ABSTRACT

CHAUDHARY, GOPAL. Analysis and Evaluation of a Fire Station VAV HVAC System (Under the guidance of Dr. Stephen Terry)

Variable Air Volume (VAV) systems are highly capable HVAC systems that can provide precise temperature controlled air to places with the strictest of air quality requirements. However, their performance depends much on the heating and cooling method they are coupled with, as well as end use of the building they are being installed in. While some work has been done in the past to look at the applicability of VAV systems in homes and industrial spaces, not a lot of effort has been made to explore their application in mixed use buildings.

The research in this technical paper focusses on the analysis and modelling of a VAV system in a modern day Fire Station (mixed use). This presents an opportunity to contrast real world data from that system with computer generated models. The comparison attempts to discover trends between a multitude of variables such as the heating practices, internal loads, fuel used, and cost of operations.

The results obtained from the analysis help highlight the importance of correct design and control of VAV systems for use in mixed use buildings. The optimum solution to have a hot water loop instead of electric strip heat is modelled and presented. The benefit of having two separate Roof Top Units (RTU) instead of a single unit are discussed. Finally, other minor recommendations that may improve the energy efficiency of the building are also proposed.

Analysis and Evaluation of a Fire Station VAV HVAC System

by

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DEDICATION

I dedicate this thesis to my mom and dad, without whom my life would be empty. Their unending support and love is what got me to go the extra mile and to achieve my goals. I would also like to thank my girlfriend and the love of my life Poonam. Despite being thousands of miles away she has always managed to keep me company and support me in my darkest times. Last but not the least, my sister who is happily married has been and always will be big part of my life.

BIOGRAPHY

Gopal Chaudhary was born to Ravi Chaudhary and Madhura Chaudhary in the last decade of the 20th century in Patna, Bihar (India). He has one sibling, an older sister who is currently in the UK. Being the son of man working in a government job in India meant lots of transfers and relocations. This also meant new places to explore and new friends to make. Gopal was interested in science from a very young age. Although his interest in science was more in the dark realm of 'deconstruction'. Gopal would find himself in hot water on multiple occasions after "accidentally' breaking TVs, speakers or toys with the single intent of studying them from the inside. His interest in science and technology grew as he became older. Gopal attended many Catholic schools that are prevalent in the eastern part of India until his 8th grade, after which he enrolled at the Central School (Kendriya Vidyalaya Ganeshkhind) in Pune, Maharashtra. Here he completed his schooling and graduated with top honors.

Immediately following high school, Gopal enrolled at Sinhgad Institute of Technology, Lonavala, Maharashtra where he earned his B.E. degree in mechanical engineering with distinction. His interest in general knowledge and technology awarded him great success in national level quiz competitions and project competitions. Throughout his under graduate studies, Gopal maintained a first class with distinction.

Gopal was then recruited by Mahindra and Mahindra Automotive as a Quality Engineer at their vehicle engine assembly plant in Igatpuri. The eighteen months Gopal spent here were his toughest and least enjoyable. Working in factory was definitely not his dream job, coupled with the fact that he had the work environment was not conducive to his personal

and professional growth, made things worse. The work was not exciting at all and all hope seemed lost. However, every cloud has a silver lining; for him, it was desire to pursue his masters. Having already given his GRE, Gopal used the miserable experience at Mahindra as a catalyst to reignite his interest in pursuing higher studies. He applied to a couple of colleges in the USA and was accepted at both NC State and Carnegie Institute of technology. It was a tough decision but eventually NC State won out.

In 2104, Gopal's dream came true and he started his M.S. in Mechanical Engineering at NC State. Here he became a part of something that would shape his future and his career in major way; the Industrial Assessment Center. Having worked at the IAC for over one and a half years, he has earned valuable knowledge and skill. Having a wonderful team and an excellent set of superiors and guides helped him choose his career path in the field of energy efficiency. Gopal will complete his MSME degree in November 2015.

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I would like to thank first and foremost Dr. Stephen Terry for providing me the guidance and support needed to make this thesis a reality. I would also like to thank the following for their help in the computation, writing, and completion of this thesis: Alexander Viola, Taylor Atkins, Jackson Wooten, Daniel Paprocki, Mahlon Tucker Daniels, Sarah Meiers, Adefolaoluwami Odeniyi, Juliet Simpson and James Turner.

