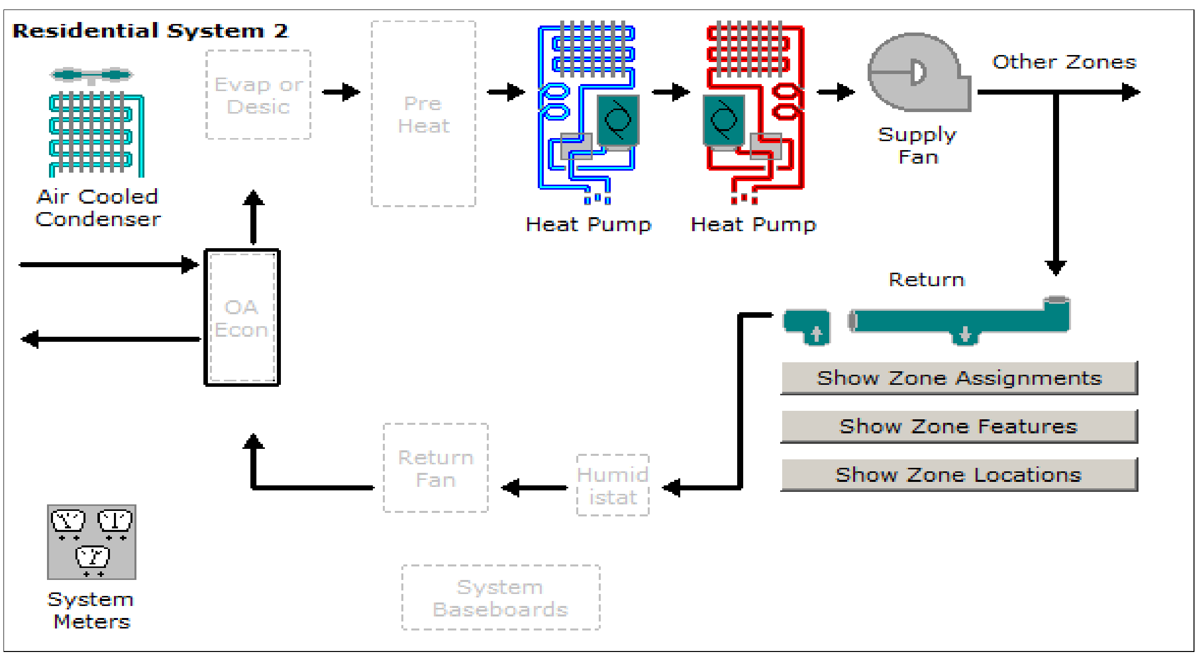


Figure 18. System components used in eQuest

It should be noted that the capability of changing the location of the house through this software could be a supportive feature to the study. It can help in exploring the effect of different weather on the system's performance hen the system has various flow rates.

3.5.3 Model Simulation

At this stage, the model is ready to simulate all variables, input data and the energy use. Also, it can explore any changes in the parameters that could occur when modifying the variables. The two types of the analysis can be obtained from eQuest software when running the simulation. The first is the reports of analysis and investigation of the annual energy use, and the second is the reports of analysis and investigation of hourly energy use.

As mentioned before the variables are the airflow rate supplied cfm and the corresponding pressure inW.g which obtained previously by the experiment. While the other parameters are:

1. The total annual energy use of the model which can be obtained from the output data reports of the total annual energy use.
2. Hourly energy use of the building in terms of sensible, latent and total loads which can be obtained from the hourly output data reports too.
3. Hourly return air humidity ratio obtained from the hourly output data reports to indentify space comfort.
4. Hourly energy use of the fan obtained from the hourly reports.
5. Hourly change in the temperature of air supplied to the space obtained from the output data hourly reports.
6. Operation hours of the system obtained from the output data hourly reports.

To carry out the simulation, first, the model was adjusted to the new variables of the air flow and pressure. The modifications made to the model through detailed data wizard. The data that has been changed can be found in Table 3. The tested flow rate starts from 900-1400 cfm with the consequent pressure across the fan that was obtained from the test. Figures 19 and 20 show the detailed data wizard conducted with the variables that have been changed.

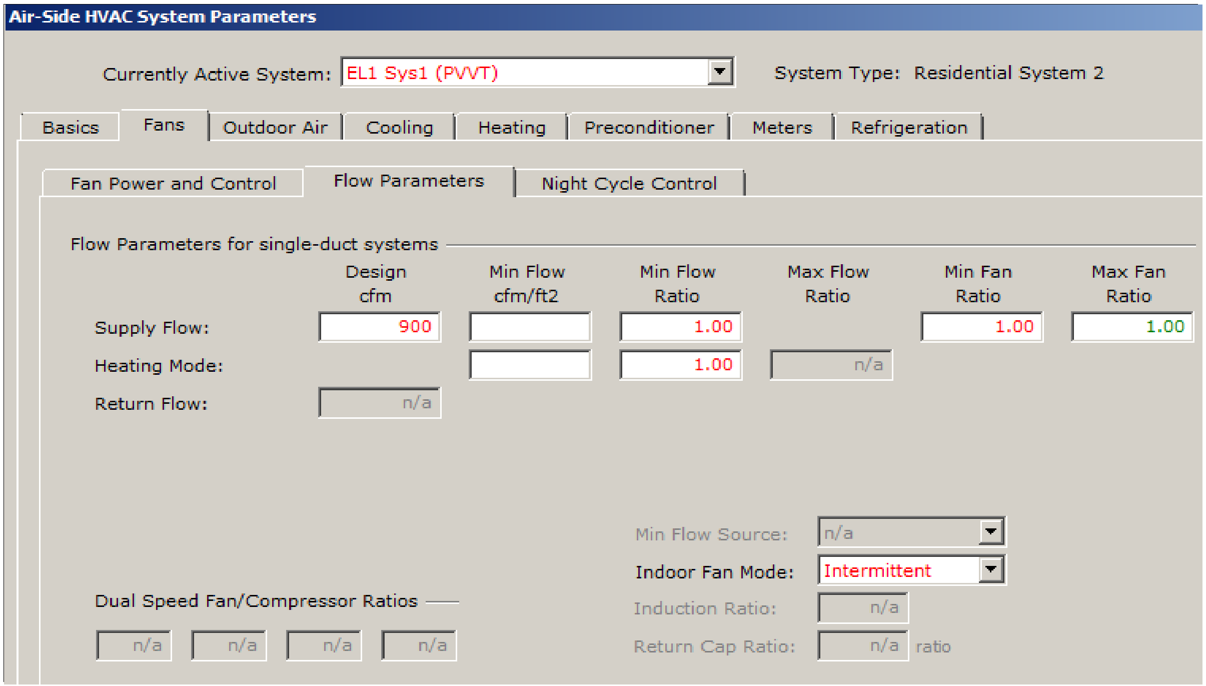


Figure 19. Flow rate (cfm) in HVAC system variables used in eQuest

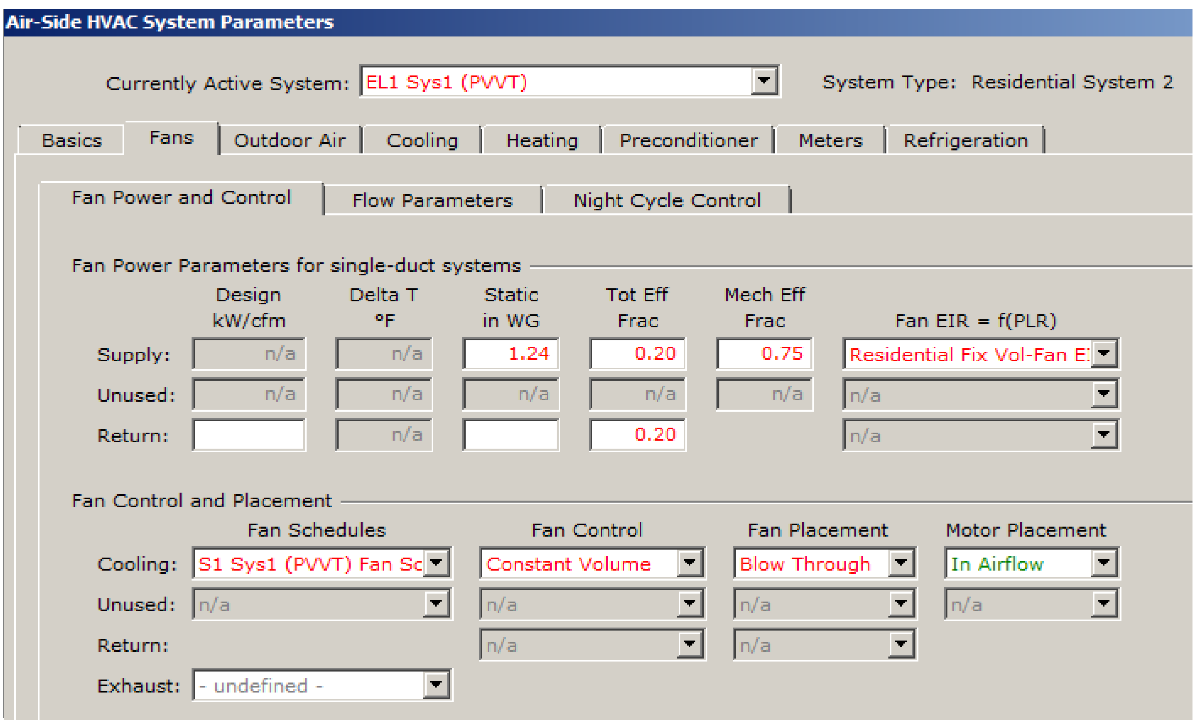


Figure 20. Pressure difference in HVAC system variables used in eQues

CHAPTER 4

Simulation Results

4.1 Introduction

In an HVAC system the supply airflow rate is a major component affecting the performance of the system. Hot air is supplied to heat a room and cold air is supplied to cool and dehumidify a room. The measurement of airflow rate is often necessary to confirm that the HVAC system is performing as designed. In this research, an energy simulation program (eQuest) is used to study the effect of airflow measurements on energy consumption. Furthermore, the measured performance fan curve (a graphic representation of the fan output (cfm) versus the total head pressure across the fan) is considered by the simulation program. All results used in this study are to demonstrate the effect of airflow rate on the performance of HVAC system. Analysis of the simulation results are described in details in the next sections.

4.2 Simulation Results

The annual and hourly energy consumption simulation results are obtained for each HVAC airflow supply rate (cfm) within a single family residential house. The house is using a three ton heat pump system with constant fan speed and constant volume. Various design air flow supply rate (cfm) are selected. The airflow rates selected are different than those specified by manufacturers and vary from 900 cfm to 1400 cfm. Simulation results in term of energy consumption used for different systems (heating, cooling and fan) are used to analyze the effect of air flow rates (cfm) on system performance and space thermal comfort. The impact of weather condition on air flow is also considered.

The weather conditions are selected from the US ASHRAE Climate Zones. Two extreme weathers are selected: humid hot in Miami, FL and dry hot weather in Las Vegas, NV.

Two other locations are selected as mixed humid moderate in Greensboro, NC and humid cold in New York, NY. The extreme weather conditions are discussed in detail in the annual energy analysis and in the hourly energy analysis sections.

4.3 Energy Usage Analysis

4.3.1 Annual Energy Usage Analysis

The performance of the house air conditioning system is determined using energy simulation software, considering different airflow rates and fan static pressures. The performance is discussed in terms of annual cooling and heating energy use and fan energy use. The house model is prepared using eQuest, where the baseline is considered for a 1200 cfm airflow supply fan. The airflow has been selected to meet the specification of 400 cfm per ton of cooling. The analyzing of the annual cooling, heating, fan and total energy use for the four locations, indicate that the total energy use increases with an increasing in the air flow rates supplied to the space.

In Miami where the hot humid weather, the impact of supply different air flow rate are tested. Results show that with higher air flow the cooling energy use is decreased while both of fan and total energy are increased with no change in heating energy use. In Las Vegas, where the hot dry weather, results show with higher air flow, the cooling energy use is slightly decreased while both of fan and total energy use are increased with a relatively constant amount of heating. Results of energy use in Miami, Las Vegas as shown in Table 4, and results of energy use in Greensboro and New York as shown in Table 5.

Table 4

Annual energy use for Miami & Las Vegas (KWh x1000)

Airflow rate (cfm)	Miami				Las Vegas			
	Cooling	Heating	Fan	Total	Cooling	Heating	Fan	Total
900	4.89	0.02	2.24	16.09	3.3	1.17	2.12	15.69
1000	4.84	0.02	2.15	15.95	3.3	1.17	2.08	15.63
1100	4.85	0.02	2.23	16.04	3.28	1.17	2.12	15.65
***1200**	**4.84**	**0.02**	**2.47**	**16.27**	**3.27**	**1.16**	**2.26**	**15.79**
1300	4.8	0.02	2.7	16.46	3.27	1.16	2.36	15.88
1400	4.76	0.02	2.91	16.63	3.26	1.16	2.41	15.92

(*) Baseline.

Table 5

Annual energy use for Greensboro & New York in (KWh x1000)

Cfm	Greensboro				New York			
	Cooling	Heating	Fan	Total	Cooling	Heating	Fan	Total
900	1.48	3.09	2.06	15.97	1.08	4.32	2.45	17.33
1000	1.47	3.08	2.10	15.98	1.07	4.3	2.53	17.38
1100	1.48	3.07	2.16	16.05	1.07	4.29	2.62	17.45
***1200**	**1.47**	**3.06**	**2.27**	**16.15**	**1.07**	**4.28**	**2.72**	**17.55**
1300	1.46	3.06	2.37	16.22	1.06	4.28	2.8	17.61
1400	1.45	3.06	2.43	16.28	1.05	4.28	2.84	17.64

(*) Baseline.

4.3.2 Hourly Energy Usage Analysis

Hourly data is also obtained from the simulation model in order to investigate the variation of supply airflow rate on whole air conditioning system performance. The data include:

1. Fan energy use
2. Temperature of air leaving the cooling coil of the system
3. Space humidity ratio
4. The sensible, latent and total load of the space

In the following section, the hourly data for only of one day (May 15) is presented and discussed in details mainly for two locations, Miami and Las Vegas. The results for Greensboro and New York locations are presented in Appendix B.

4.3.3 Fan Energy Usage

Figures 21 and 22 show the hourly fan energy consumptions with various supplied airflow rates, as 1000 cfm, 1200 cfm, and 1400 cfm for Miami and Las Vegas, respectively. It is indicated that with elevated airflow supply, the fan energy consumption are increased.

4.3.4 The Temperature of Air Leaving the Cooling Coil of the System

System cooling coil removes the heat from the air passing through it due to the heat exchange between the cooled coils surface area and the passing air. The amount of air flow can affect the temperature of leaving air. Temperature of air leaving the cooling coil are increased with higher amount of air (cfm) passing through the coils. This means the more of cfm across the coils would result a rising in the temperature and vice versa. Figures 24 and 25 show the temperatures of air leaving the cooling coil with different airflow rates supplied on May 15th for Miami and Las Vegas respectively.

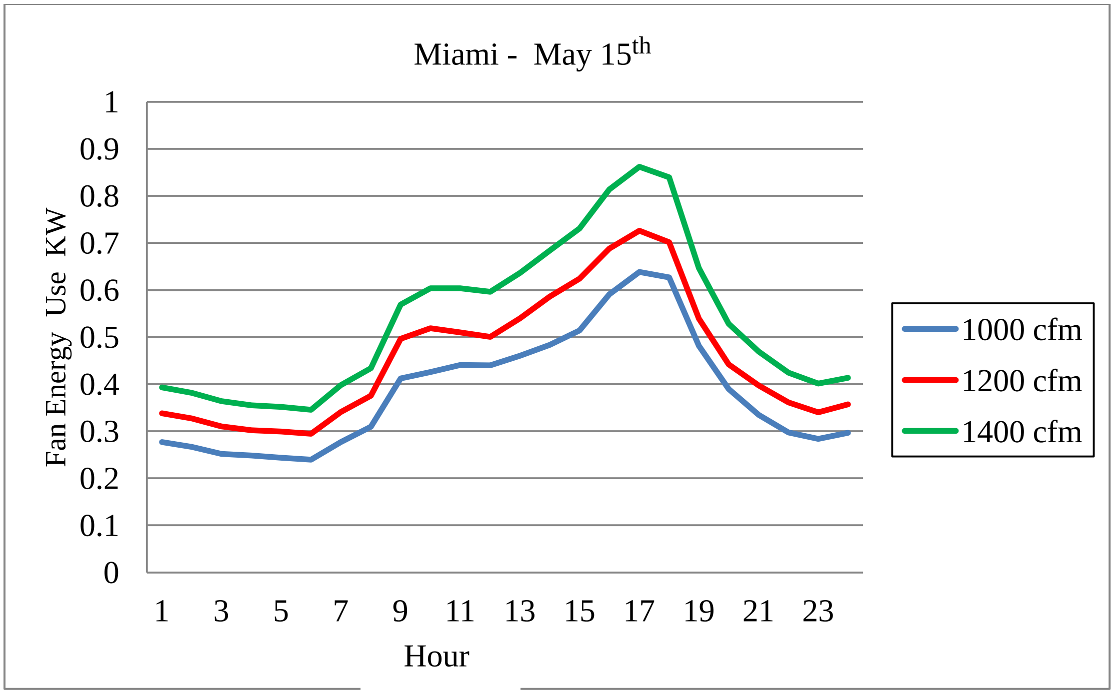


Figure 21. The hourly fan energy use for the day on May 15th in (Miami)

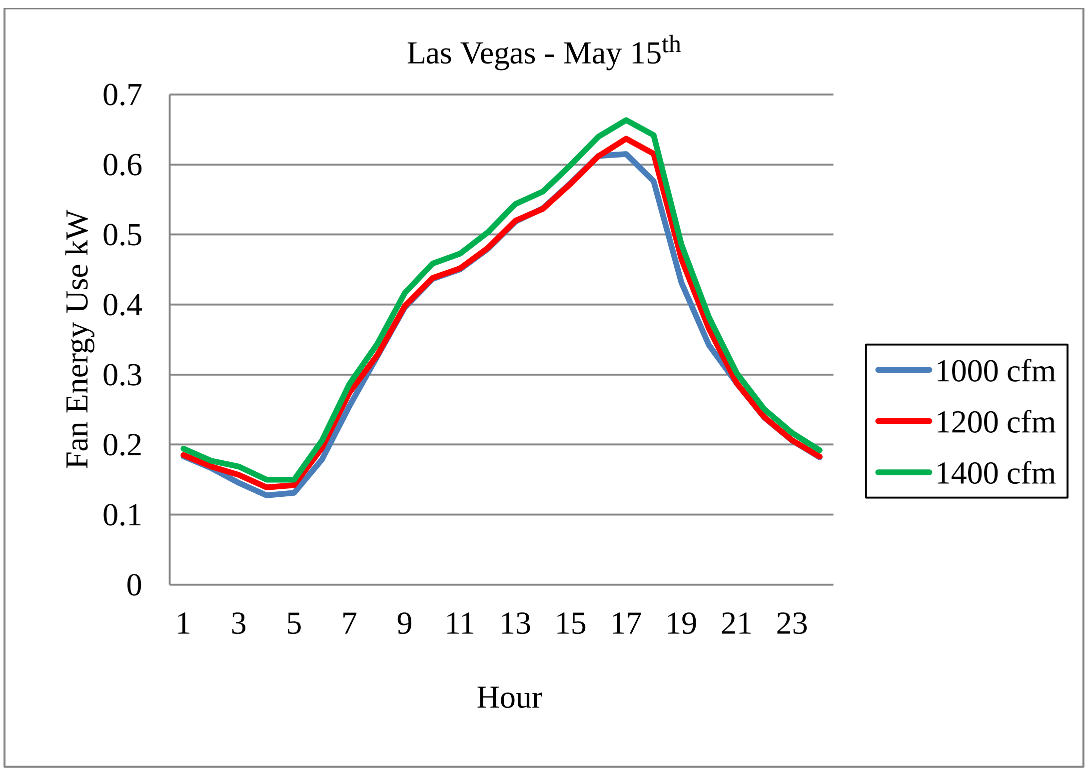


Figure 22. The hourly fan energy use for the day on May 15th in (Las Vegas)

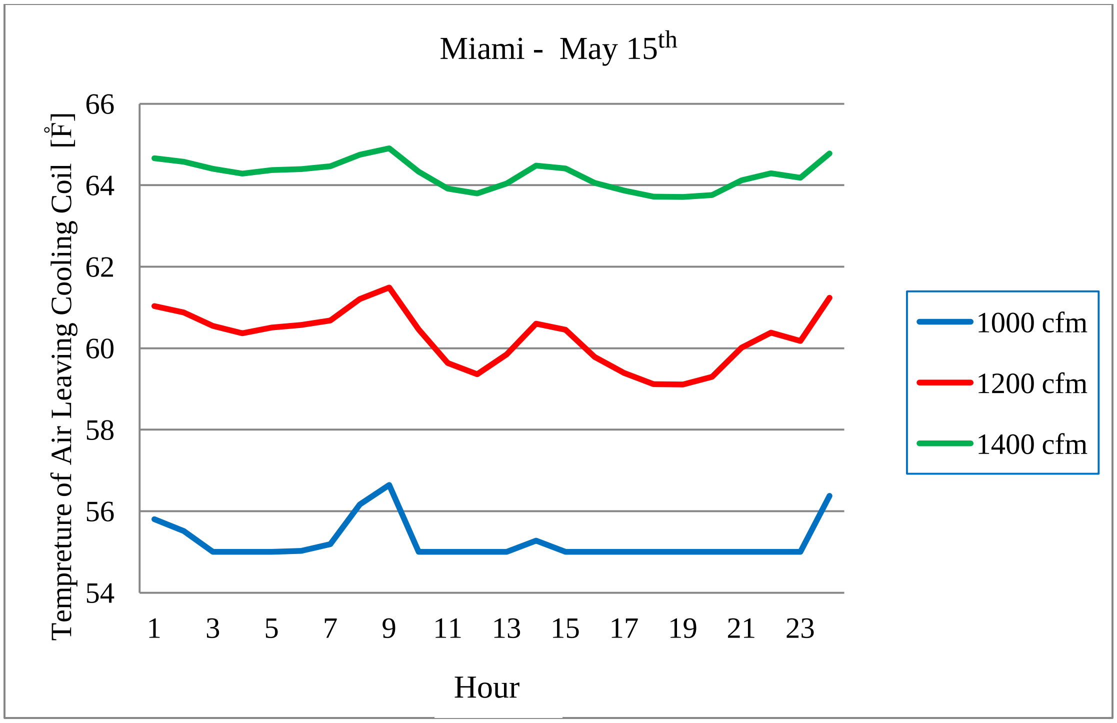


Figure 23. Temperatures of Air Leaving cooling coil on May 15th in (Miami)

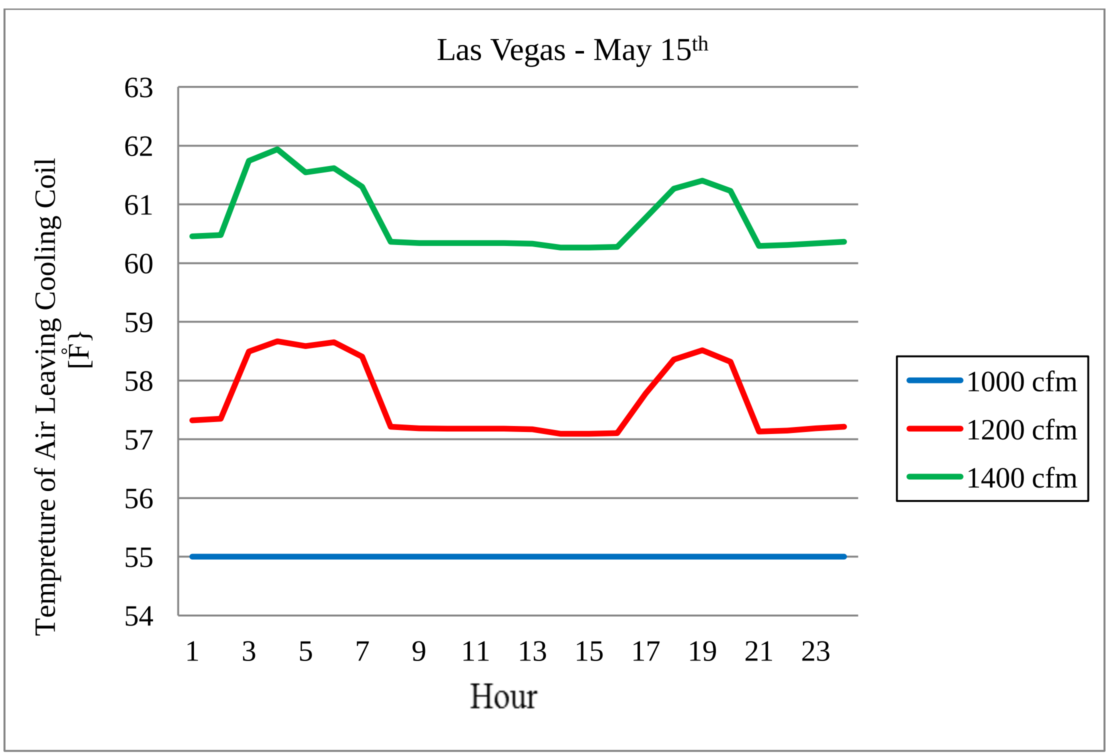


Figure 24. Temperatures of Air Leaving cooling coil on May 15th in (Las Vegas)

4.3.5 Space Humidity Ratio

Space humidity ratio (W) is the ratio between 1 lb of actual mass of water vapor to the 1 lb mass of dry air (lb H_2O / lb dry air). Humidity ratio in the space affects the occupant thermal comfort, especially in locations with a humid weather. While controlling the moisture in a space, the humidity ratio is considered as an important factor of HVAC system that may affect the thermal comfort level. Designing the space with 75℉ of dry bulb temperature and 50% RH give a humidity ratio HR of (0.0092 lb/lb) for 400 cfm per ton cooling load is achieved for an acceptable level of thermal comfort.

The hourly simulation results during May 15th for different locations give an idea about the effect of airflow rates (cfm) on space's relative humidity. Figures 25 and 26 show the space humidity ratio (RH) with different air flow rates supplied.

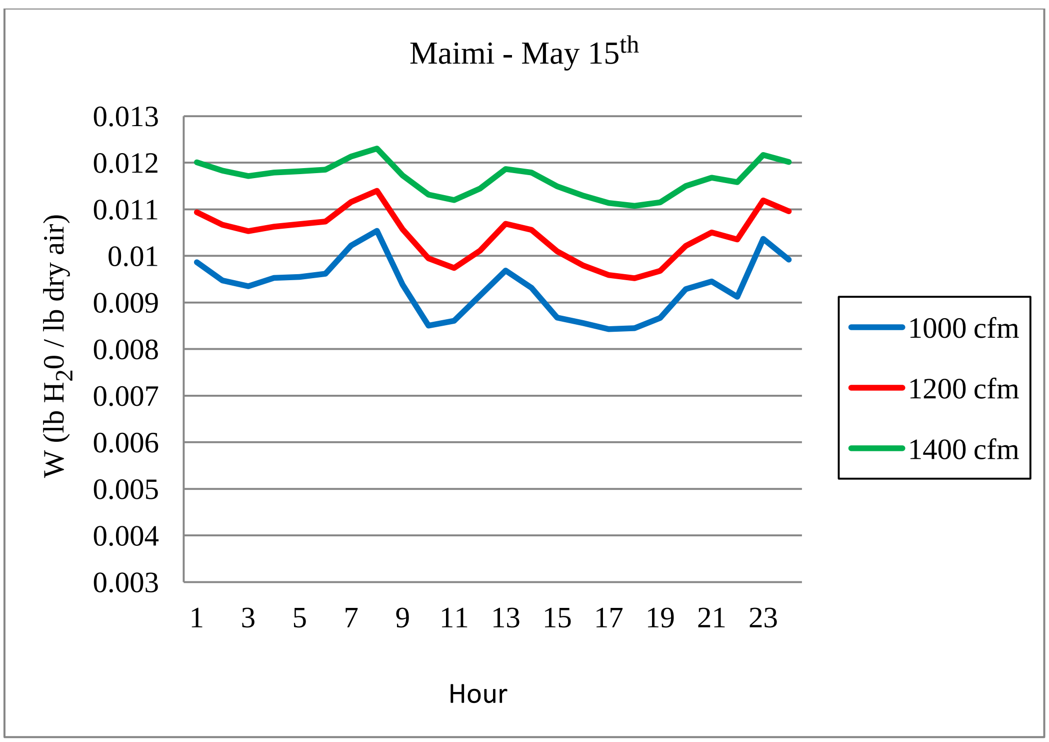


Figure 25. Return air humidity ratio W (lb H_2O / lb dry air) on May 15th in (Miami)

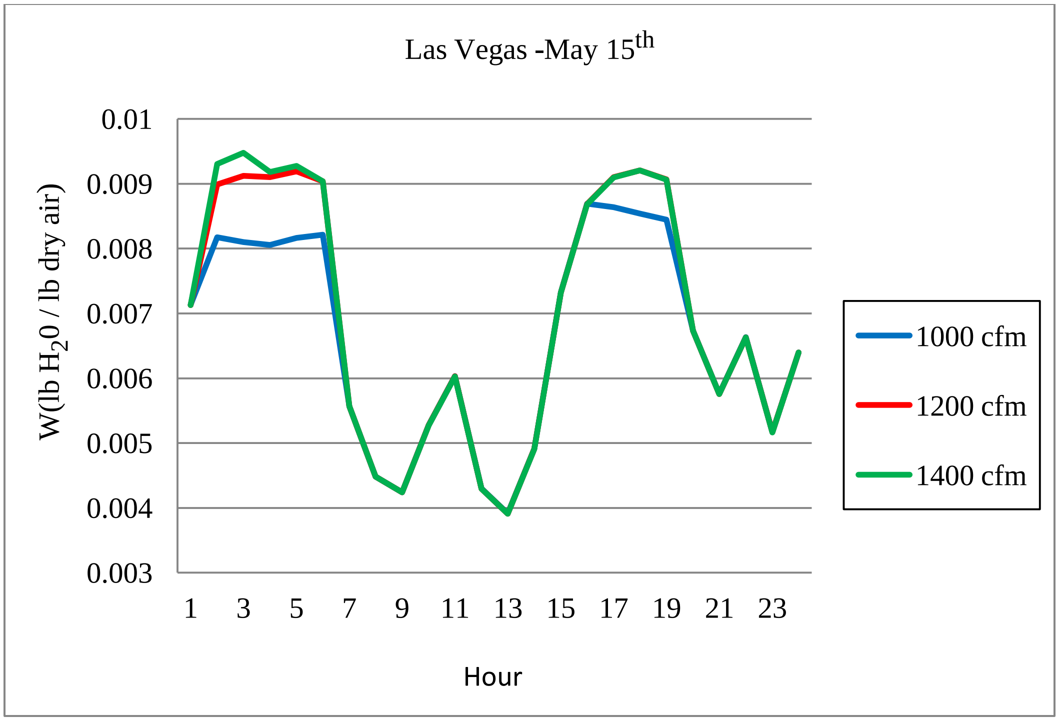


Figure 26. Return air humidity ratio (lb H_2O / lb dry air on May 15th in (Las Vegas)

4.3.6 Sensible, Latent and Total Loads of the Space

Air conditioning system removes heat from the space and that represents the sensible, latent and total loads of the space. Different amount of air supplied from the system to the space can manipulate these loads. The simulation results show the loads are affected differently with different climates. In Miami where the weather is very hot humid, the results show slightly decrease in the total load with the increase of the supply airflow rates (cfm). Also, the sensible load increases, while degradation in the latent load is noticed. In Las Vegas where the weather is hot and dry, the total load remains almost constant, with a slight increasing in term of sensible load and decreasing in latent load. Figures 27 and 28 show the results of the variation loads as a function of the supply air flow rates (cfm) used on May 15th in Miami and Las Vegas respectively.

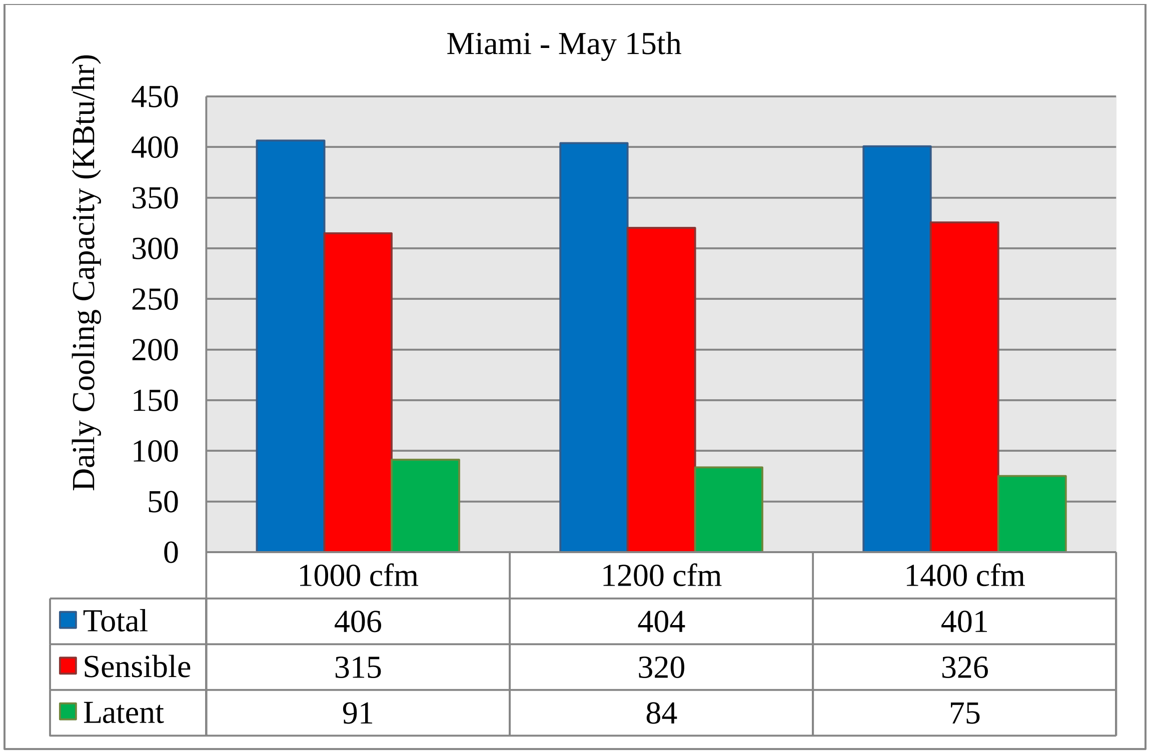


Figure 27. Daily total, sensible and latent cooling loads on May 15th in (Miami)