I would also like to extend my heartfelt gratitude to the Emily Barrett and her team at the Town of Cary – Transport and Facilities Dept. Without their help and supervision, the project would have been a lot longer and a lot more challenging.

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1 INTRODUCTION

1.1 Basics of a VAV Systems

Air conditioning is an indispensable part of modern life. Its versatility of use can never be underestimated, from manufacturing to health care, from homes to offices, from cars to airplanes etc. Though very useful and practical, its impact on energy costs cannot be ignored. In 2014, 41% of total U.S. energy consumption was consumed in residential and commercial buildings, or about 40 quadrillion British thermal units [1]. Additionally, nearly 100 million US homes and offices now have air conditioning [2]. With such vast penetration of air conditioning and such large amounts of energy being used, the correct selection and design of an HVAC system becomes paramount. This allows for significant opportunities to optimize energy usage and improve air conditioning performance.

HVAC systems may be classified in many ways based on usage, fuel use, capacity etc. One such classification based on air volume lists two systems, namely a Constant Air Volume (CAV) system and a Variable Air Volume (VAV) system. Constant Air Volume (CAV) is a type of heating, ventilating, and air-conditioning (HVAC) system. A CAV system, as defined by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), is a heating, ventilating, and/or air-conditioning (HVAC) system in which, the supply air flow rate is constant, but the supply air temperature is varied to meet the thermal loads of a space. Inversely, A VAV, is defined as a type of HVAC system that varies the airflow at a constant temperature.

A simple VAV system as shown in Figure. 1 [3] incorporates one supply duct that, when in cooling mode, distributes supply air at a constant temperature of approximately 55 °F (13 °C). Because the supply air temperature is constant, the air flow rate must vary to meet the rising and falling heat gains or losses within the thermal zone served.

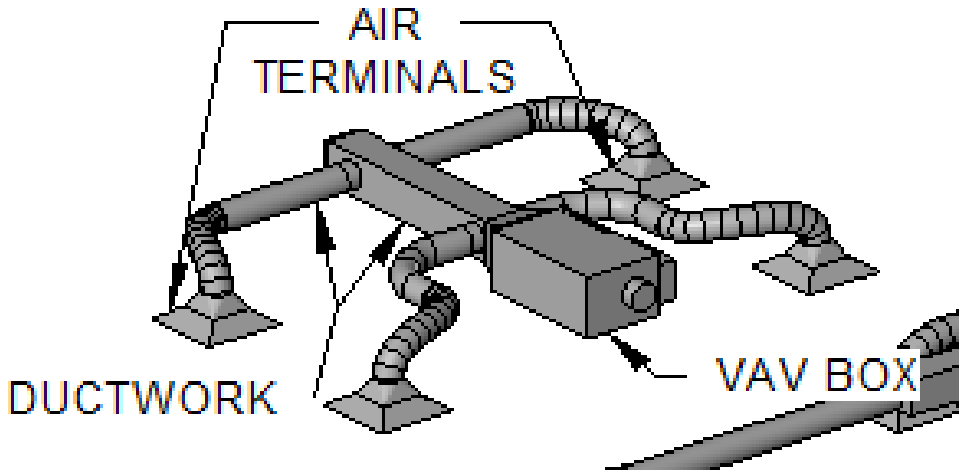


Figure 1: Standard VAV System

Such systems no matter how simple have a range of benefits such as:

i) Precise Temperature Control - In a single-zone VAV unit, the fan speed varies depending on the actual space temperature and the temperature set point, while the compressor modulates the refrigerant flow to maintain a constant supply air temperature. The result is precise space temperature control.

ii) Energy Savings and Reduced Wear - VAV fan control, especially with modern electronic variable-speed drives, reduces the energy consumed by fans, which can be a substantial part of the total cooling energy requirements of a building. Modulating control of the compressor also reduces wear and delivers further energy savings.

iii) Increased Dehumidification - Because VAV air flow is reduced under part-load conditions, air is exposed to cooling coils for a longer time. More moisture condenses on the coils, dehumidifying the air. Thus, although a constant-volume and a single-zone VAV unit maintain the same room temperature, the VAV unit provides more passive dehumidification and more comfortable space conditions

1.2 Project Objective

Fire Station 8 is part of the Town of Cary's network of new generation fire stations. This study will involve data collection and analysis, modeling of the subject facility and the HVAC system using Department of Energy's eQuest software, determination of optimum system design for the facility and other recommendations to help improve the energy efficiency of the building. The main objective of this study is to analyze the current state of VAV system at the Town of Cary Fire Station 8 and determine the root cause of discrepancy between design energy use and actual energy use during peak heating days. From analysis it will be determined why operating the HVAC system is costing the facility more than projected.