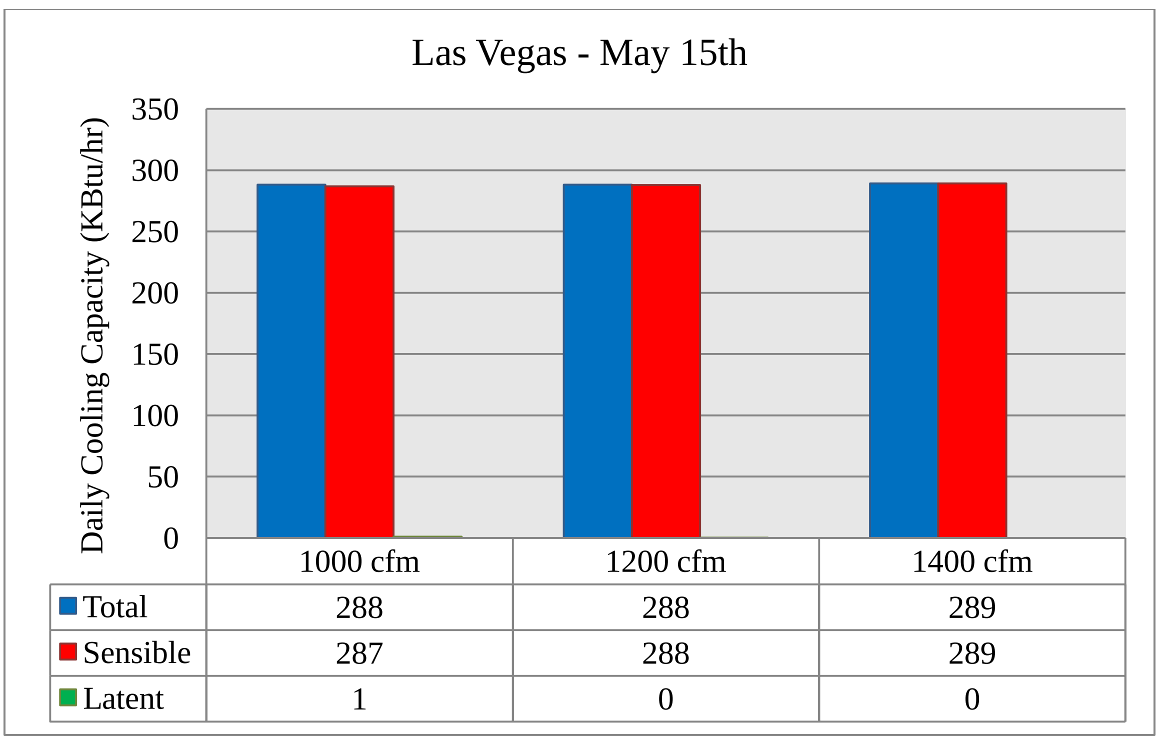
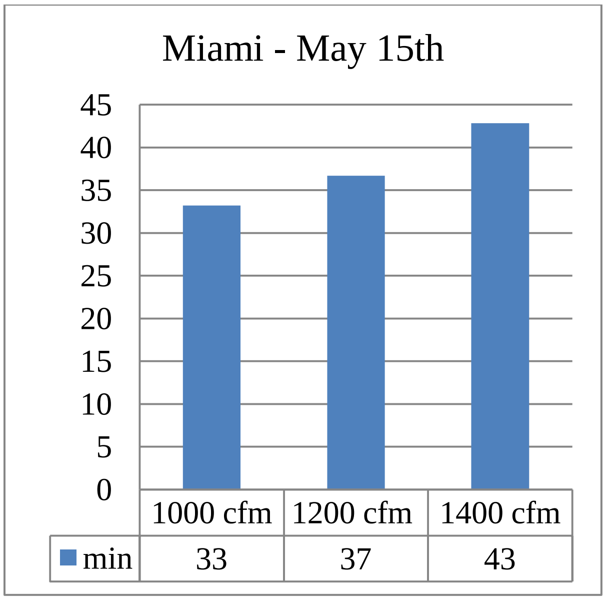


Figure 28. Daily total, sensible and latent cooling loads on May 15th in (Las Vegas)

4.4 System's Operation Time.

System's operation time is the average minute operating per hour during the 24 hours of the day. This is determined form the model hourly output as function various charging of air flow rates (cfm). Two locations are selected for this analysis to represent the extreme and moderate weathers. These locations are Miami and Greensboro that tested on May 15th. Figures 29 show the increase in the system operation time with more amount of airflow rate supplied by the system. A big difference observed in operation time in Greensboro than in Miami.



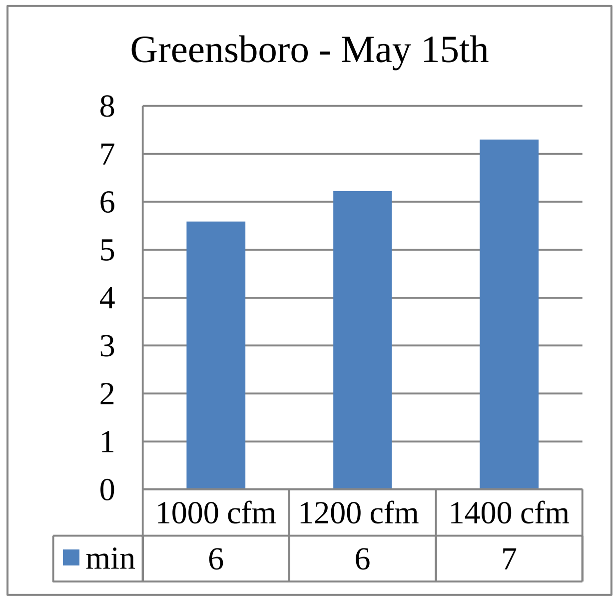


Figure 29. System operation min. per hour, left Fig.for Miami and right Fig. for Greensboro.

4.5 Overall Results for the Four Cities

The energy use for space cooling/ heating and fan (kWh) as function of supplied air flow rate (cfm) are demonstrated in Figures 30, 31, 32 and 33 for the locations of Miami, Las Vegas, New York and Greensboro, respectively. Table 6 shows the total, sensible and latent loads obtained in Greensboro and Las Vegas. Results for Greensboro and New York are presented in Appendix B.

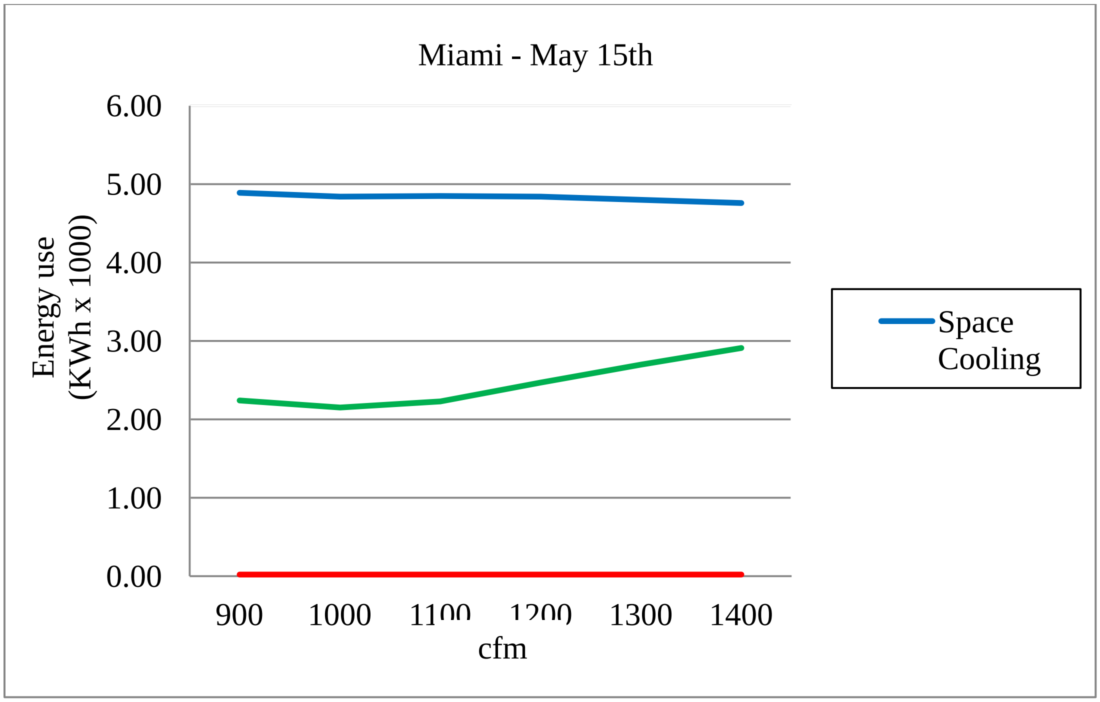


Figure 30. Total annual energy use for cooling, heating and fan in Miami

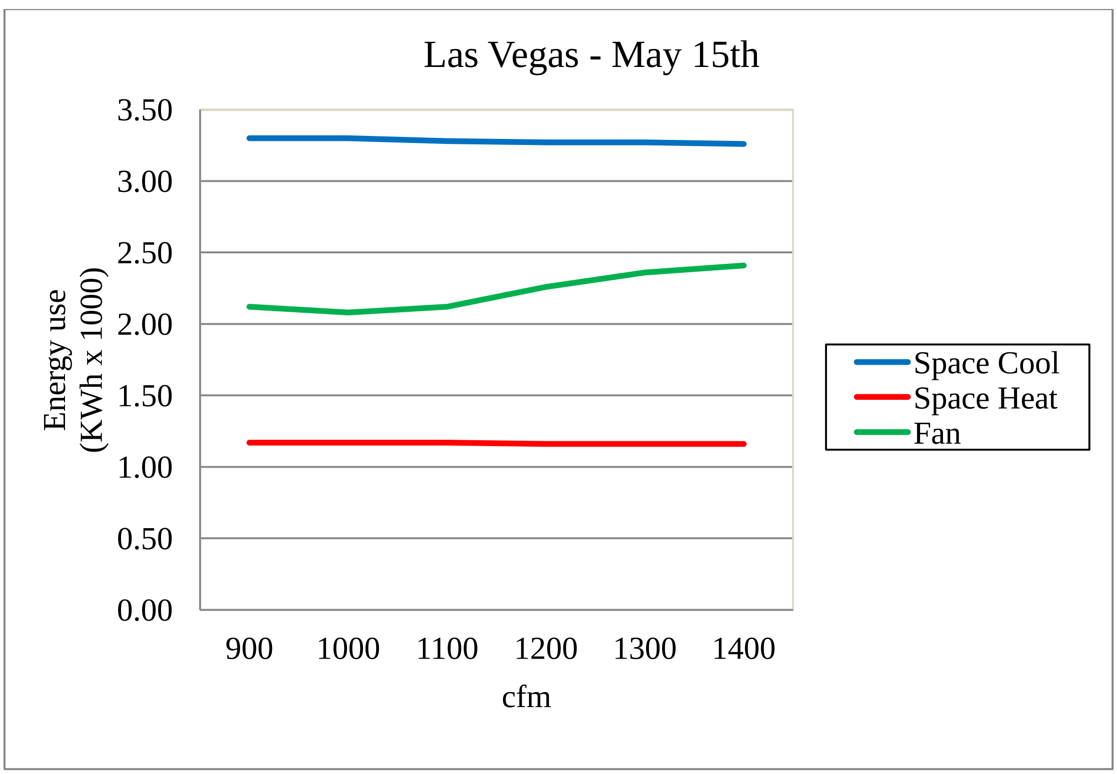


Figure 31. Total annual energy use for cooling, heating and fan in Las Vegas

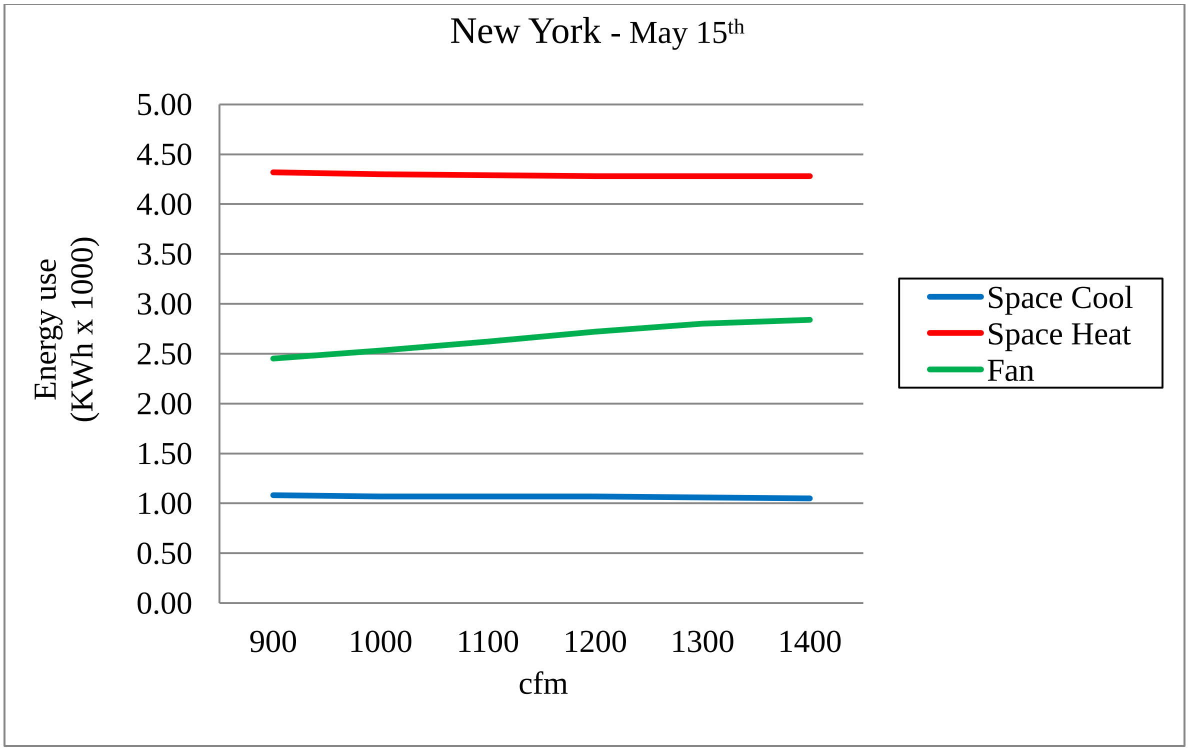


Figure 32. Total annual energy use for cooling, heating and fan in New York

Figure 33. Total annual energy use for cooling, heating and fan in Greensboro

Table 6

Total/Sensible/Latent Loads for August1st in Greensboro & in (KBtu/hr)

Air Flow	1000 cfm		1200 cfm		1400 cfm	
City	Greensboro	New York	Greensboro	New York	Greensboro	New York
Total	268313	268866	270455	268317	267206	265876
Sensible	201073	221951	205070	225585	209001	229734
Latent	67240	46915	65385	42731	58204	36142

Data in Table 6 indicates the total, latent and sensible load at the day of August first for Greensboro and Ney York locations. The total load is higher with the baseline (1200 cfm) when compare it with (1000 and 1400 cfm) for both locations. The sensible load is increased and the latent load is decreased with a higher air flow rate supplied by the system. The reason of increasing in the sensible load can be due to the higher temperature of air supplied to the space with higher cfm.

CHAPTER 5

Discussion and Conclusion

5.1 Discussion

The consumers naturally are deeply concerned about energy costs and thermal comfort in residential homes. The important factor of lowering energy costs leads a number of people to choose a newer air-conditioning system. The increasing of airflow rate supply, reduce the latent cooling heat and increase the cooling sensible heat. Warmer coil surface temperatures will result in lower rates of moisture removal and also shortening the cycle running time by the improved sensible cooling performance.

Although all experiments show that under or overcharging airflow rate cfm supplied, a system can have a significant negative effect on performance. The annual and daily energy consumption in all different locations were increased with higher airflow supplied, and affected the system performance with lower airflow supplied. The major effect of low airflow on system performance seems to be accelerated decrease in evaporating temperatures. The system performance not only affected by different airflow supplied, some other factors can effected on the system performance, such as improper system installation, improper duct design, duct leakage, wrong fan speed selection, use of dirty filters, no maintenance, installation of larger outdoor units without changing the indoor unit, any additional devices which increased system static pressures. Maintenance issues for the components such as filter changes and coil cleaning may be useful to improve the field of system performance.

The thermal load on buildings increases by outdoor temperatures increase, while the efficiency and capacity of air conditioners decrease in increasing outdoor temperatures.

the greatest.

5.2 Summary

As a summary for the Figures and Tables have been done in this study, the annual and hourly energy consumption simulation results are obtained for each HVAC airflow supply rate (cfm) within a single family residential house. The house is using a three ton heat pump system with constant fan speed and constant volume.

The energy use for space cooling/ heating and fan (KWh) as function of supplied air flow rate (cfm) are demonstrated in Figures 30, 31, 32 and 33 for the locations of Miami, Las Vegas, New York and Greensboro, respectively. Table 6 shows the total, sensible and latent loads obtained in Greensboro and Las Vegas. Results for Greensboro and New York are presented in Appendix B. Table 7 show the general data that obtained from the tests and simulation.

Table 7

Daily total/sensible/latent loads in the all locations (KBtu/hr)

Air Flow Rate	cfm	900	1000	1100	1200	1300	1400
Miami- May-15th	Total	407300	406333	405053	403840	402441	400743
	Sensible	315440	314882	316905	320120	323070	325574
	Latent	91450	91450	88148	83720	79370	75169
Las Vegas May-15th	Total	289032	288078	287153	288053	288879	289317
	Sensible	287859	286925	286286	287902	288879	289317
	Latent	1173	1153	866	150	0	0
Greensboro Aug-1st	Total	268534	268313	269680	270455	269066	267206
	Sensible	201503	201073	202645	205070	207172	209001
	Latent	67031	67240	67034	65385	61894	58204
New York Aug-1st	Total	269529	268866	268766	268317	267477	265876
	Sensible	222501	221951	223150	225585	227830	229734
	Latent	47028	46915	45616	42731	39647	36142

5.3 Conclusion

The simulation study was conducted to investigate and evaluate the adequacy of handler supply airflow rate for a single-family residential house by using four different climates zones. The impacts of different supply airflow rate in different weather conditions on HVAC heat pump are analyzed. The locations are selected from the US ASHRAE Climate Zones. Two locations were selected as extreme weather of humid hot in Miami, and dry hot weather in Las Vegas. The other locations were selected as mixed humid moderate in Greensboro and humid cold in New York. The output data presented in this study shows that evaporator airflow supplied was often deficient relative to manufacturer guidelines which typically call for 350 to 450 cfm/ton. **Under or over supplying of airflow rate, a significant effect would occur on performance of the system.** All tests show higher average supply of airflow (cfm) were accompanied with higher electrical consumption. All tests indicated higher airflow rates supplied in all observed locations; this relates to the increase of the following items:

1. The annual and hourly total energy used.
2. The annual and hourly fan energy used.
3. The temperatures of air leaving the cooling coils are affected by the increasing the supply of the airflow across the coils which increases the temperature and the vice versa.
4. Humidity ratio (Lb H_2O / Lb dry air) in the space is affected by higher or lower airflow rate supply and lead to uncomfortable environment, especially in location with a humid weather. However the humidity ratios are increased in all tests in all locations.

The detailed results of other items in the selected locations generally show:

- In Miami, where the hot humid weather, the impact of supply higher air flow rate results show that the cooling energy usage is decreased while both of fan and total energy are increased and no change in heating energy usage regarding the latent heat releases. The tests show in one day on May 15th; that the loads are affected differently with different climates zone. Regarding the weather in Miami, a slight decrease in the total load along with the increasing of the airflow supply was observed. Also the sensible load is increased while degradation in latent load is observed because of the very hot and humid weather.

- The operations times of the system were increasing along with the more amount of airflow rate supplied and it takes relatively longer periods due to the effect of the humid weather in May.
- In Las Vegas, where the hot dry weather, all tests with higher air flow supplied indicates almost the same results as in Miami. The tests were conducted on the same day of May 15th; the cooling energy use is slightly decreased with a relatively constant amount of heating. Slight decrease in the total load when the airflow supply is increased was noticed. Also, the sensible load is increased while degradation in latent load is observed. In Las Vegas because the weather is very hot and dry (a high temperature during the day time and low temperature during night time), the total load is almost not changing or slightly increased, with a very slight increase in sensible load also and nearly there is no latent load almost zero.
- In Greensboro, the daily test conducted on August 1st where the weather is moderate and not so humid, the total load and the sensible load are increased where the latent load is decreased.

- In New York, the daily test conducted on August 1st also where the weather is cold and humid, the total load is decreased, the sensible load is increased and the latent load is decreased.

A comparative observation between the system operations time of the humid hot weather in Miami and moderate weather in Greensboro, with more supply amount of airflow rate (cfm) supplied in both location, there was a big difference in the period. In Miami location it takes a so longer period than in Greensboro location, and that because the more hot humidity in Miami than in Greensboro as shown in Table 27.

The Investigations analyzing of annually energy used in the four locations that been observed are:

- In Miami and Las Vegas, the Space cooling is higher than space heating and the fan energy used is almost same, because the weather is more hot in summer season and almost not so cold in winter season.
- In Greensboro and New York, the heating is higher than space cooling and the fan energy used is almost same, because the weather is not so hot in summer season and almost is so cold in winter season.

The different supply of airflow rate (cfm) is not only having the impact on performance of HVAC heat pump system. There are some other items to improving HVAC system performance is essential to reduce the building's energy use; and this can be done through system design, construction, equipment efficiency, maintenance scheduling, ventilation control strategies and so on, and not only by using the base airflow rate (400 cfm).

5.4 Future Work

- Investigate the effect of different airflow rate supplied on HVAC system performance in commercial building and for different types of HVAC systems.
- Conduct a detailed analysis for the space thermal comfort when supplying different air flow rate.
- Investigate the effect of airflow rate supplied on newest SEER HVAC system performance.
- Air flow rate and supply air temperature with the specific relative humidity can directly affect the space thermal comfort, thus an optimal design can vary based on different climate weather locations.
- Investigate the effect of static pressure, improper duct design and duct leakage on HVAC system performance.
- Discuss in detail the amount of air supplied by each cooling ton for a different climate weather locations, especially for extreme climate weather locations, since the code does not specify that.

References

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Appendix A

Visual Simulations of Total Annual Energy Use

Visual of simulations for Miami - Total Annual Energy with 900 cfm Airflow Supply

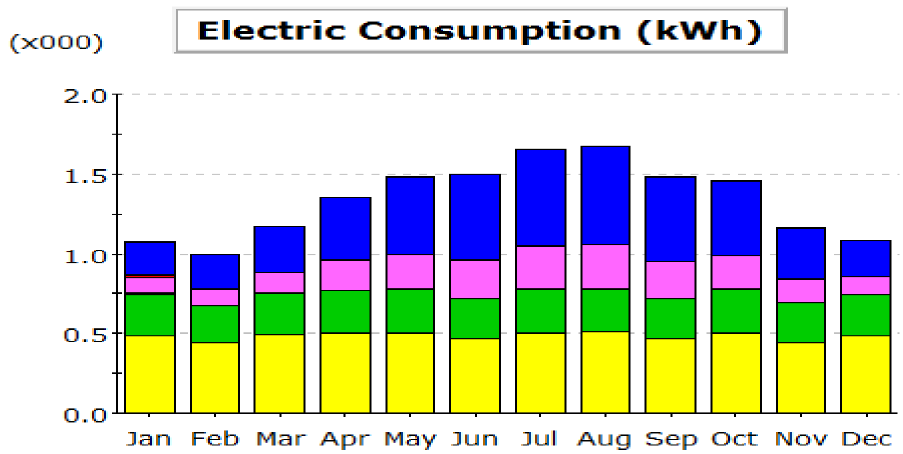


Figure A.1 Annual energy use for airflow supply 900 cfm simulation in Miami

Table A 1

Annual energy use for airflow supply 900 cfm simulation in (Miami)

Month	0.21	0.21	0.28	0.4	0.49	0.54	0.61	0.61	0.53	0.47	0.32	0.23	4.89
Space Cool	0	0	0	0	0	0	0	0	0	0	0	0	0
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0.02	0	0	0	0	0	0	0	0	0	0	0	0.02
Space Heat	0	0	0	0	0	0	0	0	0	0	0	0	0
HP Supp.	0	0	0	0	0	0	0	0	0	0	0	0	0
Hot Water	0.1	0.1	0.13	0.19	0.22	0.24	0.27	0.28	0.24	0.21	0.15	0.11	2.24
Vent. Fans	0	0	0	0	0	0	0	0	0	0	0	0	0
Pumps & Aux.	0	0	0	0	0	0	0	0	0	0	0	0	0
Ext. Usage	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Misc. Equip.	0	0	0	0	0	0	0	0	0	0	0	0	0
Task Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Area Lights	1.08	1	1.17	1.35	1.48	1.5	1.65	1.67	1.48	1.46	1.16	1.09	16.09
Total	0.21	0.21	0.28	0.4	0.49	0.54	0.61	0.61	0.53	0.47	0.32	0.23	4.89

Visual of simulations for Miami - Total Annual Energy with 1000 cfm Airflow Supply

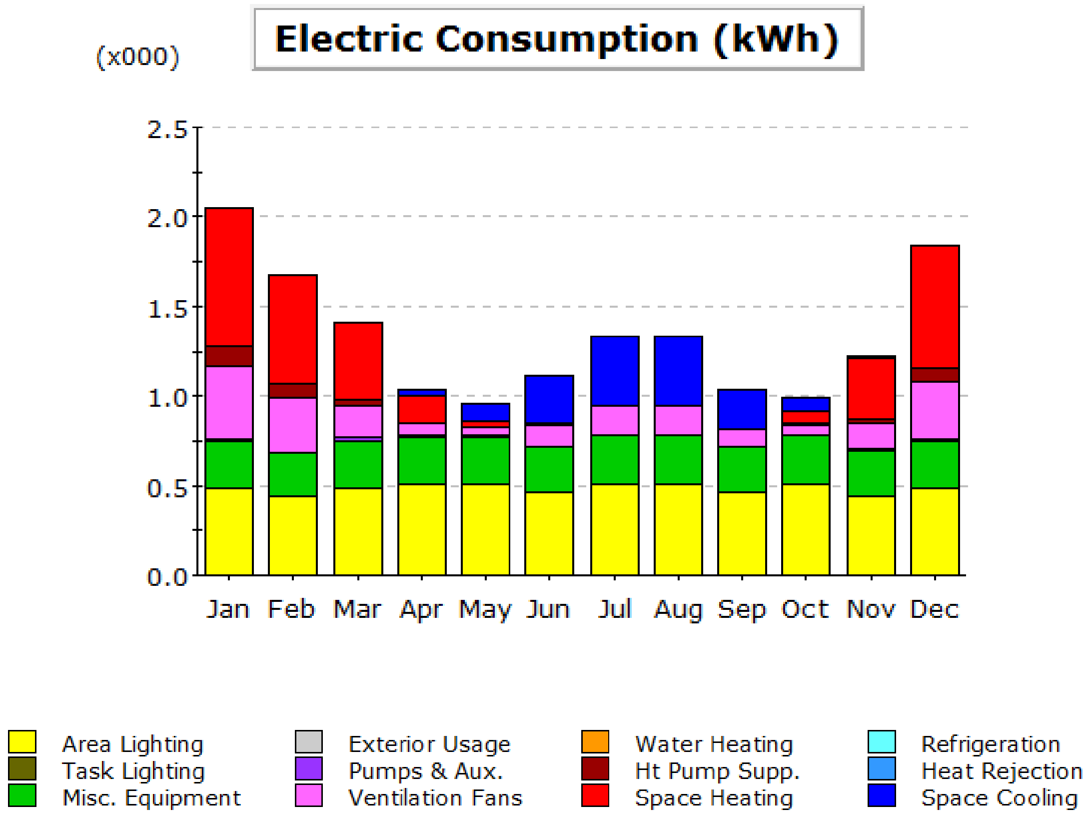


Figure A.2 Annual energy use for airflow supply 1000 cfm simulation in (Miami)

Table A 2

Annual energy use for airflow supply 1000 cfm simulation in (Miami)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.47
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.34	0.68	3.08
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.41	0.3	0.18	0.07	0.05	0.12	0.17	0.17	0.1	0.06	0.14	0.32	2.1
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.08
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.04	1.68	1.41	1.04	0.95	1.12	1.33	1.33	1.03	0.99	1.22	1.83	15.98

Visual of Simulations for Miami - Total Annual Energy with 1100 cfm Airflow Supply

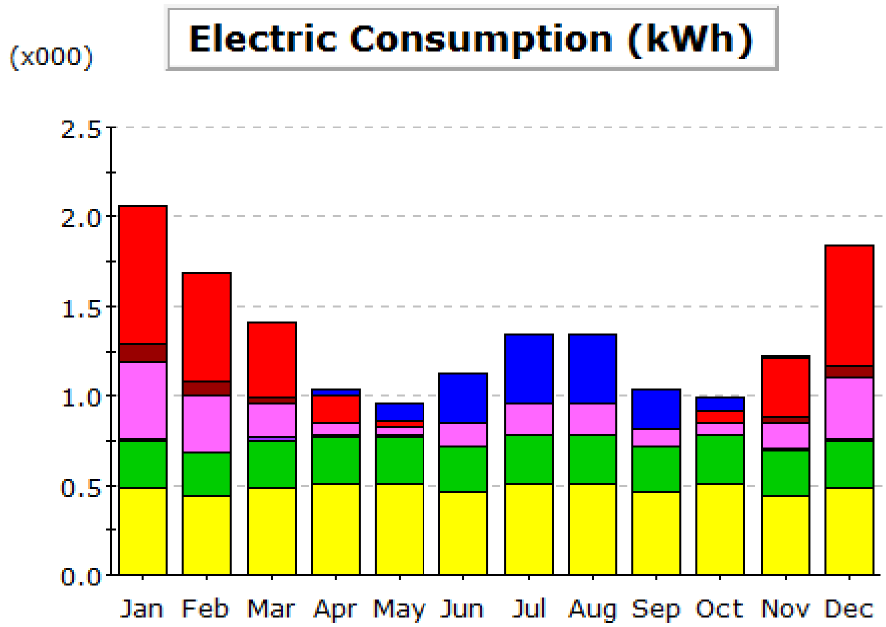


Figure A.3 Annual energy use for airflow supply 1100 cfm simulation in (Miami)

Table A 3

Annual energy use for airflow supply 1100 cfm simulation in (Miami)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.28	0.39	0.38	0.22	0.07	0.01	0	1.48
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.34	0.68	3.07
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.43	0.32	0.19	0.07	0.05	0.13	0.18	0.18	0.1	0.06	0.15	0.33	2.16
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.08
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.06	1.69	1.41	1.04	0.95	1.12	1.34	1.34	1.03	0.99	1.22	1.84	16.05

Visual of Simulations for Miami - Total Annual Energy with 1200 cfm Airflow Supply

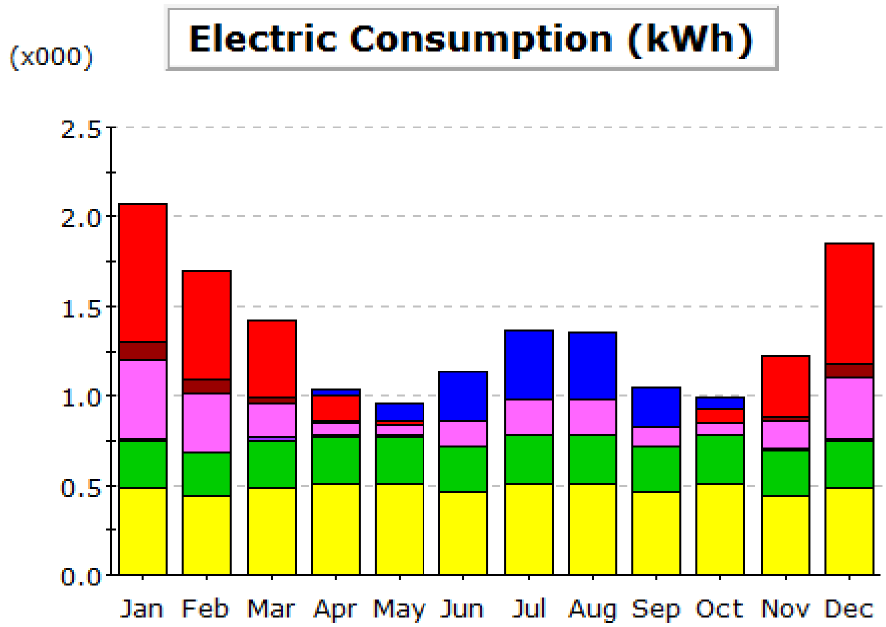


Figure A.4 Annual energy use for airflow supply 1200 cfm simulation in (Miami)

Table A 4

Annual energy use for airflow supply 1200 cfm simulation in (Miami)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.47
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.32	0.19	0.07	0.05	0.14	0.2	0.2	0.11	0.06	0.15	0.34	2.27
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.69	1.42	1.04	0.96	1.14	1.36	1.36	1.04	0.99	1.22	1.85	16.15

Visual of Simulations for Miami - Total Annual Energy with 1300 cfm Airflow Supply

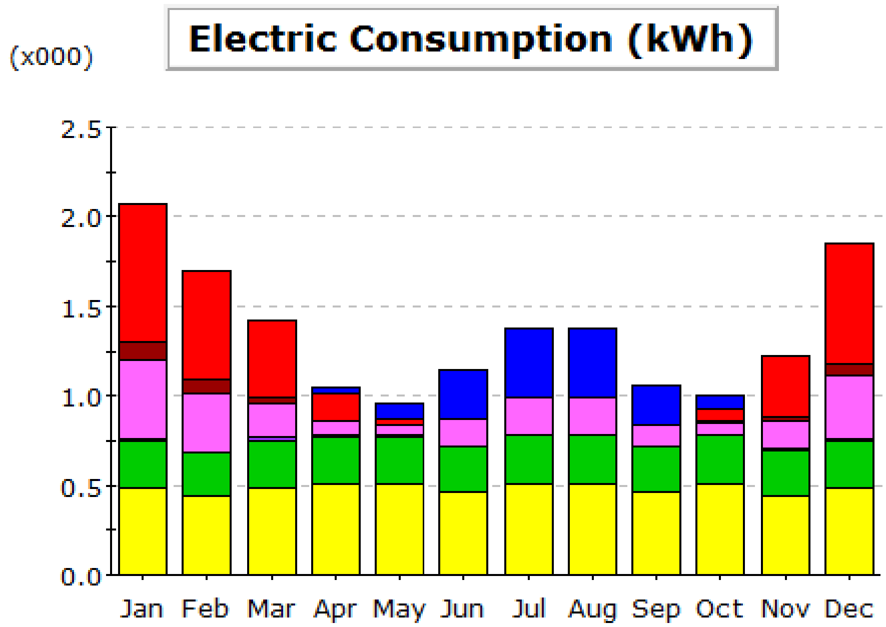


Figure A.5 Annual energy use for airflow supply 1300 cfm simulation in (Miami)

Table A 5

Annual energy use for airflow supply 1300 cfm simulation in (Miami)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.46
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.33	0.19	0.07	0.06	0.15	0.22	0.22	0.12	0.07	0.15	0.34	2.37
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.7	1.42	1.04	0.96	1.15	1.38	1.38	1.05	1	1.23	1.85	16.22

Visual of simulations for Miami - Total Annual Energy with 1400 cfm Airflow Supply