Along with the analysis of the facility, an in depth look into different VAV systems and their comparison with CAV systems is also provided. The nuances of a VAV system are detailed to enable better selection of such a system for a particular application.

1.3 Literature Review

Study of VAV systems and their applicability in different buildings has been done on multiple occasions by researchers. Reddy, Liu and Richman, and Claridge evaluated the potential of VAV terminal reheat systems in North American buildings for satisfactory zone ventilation back in 1998 [4]. The paper compared the differences in energy use and ventilation air flow rates supplied to different zones in a building using outdoor air ventilation strategies. A simplified simulation methodology was used to predict heating and cooling energy use of a terminal reheat VAV system.

The three major results derived from the simulations are the effect of air flow rates on the performance of the VAV systems, the relation between summer and winter air flow rates with indoor air quality, and the variation in performance of a VAV reheat system based on the geographical location of a building [4].

The authors varied the ventilation levels in simulations and observed the interaction between the air flow rate and the HVAC loads. Across all simulations for buildings in different locations, increase of air flow rate of up to 20% over ASHRAE code did not necessarily eliminate the problem of deficient cooling in hotter locations like Dallas. However, the performance was satisfactory in cooler areas such as Seattle, where the cooling loads are lower. Surprisingly, it was discovered that heating energy use was virtually independent of ventilation strategy used.

The results of their study were that, in general, across all building types and geographic locations, there is no significant HVAC related savings for VAV reheat systems during the heating season. However, the authors did find that when looking at individual climate zones, colder climates showed greater savings while warmer climates [4] tend to see virtually no improvement unless the air flow rates are increased substantially.

Mehdi, Yao, and Cook did a similar study in the UK [5] with the objective of characterizing the energy performance of an HVAC system for a prototype office building. The study considered the performance of the system with either CAV or VAV systems as secondary. Simultaneous dynamic simulations of the building using TRNSYS software was done.

Results of this study [5] showed that, in the secondary part of HVAC&R systems, utilizing the VAV system (instead of CAV system) reduced the auxiliary energy consumption by 15–35% which is equal to a 5–15% reduction in the total energy consumption of the building. In addition, the amount of energy used to meet the heating and cooling demands in VAV systems was slightly higher than in CAV systems The results of this study on the performance evaluation of 36 HVAC&R systems was used as a complementary part of the existing building energy benchmarks in order to enhance the performance characterization assessment of a variety of HVAC&R systems. In terms of CO2 emissions, the best performance was delivered by a Combined Cooling, Heating and Power system (CCHP) when linked to a VAV system [5] with heat recovery whereas, the highest pollution was produced by an absorption chiller and boiler when linked to a CAV system with terminal reheat coils.

Another study, this time from Japan, looked at the potential for energy savings with variable air volume systems using advanced simulation techniques. Toshio M. and Uiche I. studied the energy consumption [6] of a VAV system and compared it with that of several other types of air conditioning systems used in the interior and the perimeter zones of a model building. Subsequently, the energy saving properties of the VAV system were verified by actual measurements in an existing building equipped with it. The dynamic hour-by-hour heating and cooling loads were calculated using the standard computer program HASP/ACLD 7101 and the standard meteorological data in Tokyo provided by SHASE (Society of Heating, Air Conditioning and Sanitary Engineers of Japan).

The results of this study showed that comparison of air conditioning systems by simulation proved that in the interior zone the annual cooling coil loads increased per system when going from a VAV system (both single and dual duct) to a CAV system. The minimum was about 60% of the maximum, and about 40% energy saving [6] was obtained. Although the single duct VAV system was found to be inferior in the perimeter zone, it could be improved by using a dual duct VAV system or by resetting the temperature of the supply air separately in each zone. As for the annual fan power consumption, the VAV system was found to be the most economical of all systems and its consumption was lower than that of a dual duct CAV system by 50% or about 33% less than of that of a terminal reheat CAV. These facts were verified by actual measurements in an existing building.