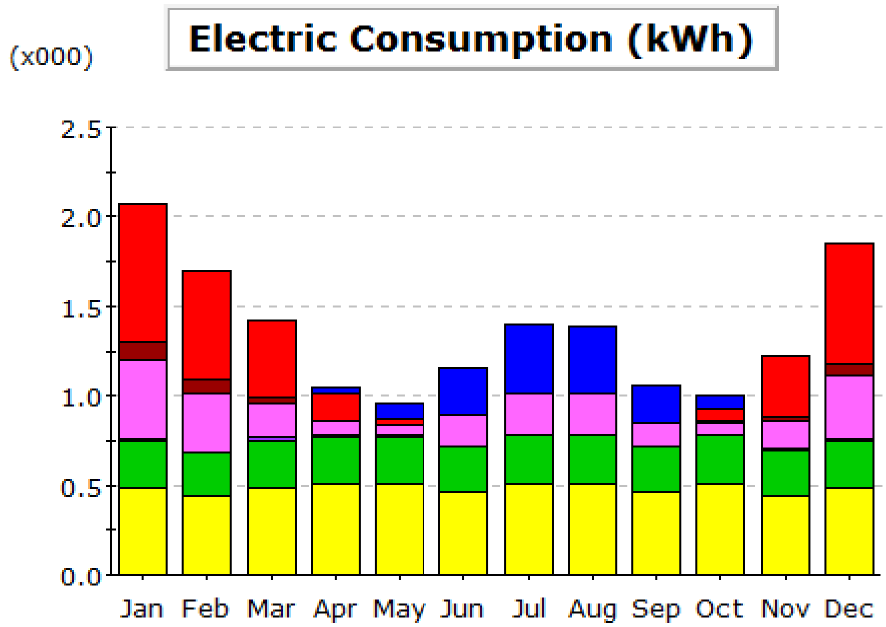


Figure A.6 Annual energy use for airflow supply 1400 cfm simulation in (Miami)

Table A 6

Annual energy use for airflow supply 1400 cfm simulation in (Miami)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.38	0.38	0.21	0.07	0.01	0	1.45
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.33	0.19	0.07	0.06	0.17	0.24	0.23	0.13	0.07	0.15	0.34	2.43
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.7	1.42	1.04	0.96	1.16	1.4	1.39	1.06	1	1.23	1.85	16.28

Visual of simulations for Las Vegas - Total Annual Energy with 900 cfm Airflow Supply

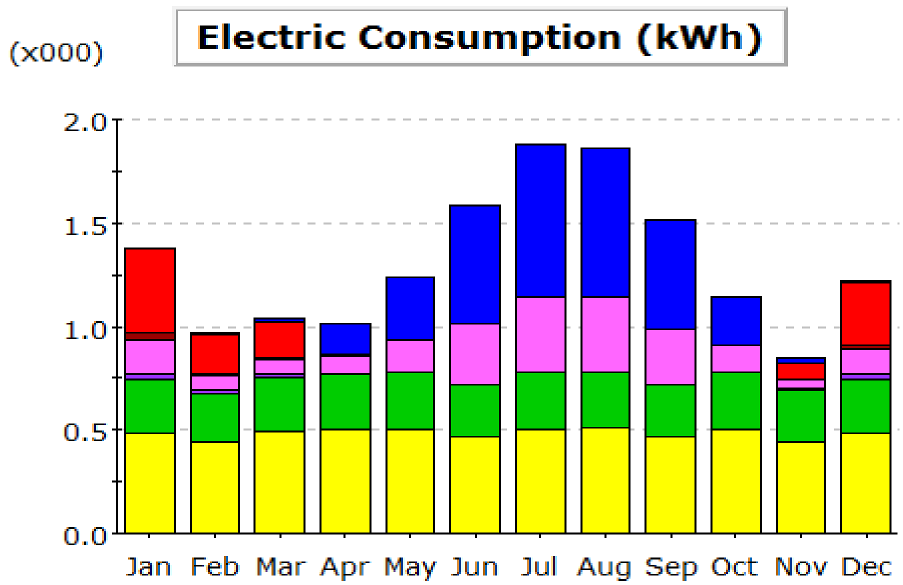


Figure A.7 Annual energy use for airflow supply 900 cfm simulation in Las Vegas

Table A 7

Annual energy use for airflow supply 900 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.3	0.58	0.74	0.72	0.53	0.23	0.03	0	3.3
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.41	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.17
HP Supp.	0.04	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.17	0.07	0.07	0.08	0.16	0.29	0.36	0.36	0.27	0.13	0.04	0.12	2.12
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.38	0.97	1.04	1.01	1.24	1.59	1.88	1.86	1.52	1.14	0.85	1.22	15.69

Visual of simulations for Las Vegas - Total Annual Energy with 1000 cfm Airflow Supply

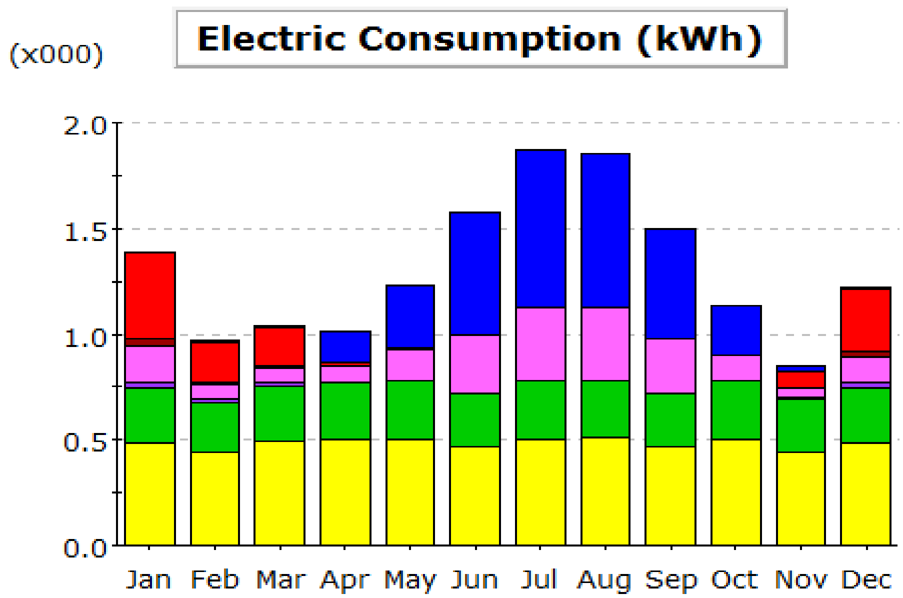


Figure A.8 Annual energy use for airflow supply 1000 cfm simulation in Las Vegas

Table A 8

Annual energy use for airflow supply 1000 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.3	0.58	0.74	0.72	0.52	0.23	0.03	0	3.3
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.41	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.17
HP Supp.	0.03	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.18	0.07	0.07	0.08	0.15	0.28	0.35	0.35	0.26	0.12	0.04	0.12	2.08
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.39	0.97	1.04	1.01	1.23	1.58	1.87	1.85	1.5	1.13	0.85	1.22	15.63

Visual of simulations for Las Vegas - Total Annual Energy with 1100 cfm Airflow Supply

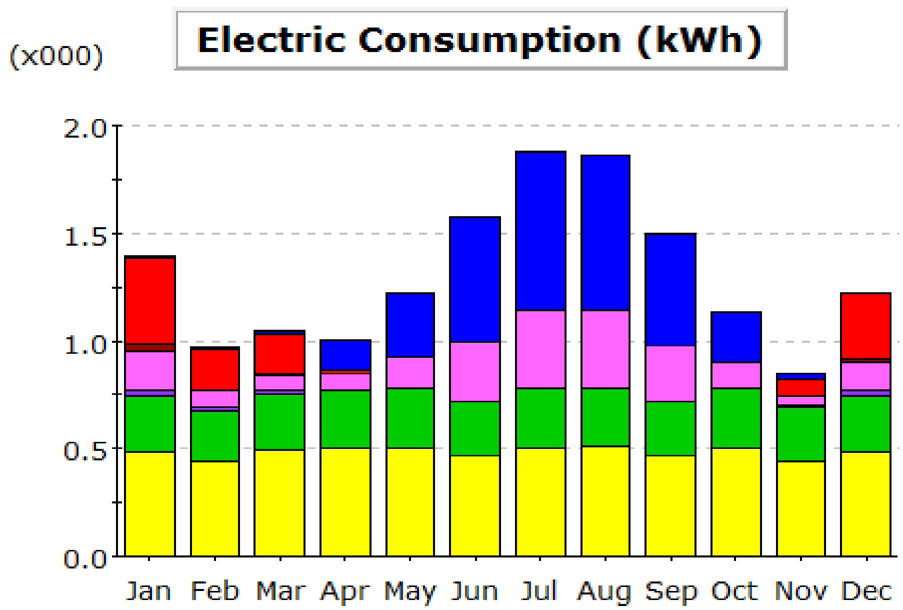


Figure A.9 Annual energy use for airflow supply 1100 cfm simulation in Las Vegas

Table A9

Annual energy use for airflow supply 1100 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.29	0.58	0.73	0.72	0.52	0.23	0.03	0	3.28
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.4	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.17
HP Supp.	0.03	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.18	0.08	0.08	0.08	0.15	0.28	0.37	0.36	0.26	0.12	0.04	0.13	2.12
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.39	0.97	1.04	1.01	1.22	1.57	1.88	1.86	1.5	1.13	0.85	1.22	15.65

Visual of simulations for Las Vegas - Total Annual Energy with 1200 cfm Airflow Supply

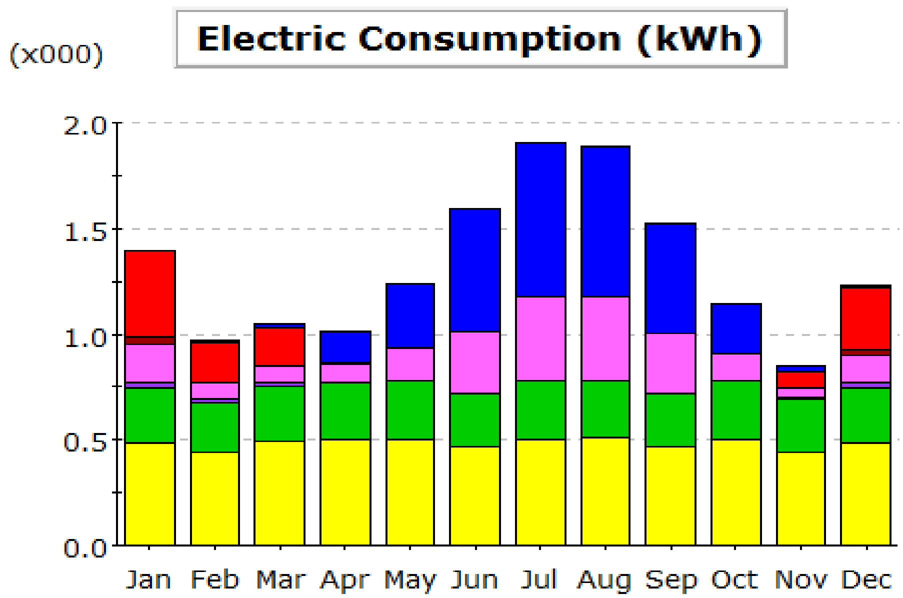


Figure A.10 Annual energy use for airflow supply1200 cfm simulation in Las Vegas

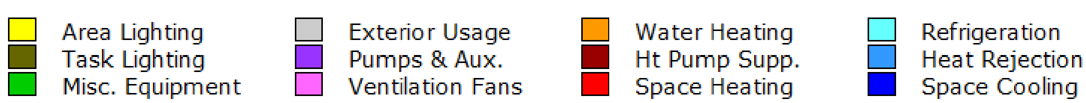


Table A 10

Annual energy use for airflow supply 1200 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.3	0.58	0.73	0.71	0.52	0.23	0.03	0	3.27
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.4	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.16
HP Supp.	0.03	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.19	0.08	0.08	0.08	0.16	0.3	0.4	0.4	0.28	0.13	0.04	0.13	2.26
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.39	0.97	1.05	1.01	1.23	1.59	1.9	1.89	1.52	1.14	0.85	1.23	15.79

Visual of simulations for Las Vegas - Total Annual Energy with 1300 cfm Airflow Supply

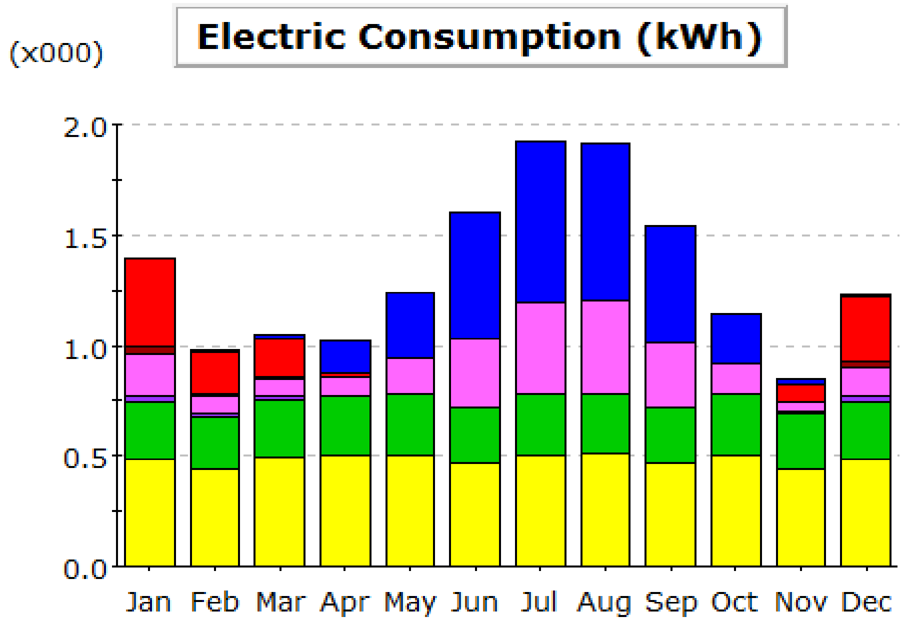


Figure A.11 Annual energy use for airflow supply1300 cfm simulation in Las Vegas

Table A 11

Annual energy use for airflow supply 1300 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.3	0.58	0.73	0.71	0.52	0.23	0.03	0	3.27
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.4	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.16
HP Supp.	0.03	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.19	0.08	0.08	0.09	0.17	0.31	0.42	0.42	0.3	0.14	0.04	0.13	2.36
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.4	0.97	1.05	1.02	1.24	1.6	1.92	1.91	1.54	1.15	0.85	1.23	15.88

Visual of simulations for Las Vegas - Total Annual Energy with 1400 cfm Airflow Supply

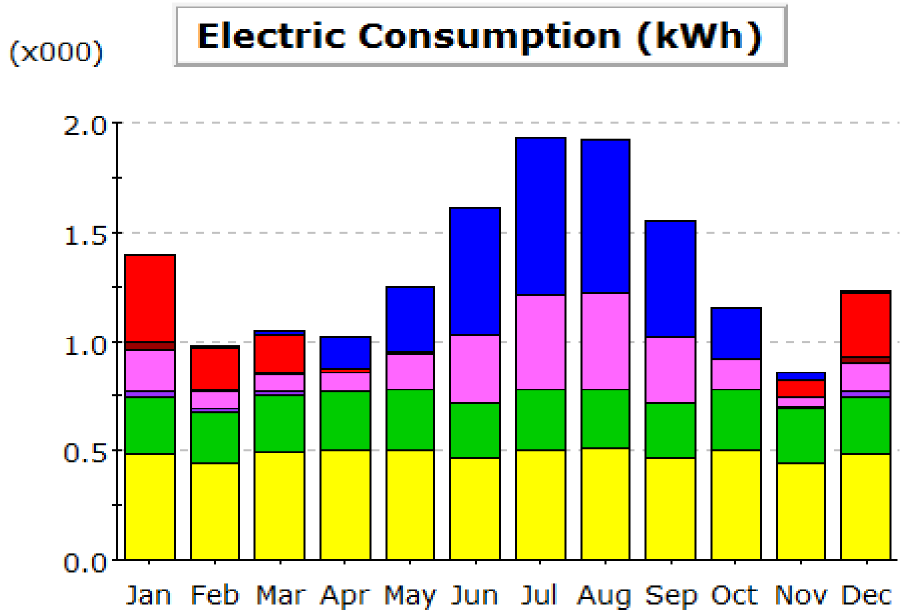


Figure A.12 Annual energy use for airflow supply1400 cfm simulation in Las Vegas

Table A 12

Annual energy use for airflow supply 1400 cfm simulation in (Las Vegas)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0.01	0.02	0.14	0.3	0.58	0.72	0.71	0.52	0.23	0.03	0	3.26
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.4	0.19	0.18	0.01	0	0	0	0	0	0	0.08	0.3	1.16
HP Supp.	0.03	0.01	0.01	0	0	0	0	0	0	0	0	0.02	0.07
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.19	0.08	0.08	0.09	0.17	0.31	0.43	0.44	0.31	0.14	0.04	0.13	2.41
Pumps & Aux.	0.02	0.01	0.01	0	0	0	0	0	0	0	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	1.4	0.97	1.05	1.02	1.25	1.61	1.93	1.92	1.55	1.15	0.85	1.23	15.92

Visual of simulations for New York - Total Annual Energy with 900 cfm Airflow Supply