S.C. Sekhar wrote a paper which presented an evaluation of variable air volume systems in hot and humid climates [7]. The main objective of this paper was to compare critically the

performance of a VAV system with an equivalent constant air volume (CAV) system for five different buildings in such a climate, whose internal loads remained constant throughout but the thermal loads due to the building envelope and orientation varied among the five buildings. The presence of a diversity in cooling loads was investigated which then lead to the exploration of potential energy savings with a VAV system.

The end result being that, while maintaining the same floor area and altering either the window wall ratio or the shape of the floor plan, substantial energy savings were accrued with the use of a VAV system. A minimum energy savings of 11.5% was expected with a VAV system [7] due to building envelope variations for identical floor plan and similar internal loads. It was also established that savings in fan energy alone could be between 50-70% depending on the shape of the building, its orientation and envelope characteristics. The energy required for space cooling was also found to be significant and was in the range of 10.9-18.5%.

A more recent study that was conducted by [8] showed the energy savings opportunities with a VAV system for use in a building in a humid subtropical climate location such as Beijing and Chengdu. This paper is of significance as the climate zone is same as that of Cary. In this paper energy-saving of variable-air-volume system was compared to constant-air-volume system and fan-coil systems. One small office building was taken as an example, year-round energy simulations of the three kinds of HVAC systems (VAV, CAV and Fan Coil System) of the same building were made in different cities across China.

The end result showed that VAV systems outperformed their CAV and fan coil system (FCS) counterparts in every city where the simulations were run. In Beijing especially, the improvement was seen to be 19.2% over CAV and 6.4% over FCS [8]. The paper was comprehensive in its coverage of the locations across China and used a standardized building to ensure equitable results. However, the main drawback of this analysis was the lack of clarity in the type of VAV system used. There was also lack of evidence to support their claims whether the energy savings were for cooling, heating or both.

As seen by the sample of studies discussed here, there have been numerous attempts to analyze and study VAV systems across several nations. While all these papers have dabbled into the benefits of VAV systems in standard office buildings, none have been able to justify its applicability for a mixed use building. There is also a no mention or comparison of VAV systems based on reheat techniques. This results in a lack of clear understanding of the applicability of VAV systems in general. Additionally, all the papers have used different modelling and simulation techniques which may color the result in some capacity and prevent the comparison of VAV with other HVAC systems on a common ground.

Taking a slightly different approach for our study, VAV systems are studied intrinsically before being compared with other HVAC systems. Our study limits this analysis to a single geographical location and a singular mixed use building. This allows us to create a new subset of research that opens up the avenue of new and more in depth research on the topic of VAV systems.

2 UNDERSTANDING VAV SYSTEMS

2.1 HVAC Zoning and Control

An HVAC system conditions the supply air to provide the spaces with an acceptable combination of humidity and temperature within the comfort zone. It also provides a sufficient amount of outside air for ventilation. The effectiveness of the system depends upon three factors:

- The quantity of air being supplied, measured in cfm (cubic feet per minute).
- The temperature of the supply air.
- Humidity

To heat or cool a space, these two factors are combined in different ways depending upon the type and design of the particular HVAC system. The combinations are:

i) Constant Volume-Variable Temperature (CV-VT)

ii) Variable Volume-Constant Temperature (VV-CT)

iii) Variable Volume-Variable Temperature (VV-VT)

As the indoor air temperature varies, the humidity also increases or decreases. Ideally, the indoor humidity remains within the comfort zone. However, in dry climates, many HVAC systems have a humidifier unit in the central air handler to increase the moisture level in the conditioned air when it is needed. In humid climates, systems may have a means of removing moisture from the supply air. Figure. 2 shows a simple CV-VT system.