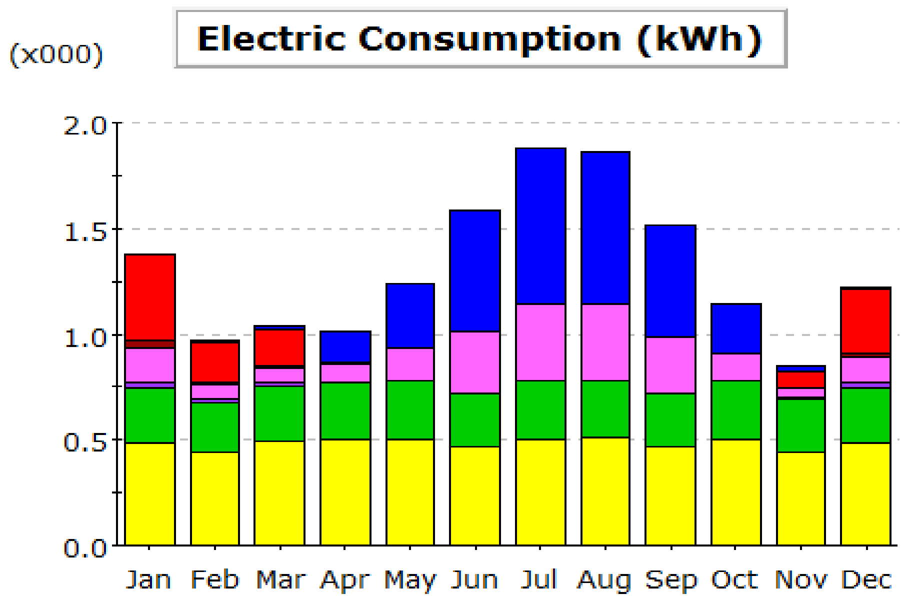


Figure A.13 Annual energy use for airflow supply 900 cfm simulation in New York

Table A13

Annual energy use for airflow supply 900 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.35	0.3	0.17	0.02	0	0	1.08
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.94	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.8	4.32
HP Supp.	0.14	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.45
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.49	0.39	0.3	0.14	0.05	0.09	0.16	0.14	0.08	0.04	0.18	0.37	2.45
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.32	2	1.85	1.34	0.97	1	1.29	1.22	0.97	0.95	1.39	2.03	17.33

Visual of simulations for New York - Total Annual Energy with 1000 cfm Airflow Supply

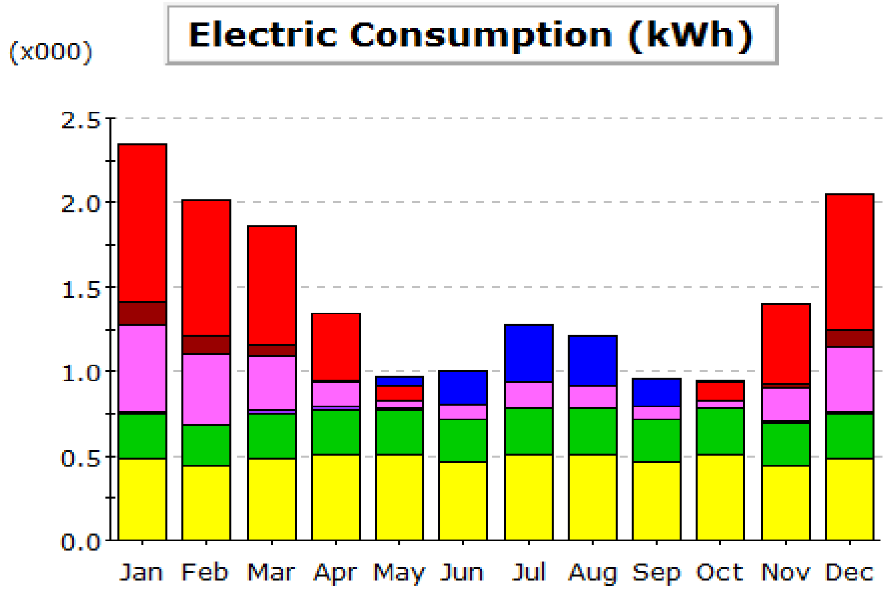


Figure A.14 Annual energy use for airflow supply 1000 cfm simulation in New York

Table A 14

Annual energy use for airflow supply 1000 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.35	0.3	0.17	0.02	0	0	1.07
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.93	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.8	4.3
HP Supp.	0.13	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.45
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.52	0.41	0.32	0.15	0.05	0.09	0.16	0.14	0.08	0.04	0.19	0.39	2.53
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.34	2.02	1.86	1.34	0.96	1	1.28	1.22	0.96	0.95	1.4	2.04	17.38

Visual of simulations for New York - Total Annual Energy with 1100 cfm Airflow Supply

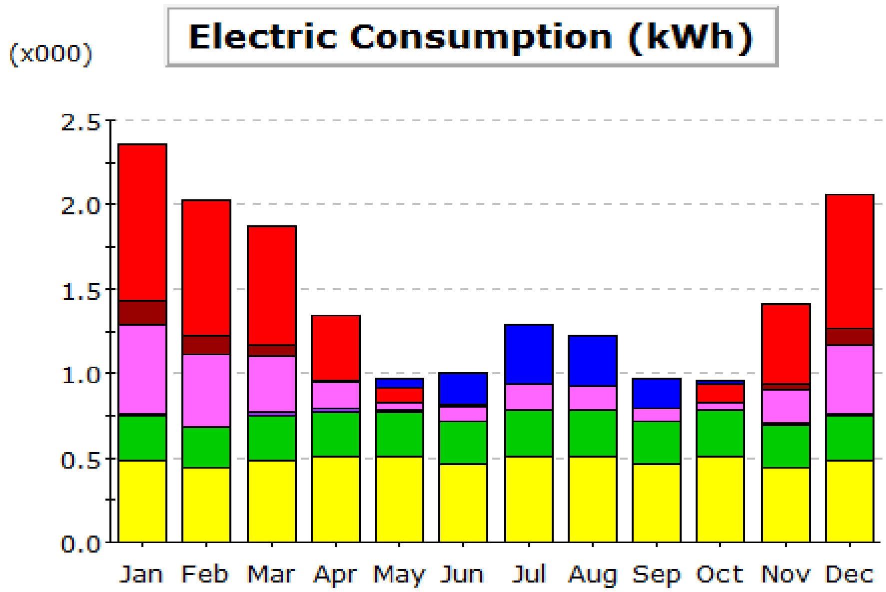


Figure A.15 Annual energy use for airflow supply 1100 cfm simulation in New York

Table A 15

Annual energy use for airflow supply 1100 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.35	0.3	0.17	0.02	0	0	1.07
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.93	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.79	4.29
HP Supp.	0.13	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.45
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.54	0.43	0.33	0.16	0.05	0.09	0.16	0.14	0.08	0.04	0.2	0.4	2.62
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.36	2.03	1.87	1.35	0.96	1	1.29	1.22	0.97	0.95	1.4	2.05	17.45

Visual of simulations for New Your - Total Annual Energy with 1200 cfm Airflow Supply

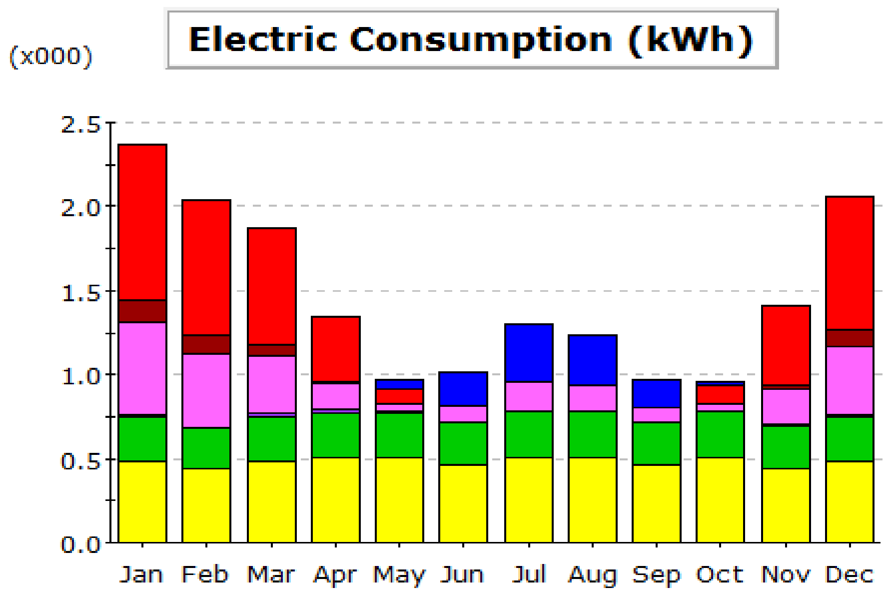


Figure A.16 Annual energy use for airflow supply 1200 cfm simulation in New York

Table A 16

Annual energy use for airflow supply 1200 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.34	0.3	0.17	0.02	0	0	1.07
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.93	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.79	4.28
HP Supp.	0.13	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.45
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.55	0.44	0.34	0.16	0.05	0.1	0.18	0.16	0.09	0.04	0.2	0.41	2.72
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.37	2.04	1.87	1.35	0.97	1.01	1.3	1.24	0.97	0.95	1.41	2.06	17.55

Visual of simulations for New York - Total Annual Energy with 1300 cfm Airflow Supply

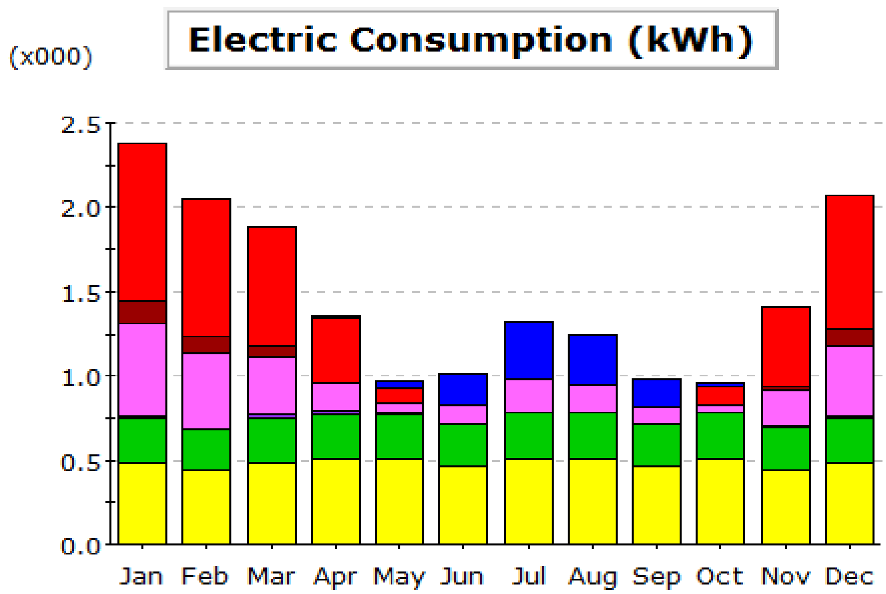


Figure A.17 Annual energy use for airflow supply 1300 cfm simulation in New York

Table A 17

Annual energy use for airflow supply 1300 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.34	0.3	0.17	0.01	0	0	1.06
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.93	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.79	4.28
HP Supp.	0.13	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.44
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.56	0.44	0.34	0.16	0.05	0.11	0.2	0.17	0.1	0.04	0.21	0.42	2.8
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.38	2.04	1.88	1.35	0.97	1.02	1.32	1.25	0.98	0.96	1.41	2.07	17.61

Visual of simulations for New York - Total Annual Energy with 1400 cfm Airflow Supply

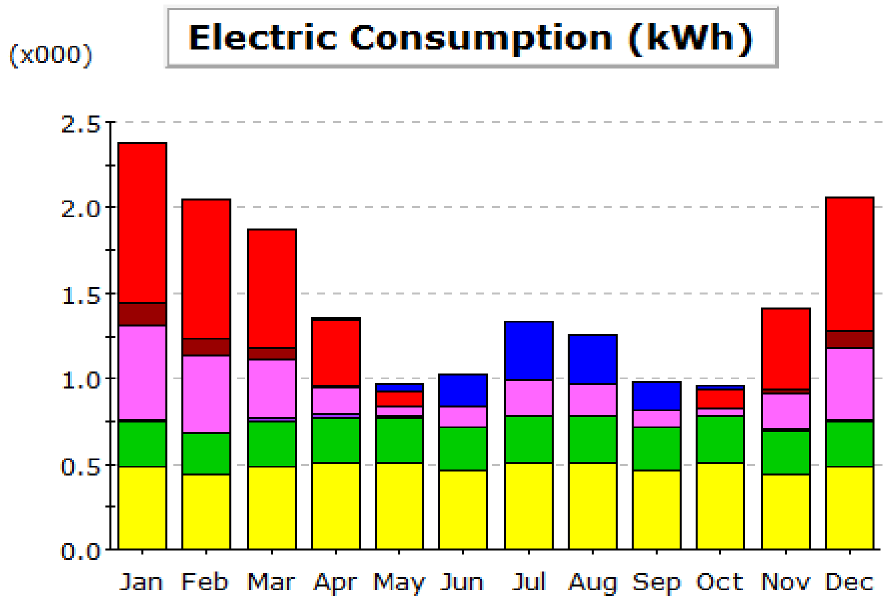


Figure A.18 Annual energy use for airflow supply 1400 cfm simulation in New York

Table A 18

Annual energy use for airflow supply 1400 cfm simulation in (New York)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0	0.05	0.19	0.34	0.29	0.17	0.01	0	0	1.05
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.93	0.81	0.7	0.39	0.09	0	0	0	0	0.11	0.47	0.79	4.28
HP Supp.	0.13	0.11	0.07	0.01	0	0	0	0	0	0	0.03	0.1	0.45
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.56	0.44	0.34	0.16	0.06	0.12	0.21	0.19	0.1	0.04	0.21	0.41	2.84
Pumps & Aux.	0.01	0.01	0.02	0.02	0	0	0	0	0	0.01	0.02	0.01	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.37	2.04	1.88	1.35	0.97	1.02	1.33	1.26	0.99	0.96	1.41	2.06	17.64

Visual of simulations for Greensboro - Total Annual Energy with 900 cfm Airflow Supply

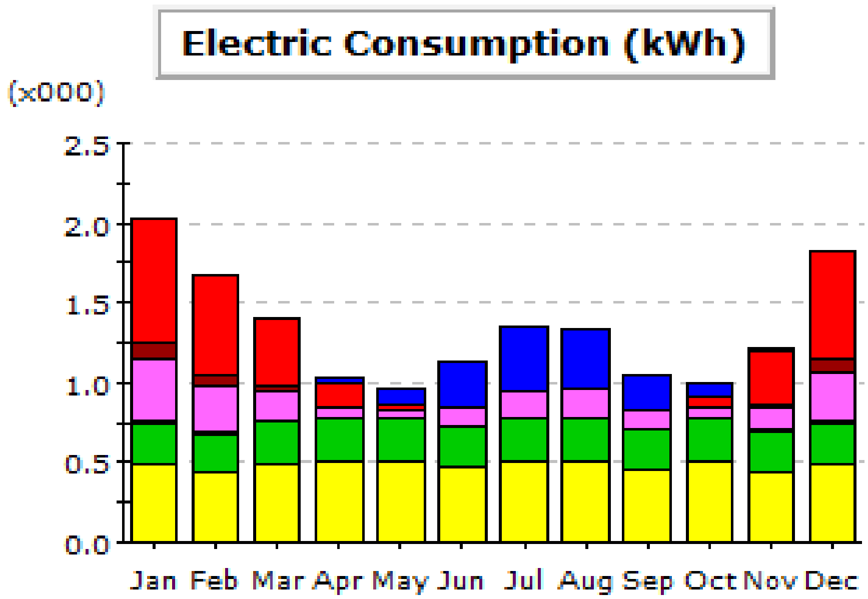


Figure A.19 Annual energy use for airflow supply 900 cfm simulation in Greensboro

Table A 19

Annual energy use for airflow supply 900 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.47
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.34	0.68	3.08
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.41	0.3	0.18	0.07	0.05	0.12	0.17	0.17	0.1	0.06	0.14	0.32	2.1
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.08
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.04	1.68	1.41	1.04	0.95	1.12	1.33	1.33	1.03	0.99	1.22	1.83	15.98

Visual of simulations for Greensboro - Total Annual Energy with 1000 cfm Airflow Supply

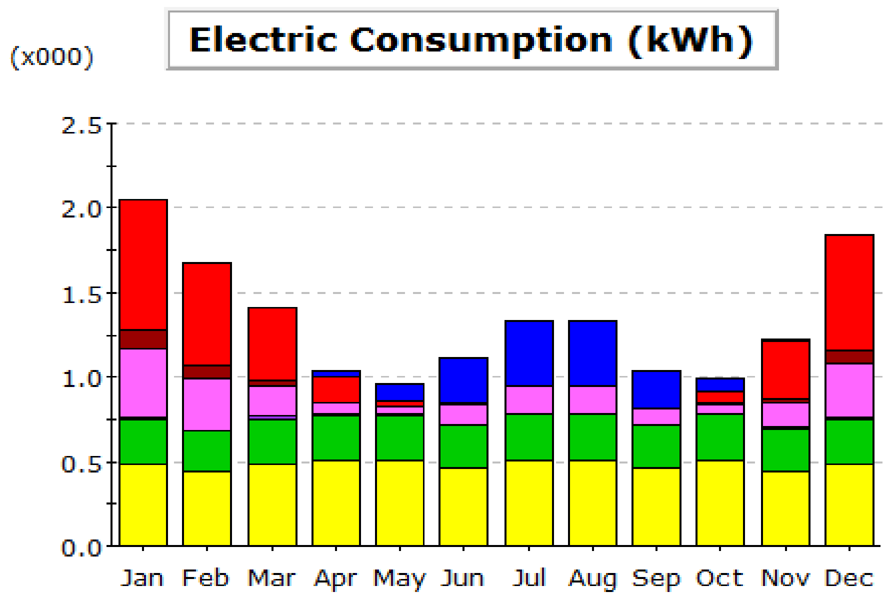


Figure A.20 Annual energy use for airflow supply 1000 cfm simulation in Greensboro

Table A 20

Annual energy use for airflow supply 1000 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.47
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.34	0.68	3.08
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.41	0.3	0.18	0.07	0.05	0.12	0.17	0.17	0.1	0.06	0.14	0.32	2.1
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.08
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.04	1.68	1.41	1.04	0.95	1.12	1.33	1.33	1.03	0.99	1.22	1.83	15.98

Visual of simulations for Greensboro - Total Annual Energy with 1100 cfm Airflow Supply

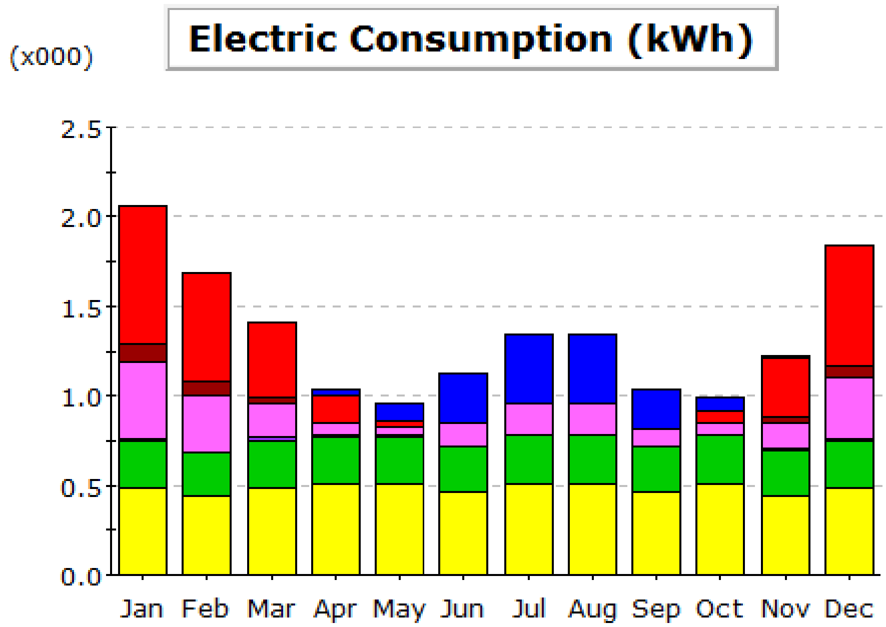


Figure A.21 Annual energy use for airflow supply 1100 cfm simulation in Greensboro

Table A 21

Annual energy use for airflow supply 1100 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.28	0.39	0.38	0.22	0.07	0.01	0	1.48
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.34	0.68	3.07
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.43	0.32	0.19	0.07	0.05	0.13	0.18	0.18	0.1	0.06	0.15	0.33	2.16
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.08
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.06	1.69	1.41	1.04	0.95	1.12	1.34	1.34	1.03	0.99	1.22	1.84	16.05

Visual of simulations for Greensboro - Total Annual Energy with 1200 cfm Airflow Supply

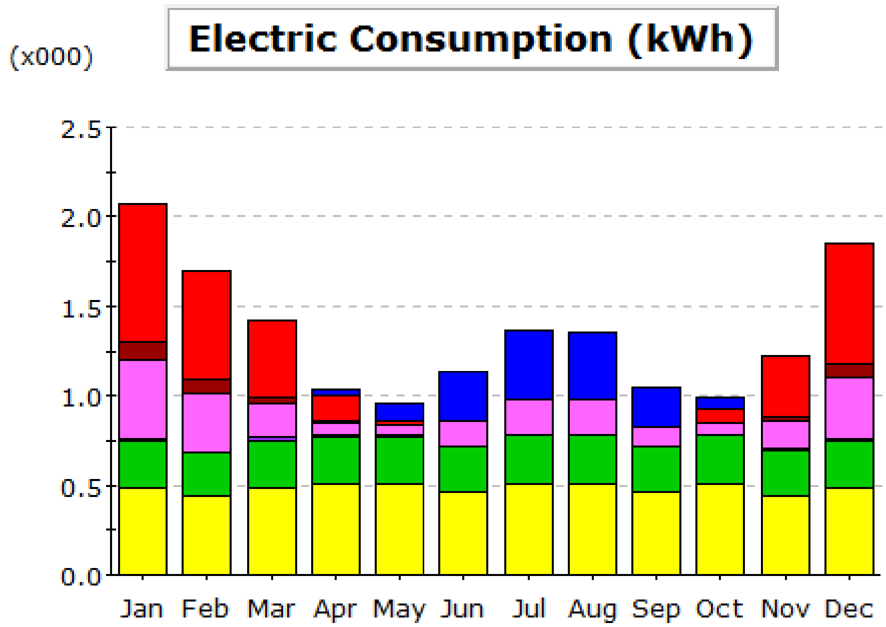


Figure A.22 Annual energy use for airflow supply 1200 cfm simulation in Greensboro

Table A 22

Annual energy use for airflow supply 1200 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.47
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.32	0.19	0.07	0.05	0.14	0.2	0.2	0.11	0.06	0.15	0.34	2.27
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.69	1.42	1.04	0.96	1.14	1.36	1.36	1.04	0.99	1.22	1.85	16.15

Visual of simulations for Greensboro - Total Annual Energy with 1300 cfm Airflow Supply

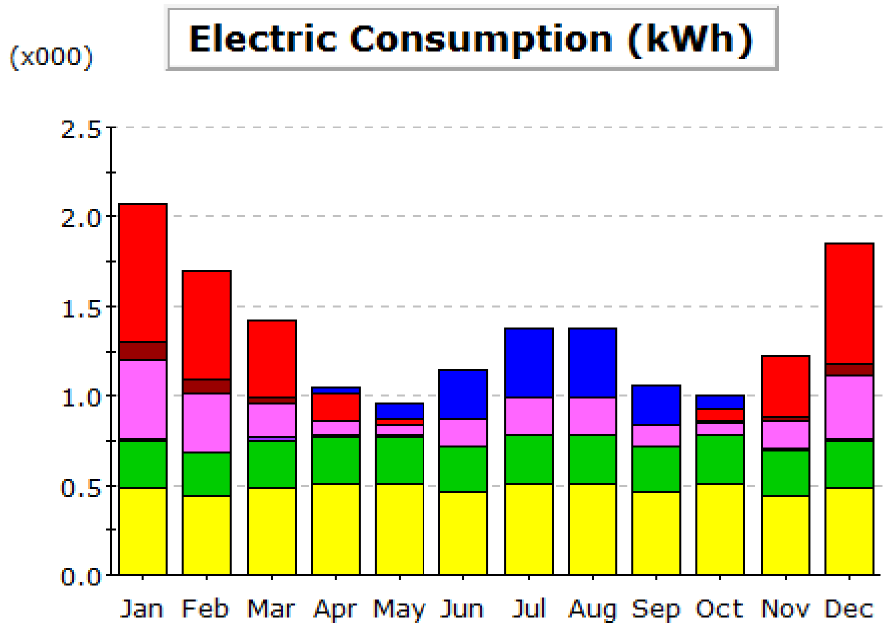


Figure A.23 Annual energy use for airflow supply 1300 cfm simulation in Greensboro

Table A 23

Annual energy use for airflow supply 1300 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.39	0.38	0.22	0.07	0.01	0	1.46
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.33	0.19	0.07	0.06	0.15	0.22	0.22	0.12	0.07	0.15	0.34	2.37
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.7	1.42	1.04	0.96	1.15	1.38	1.38	1.05	1	1.23	1.85	16.22

Visual of simulations for Greensboro - Total Annual Energy with 1400 cfm Airflow Supply

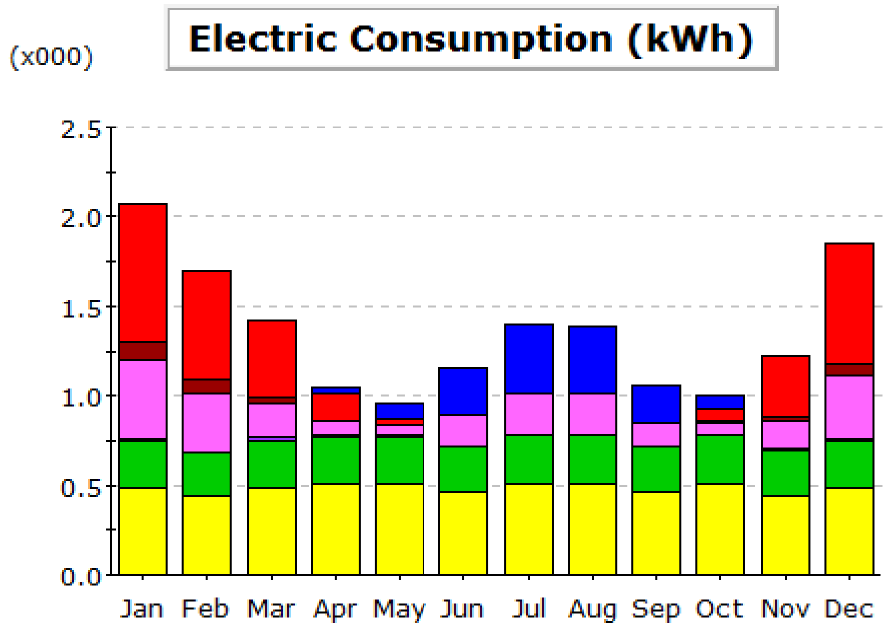


Figure A.24 Annual energy use for airflow supply 1400 cfm simulation in Greensboro

Table A 24

Annual energy use for airflow supply 1400 cfm simulation in (Greensboro)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0	0	0	0.03	0.09	0.27	0.38	0.38	0.21	0.07	0.01	0	1.45
Heat Reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space Heat	0.77	0.61	0.43	0.15	0.03	0	0	0	0	0.07	0.33	0.67	3.06
HP Supp.	0.1	0.07	0.03	0	0	0	0	0	0	0	0.03	0.07	0.32
Hot Water	0	0	0	0	0	0	0	0	0	0	0	0	0
Vent. Fans	0.44	0.33	0.19	0.07	0.06	0.17	0.24	0.23	0.13	0.07	0.15	0.34	2.43
Pumps & Aux.	0.01	0.01	0.01	0.01	0	0	0	0	0	0.01	0.01	0.02	0.09
Ext. Usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. Equip.	0.26	0.24	0.26	0.27	0.27	0.25	0.27	0.27	0.25	0.27	0.24	0.26	3.12
Task Lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area Lights	0.49	0.44	0.49	0.5	0.51	0.47	0.51	0.51	0.46	0.51	0.45	0.49	5.81
Total	2.07	1.7	1.42	1.04	0.96	1.16	1.4	1.39	1.06	1	1.23	1.85	16.28

Appendix B

Simulation Results for New York and Greensboro

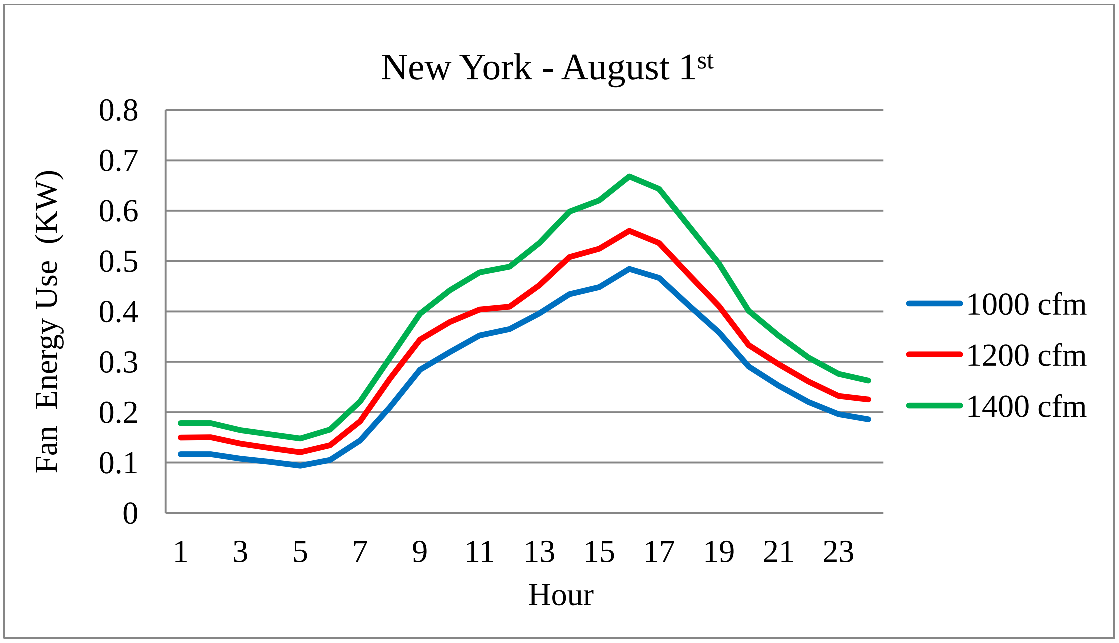


Figure B.1 The hourly fan energy use for the day on August 1st in (New York)

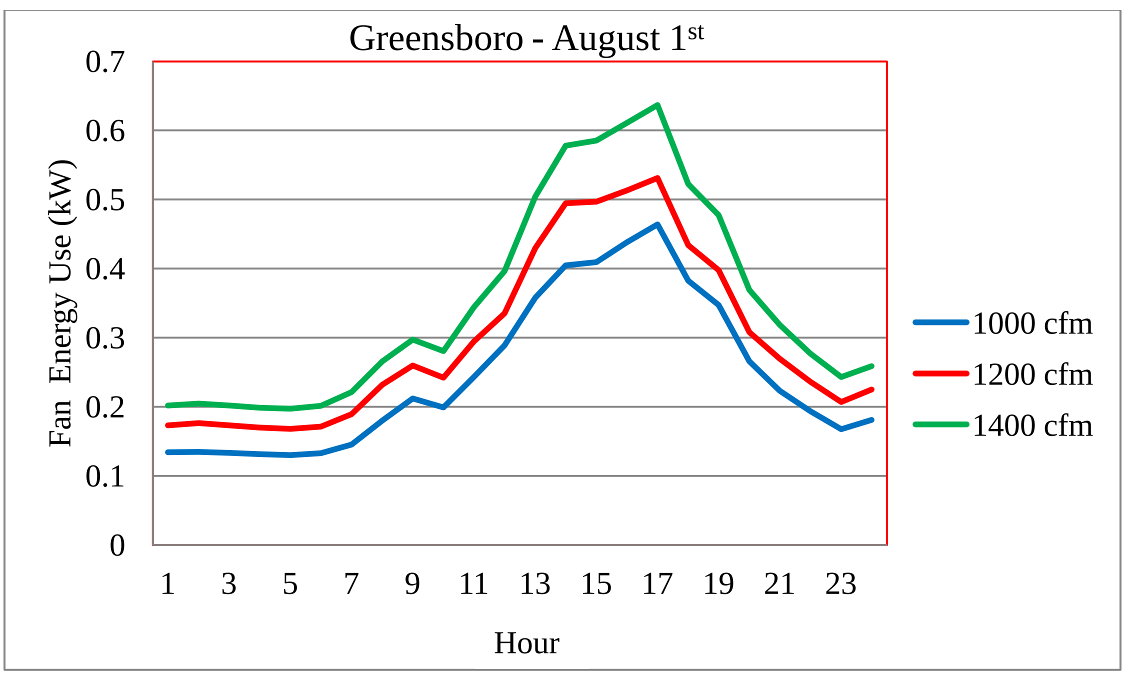


Figure B.2 The hourly fan energy use for the day on August 1st in (Greensboro)

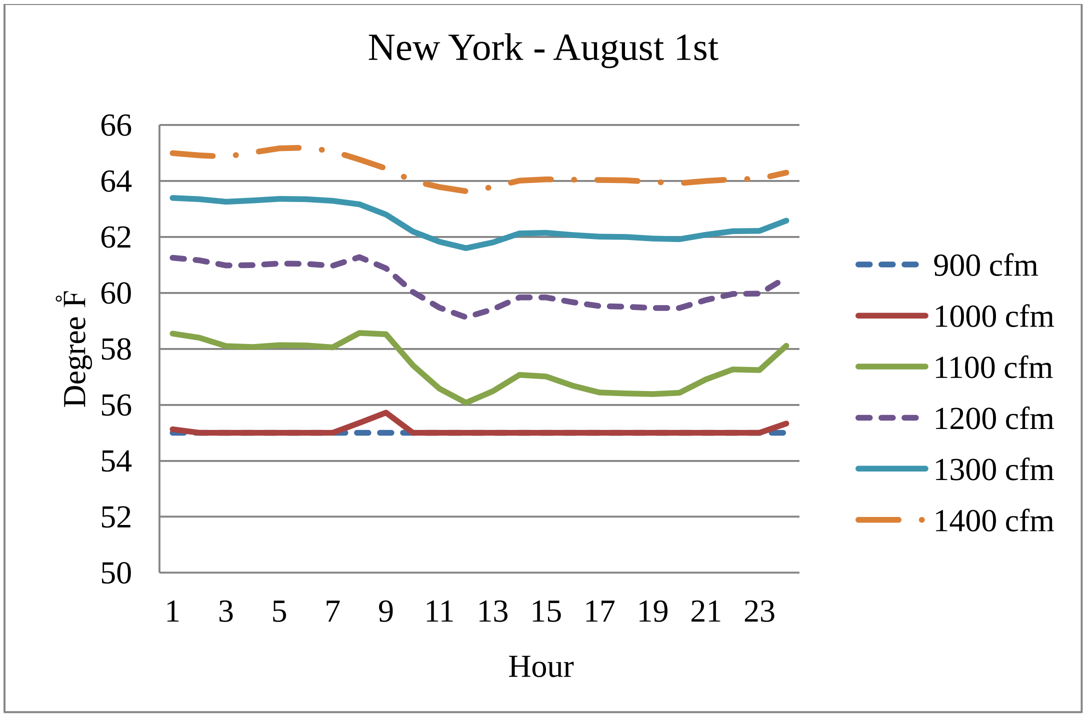


Figure B .3 Temperatures of Air Leaving cooling coil on August 1st in (New York)