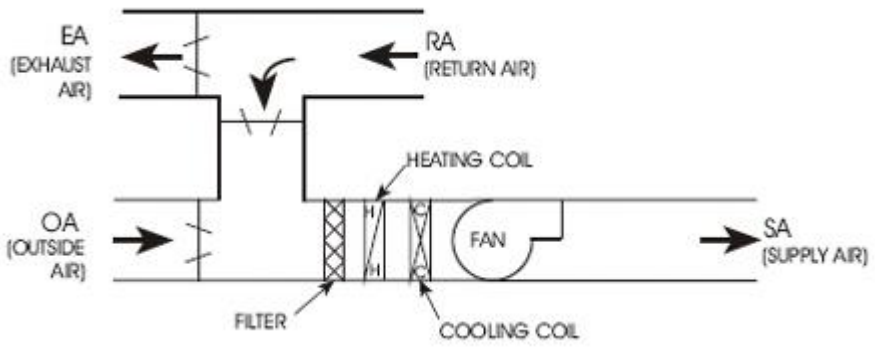


Figure 2: Single Zone CV-VT System

In large buildings, the HVAC system must meet the varying needs of different spaces. Different zones of a building have different heating and cooling needs. In the past, large commercial buildings were often designed with a central light well i.e. primarily with exterior walls for windows. This admitted light and air to the inner rooms. As building and land costs increased and HVAC systems developed, this inner light well was eliminated. Now buildings are designed with a core of inner rooms where the light well used to be (Figure. 3). The development of this inner core created a new air conditioning problem. The core spaces do not generally require heating, but do require cooling and ventilation, even in winter.

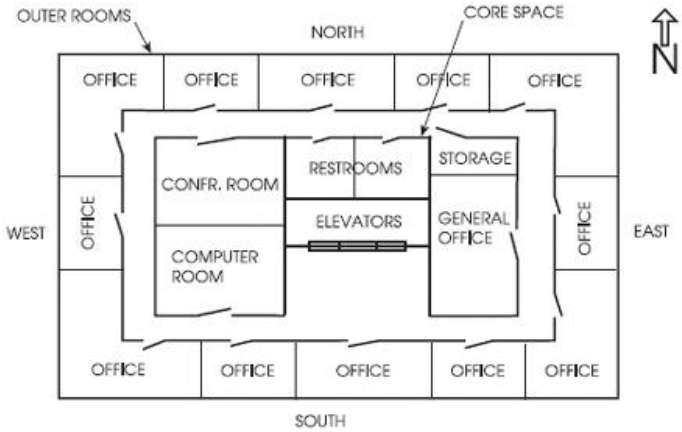


Figure 3: Typical Zoning of an Office Space

The rooms along the outer walls require either heating or cooling as well as ventilation. These outer rooms are also called perimeter spaces since they have at least one wall exposed to the outside temperatures. This means that these offices have more heat loss and heat gain than the interior rooms:

- On cold days, heat transfers from the heated spaces of the building to the colder outside air (heat loss).
- On warm days, heat transfers from the warm outside air to the cooler spaces of the building (heat gain).

The outer rooms gain or lose heat at a varying rate. For example, when the sun strikes one side of a building, that side has more heat gain than the sides that are shaded. The position of the sun, color of the wall, insulation, amount of glass, and shading all affect this solar heat gain.

The core of the building gains heat from the interior load such as people, lights, and equipment. Therefore, these spaces generally do not require heating except for a top floor where heat is lost through the roof. The normal condition is that they gain too much heat and therefore require cooling when occupied even on the coldest of winter days.

The cooling load for both core spaces and outer spaces depends on many factors such as these:

- Types of occupancy (density, active or passive work tasks).
- Heat produced by equipment.
- Type and level of lighting.

Since different spaces have different rates of heat gain and heat loss, it is impossible for an HVAC system that delivers the same temperature of air at a fixed volume to every space to provide comfort conditions for all of them. Therefore, heating and cooling must be supplied at varying rates to different zones of the building. A zone is a space or group of spaces in a building with similar requirements for heating and cooling. All rooms in a zone can be supplied with the same temperature supply air at the same flow rate.

2.2 Features of a VAV System.

Variable Air Volume (VAV) systems (Figure. 4) were developed to be more energy efficient and to meet the varying heating and cooling needs of different building zones. A zone can be a single room or cluster of rooms with single, dual or even triple ducting, each the same heat gains and heat loss characteristics.

In dual-duct systems, the air handling unit has two coils, a continuously operating cooling coil and a continuously operating heating coil. The cooling coil feeds chilled air into a cold air duct. The heating coil feeds hot air into a hot air duct. The two ducts run in parallel throughout the building. At each space, air is tapped from the two ducts by a terminal unit.

The terminal unit has a hot air damper and a cold air damper. When the space thermostat calls for heating, the hot air damper opens. When the thermostat calls for cooling, the cold air damper opens.