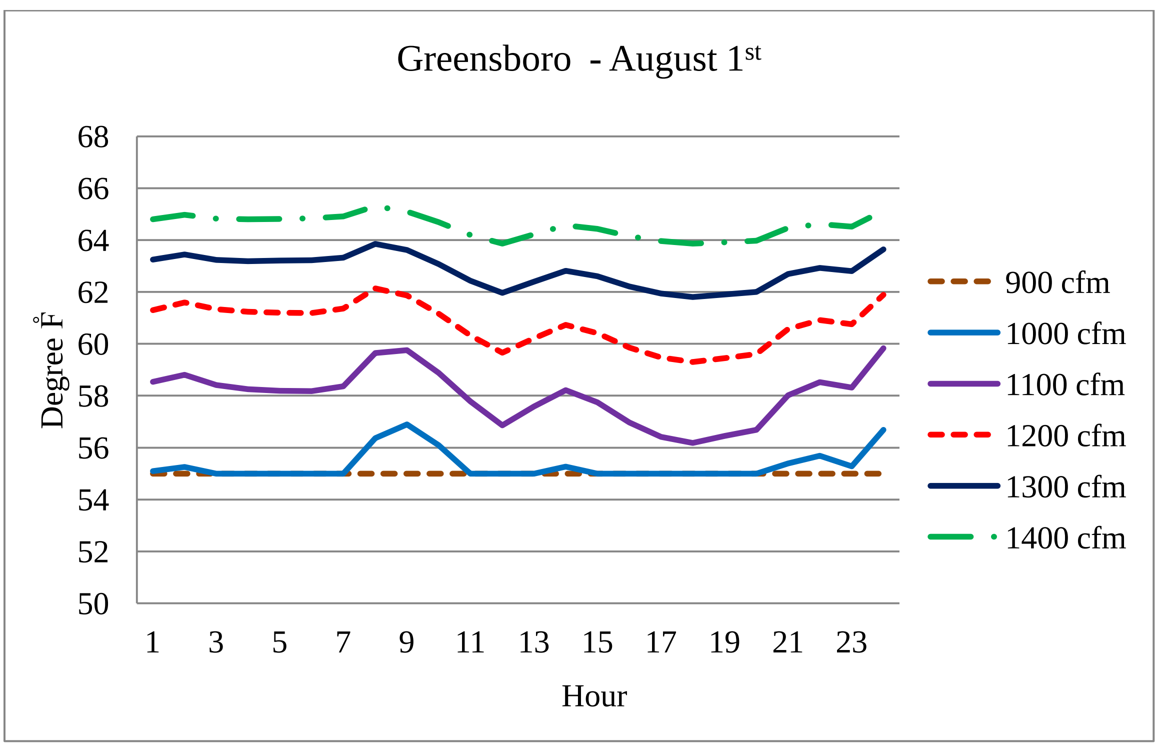


Figure B.4 Temperatures of Air Leaving cooling coil on August 1st in (Greensboro)

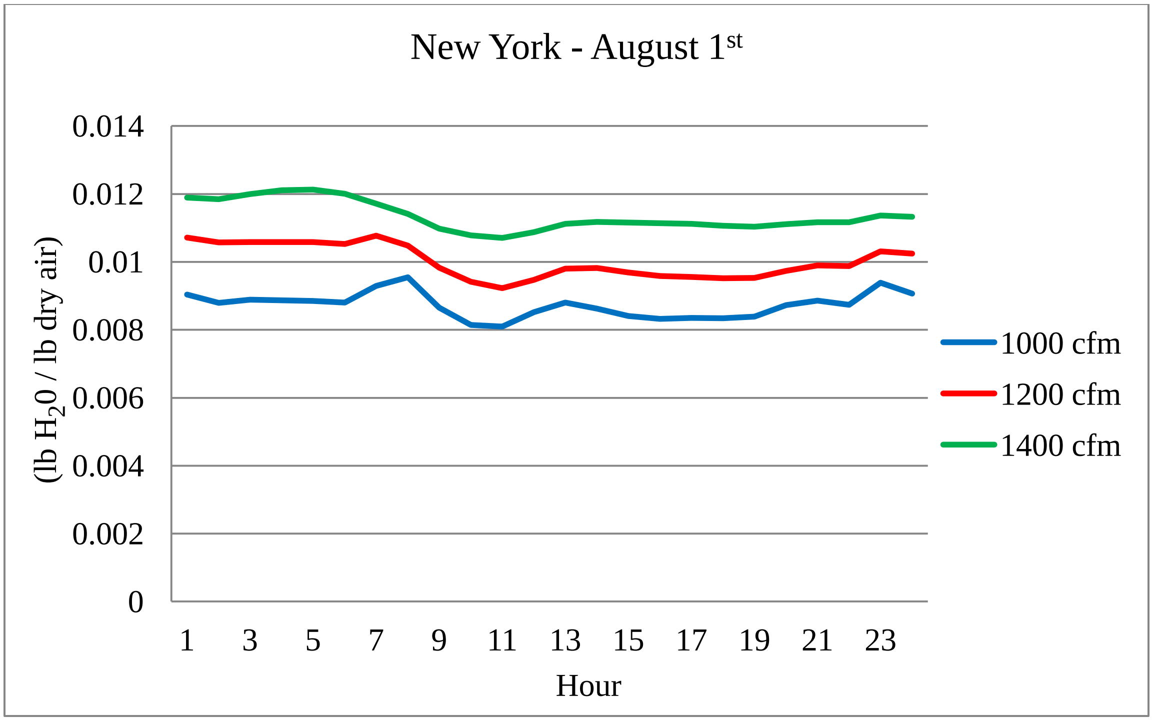


Figure B.5 Return air humidity ratio (lb H_2O / lb dry air on August 1st in (New York)

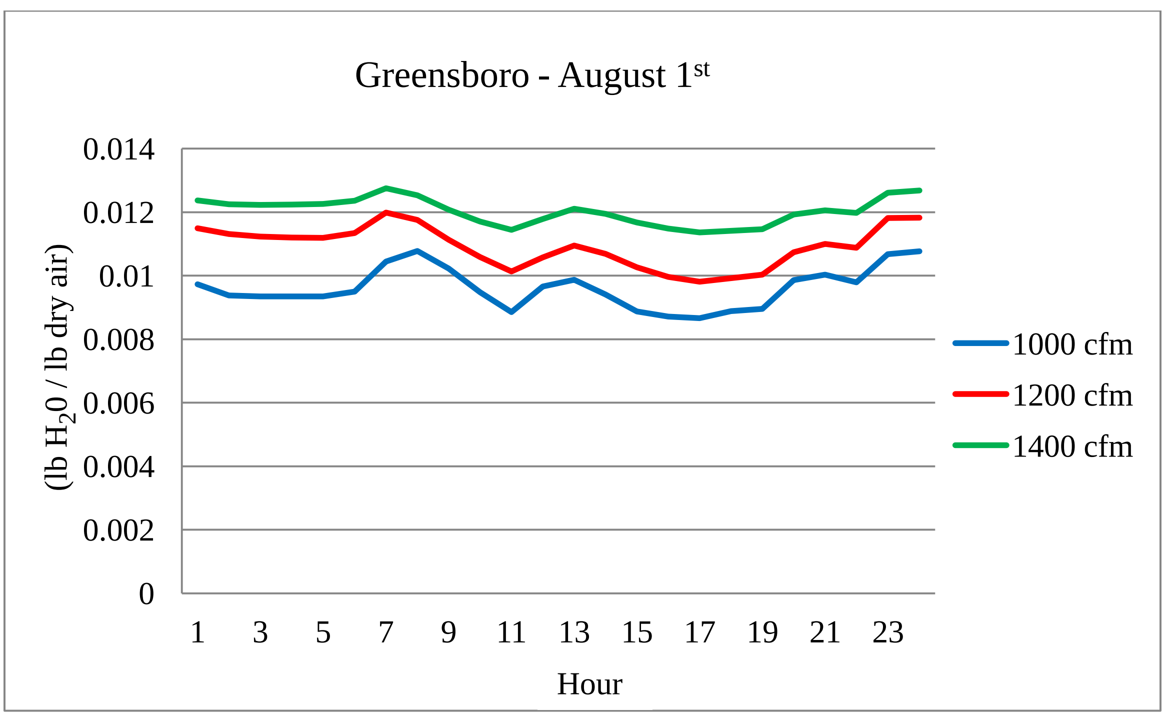
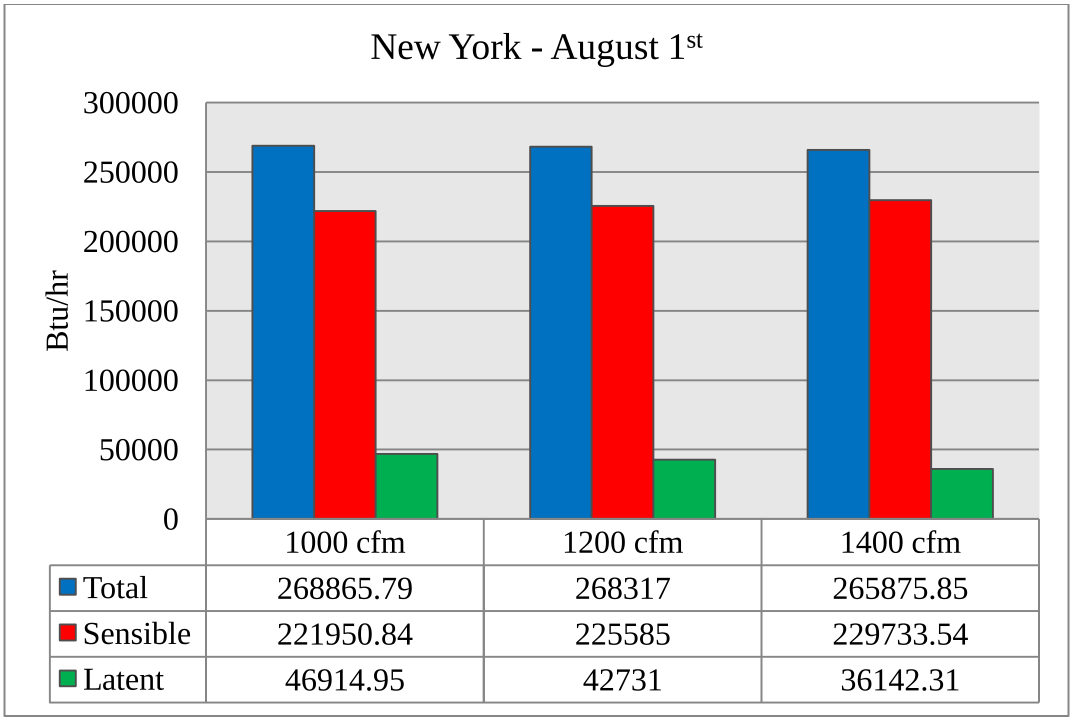
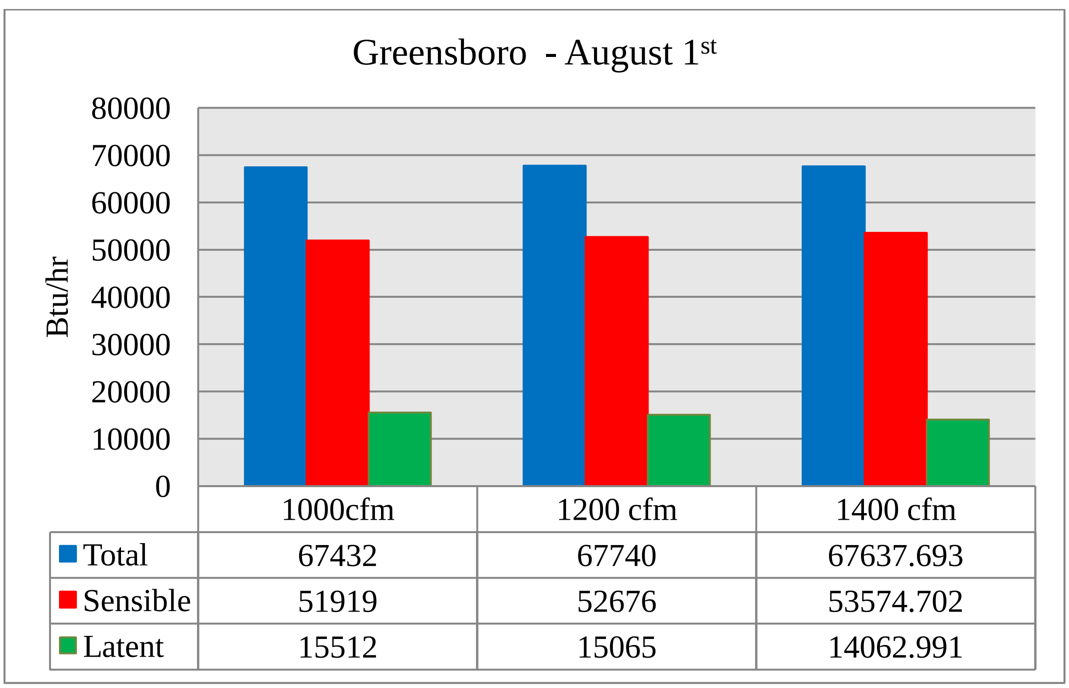


Figure B.6 Return air humidity ratio (lb H_2O / lb dry air on August 1st in (Greensboro)



	1000 cfm	1200 cfm	1400 cfm
Total	268865.79	268317	265875.85
Sensible	221950.84	225585	229733.54
Latent	46914.95	42731	36142.31

Figure B.7 Daily Total, Sensible and Latent cooling loads on August 1st (New York)



	1000cfm	1200 cfm	1400 cfm
Total	67432	67740	67637.693
Sensible	51919	52676	53574.702
Latent	15512	15065	14062.991

Figure B.8 Daily Total, Sensible and Latent cooling loads on August 1st in (Greensboro)

Appendix C

ASHRAE Climate Zones

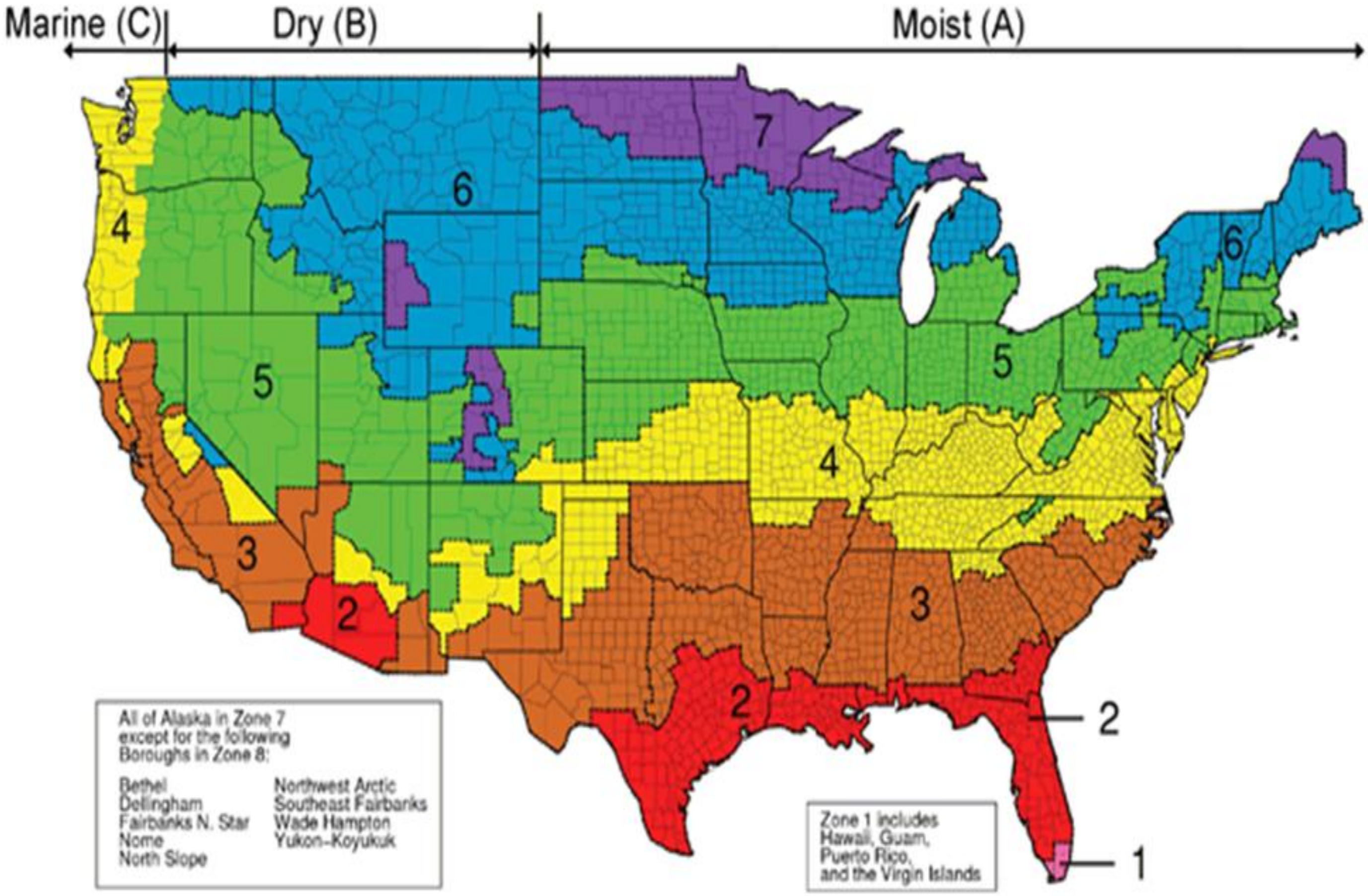


Figure C.1 ASHRAE Climate Zones Related to Map of USA