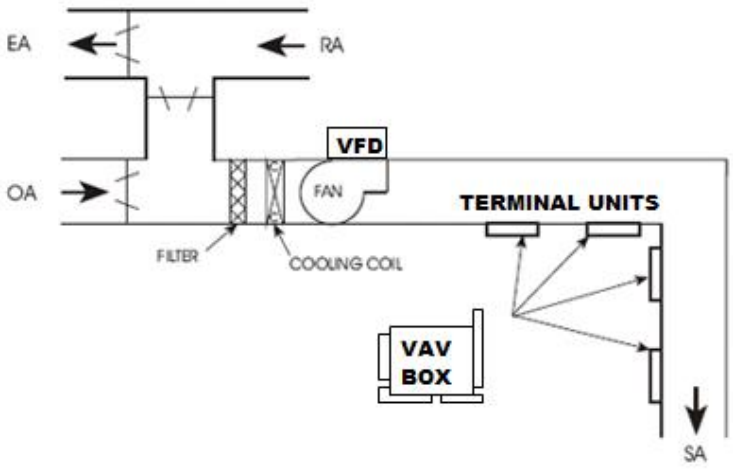


Figure 4: A VAV System In Operation

Efficiency suffers if a terminal unit mixes chilled air with heated air under any conditions. The system may be designed to do this deliberately under low conditioning loads to maintain a minimum air flow into the spaces. VAV systems can save as much as 30 percent in energy costs as compared to conventional dual duct systems. In addition, they are economical to install and to operate. Duct sizes and central air handling units are smaller and the design and installation is generally much simpler.

The main duct for a typical VAV system is designed to provide air at 55°F which is called primary air. Room thermostats control the amount of primary air delivered to each zone through modulating dampers for each zone. These dampers vary the volume of air to each zone according to the cooling needs.

Early VAV systems varied the fan cfm output according to the total need of the zones. The fan was sized for the maximum probable load. As the air volume for the zones varied, the static pressure (SP) in the main duct tended to vary. An SP sensor in the main duct controlled the fan output to maintain a constant supply duct static pressure. The fan output was varied either by fan inlet vanes or by a damper at the fan outlet. These systems were variable volume-constant temperature (VV-CT).

Early VAV systems were cooling only, so a separate source of heat was needed, particularly for the outer rooms. This was usually supplied by perimeter heating in the rooms. These early VAV systems were low-cost to install. However, depending upon the position of the zone dampers, the zones were subject to delivering too much cold supply air, which sometimes created drafts and air noise. These systems were very difficult to balance.

Figure 5: VAV Terminal Unit

To better control the secondary air delivered to each zone, the VAV terminal unit was developed. A terminal unit is a small metal box (Figure. 5) located in the supply air duct just

before the outlet of each zone. A terminal unit is also called a VAV box, VAV unit, or outlet box.

Each terminal unit (Figure. 6) receives primary air from the central air handling unit at the same temperature (about 55°F). The terminal unit contains a primary-air damper also known as a butterfly damper which modulates (continuously changes position) according to signals from the automatic control system. For this type of box, the primary-air damper regulates the volume of cold primary air delivered to the terminal unit according to the needs of the spaces. This is also the volume of secondary air delivered to that space. Figure. 6 shows only the key features of a terminal unit.

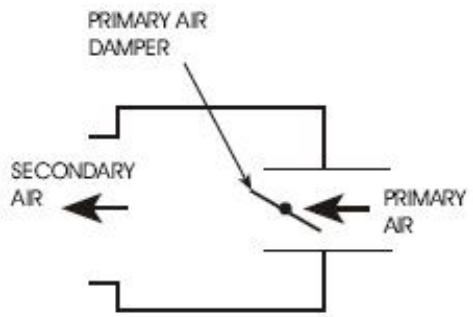


Figure 6: Simple VAV Terminal Unit Without Controls

In its simplest form, the VAV terminal unit provides cooling only. The core spaces generally do not require heating, and the outer rooms are heated by another means. Other terminal units may include a means of heating. The big advantage of VAV systems with terminal units is that they are able to meet the comfort requirements of different zones in a building without heating and cooling at the same time.

There are instances where precise temperature control may be needed especially for large buildings or mixed use buildings. In this case, the VAV system may also include a reheat coil system. A reheat system works by cooling air to a temperature low enough to condense or remove moisture (for humidity control) and to offset the largest heat gain generated anywhere in the building. The cold (~50˚F - 55˚F) air is pushed out everywhere in the building and then warmed up throughout the reheat system, depending on the space cooling needs. This system provides the flexibility of delivering different amounts of cooling to different zones at different times. Reheat systems are very precise but they are energy intensive if not controlled correctly

VAV systems are either pressure dependent or pressure independent. The first VAV terminal units were pressure dependent. They had no means for limiting the quantity of supply air. In pressure dependent systems, the volume of air supplied by the terminal unit varies depending upon the static pressure (SP) in the primary air duct. The primary-air damper in the terminal unit is controlled by a thermostat in the space. However, the airflow through the damper varies according to the SP in the main duct. Terminal units that are close to the supply fan are likely to supply too much primary air. Terminal units that are farthest from the supply fan are not likely to supply enough primary air.

Pressure independent terminal units have flow-sensing devices that limit the flow rate through the box. They can control the maximum and minimum cfm that can be supplied and are therefore independent of the SP in the primary air duct. Almost all HVAC systems

installed or retrofitted at present have pressure independent VAV terminals. Pressure independent systems can be balanced and will allow the correct airflow from each terminal. Because each terminal unit regulates its primary air volume independently, the volume (cfm) of primary air delivered by the central air handling unit varies according to the demands of the terminal units in the system. This means that the supply fan in the central air handling unit must vary its output in order to meet the needs of all the terminal units. If the primary-air dampers of most terminal units are full open, the cfm required for the entire system is high. If most terminal unit dampers are closed, the cfm required for the system is much less.

In many current systems, the rpm (speed) of the central supply fan is regulated by the control system to meet the changing demands of the system. A static pressure (SP) sensor in the primary air duct sends a signal to a controller that regulates the fan speed to maintain a constant SP in the primary air duct. The location of the SP sensor in the primary duct is critical to the performance of the system. It is best placed near the terminal unit that is most difficult to supply. This is the location that has the greatest pressure drop from the fan. If the sensor is placed too close to the supply fan, the SP in the supply duct will be too high during periods of low cfm demand.

2.3 Types of VAV Systems

Early VAV systems were not highly regarded by HVAC technicians. They were considered almost impossible to balance and to keep in balance. Today, pressure independent VAV systems are widely regarded as the best HVAC system design available. This change is largely a result of improvements in the terminal unit. Many VAV systems and terminal units

have been developed to provide for the particular needs of a building. VAV systems may be of many different classifications based on controls, terminal units, and method of temperature control.

The following four types of VAV systems shown in Table 1 are most commonly used and will be discussed in brief.

Table 1: Common Types of VAV Systems

Classification	Ducting Method	Primary Air Flow	Secondary Air Flow	Notes
VAV with Cooling Only	Single	Variable	Variable	
VAV with Reheat	Single	Variable	Variable/Constant	May have fan powered reheat units
VAV with Dual Duct	Dual	Variable	Variable	
VAV with Bypass	Single	Constant	Variable	Variation may include a changeover

2.3.1 VAV with Cooling Only

In a single-zone VAV system, the temperature sensor in the zone is used to vary the cooling or heating capacity and the airflow delivered by the supply fan to maintain supply-air temperature at a desired set point. Figure 7 Shows a typical single zone VAV.

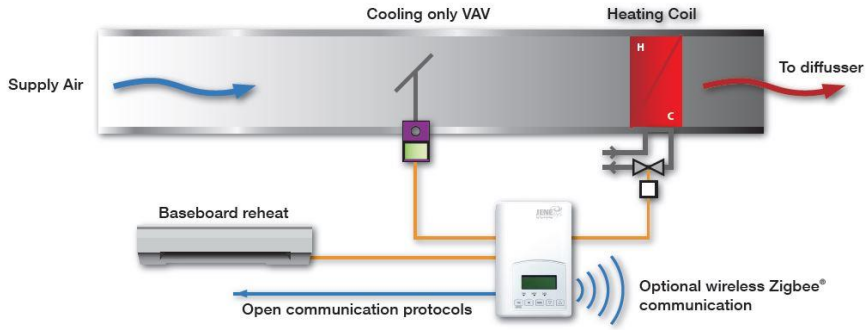


Figure 7: Typical Single Zone VAV with Cooling Only (Heating Coil is Separate)

Traditionally, single-zone VAV has been used for larger, densely occupied zones that have variable cooling loads. Common examples include gymnasiums, cafeterias, lecture halls, auditoriums, large meeting rooms, churches, and arenas. Therefore, it has been available primarily in larger air handling units and large, packaged rooftop equipment. This type of equipment has long been available with variable-speed fans and cooling/heating that can be staged or modulated to control discharge air temperature. However, due to increased focus on reducing energy use, single-zone VAV is starting to be used more often in K-12 classrooms, retail stores, dormitories, and even offices. As its popularity has increased for these smaller zones, this functionality is beginning to be offered in smaller equipment, such as small packaged rooftop units or direct expansion (DX) split systems, fan-coils, classroom unit ventilators, and water-source heat pumps.

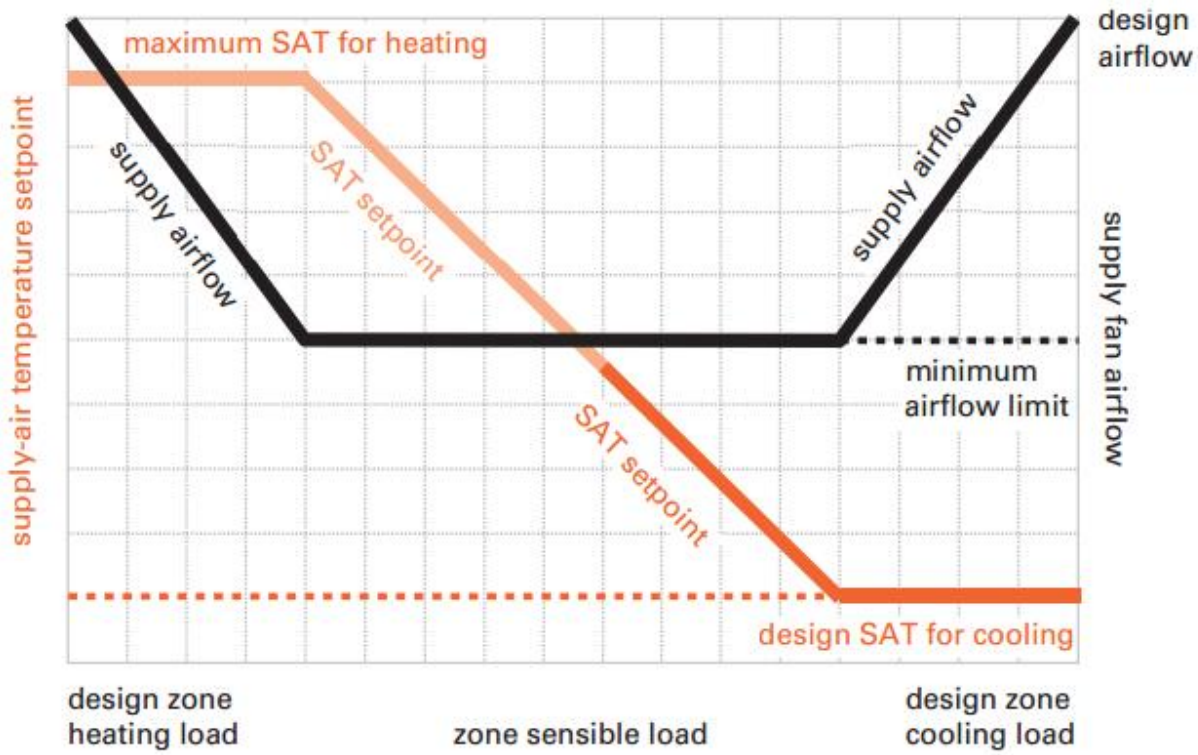


Figure 8: Typical Control Scheme for a Single Zone VAV with Cooling Only

Figure 8 depicts an example control sequence for a single-zone VAV system [9] that uses a variable-speed fan. (Note that control can vary by manufacturer, and there are nuances that depend on whether chilled water or DX is used for cooling and whether the type of heater used can support variable airflow.) When the zone is at design sensible cooling load (right-hand side of the chart), this system delivers maximum supply airflow at the design supply-air temperature (SAT) for cooling (e.g., 55°F).

As the zone cooling load decreases, supply airflow is reduced as needed to maintain the desired temperature in the zone. This is accomplished by varying the speed of the fan motor. Cooling capacity is then staged or modulated to maintain SAT at the same design set point. If the system has an airside economizer, the economizer may provide all or part of the cooling needed to achieve the SAT set point. Eventually, the zone sensible cooling load decreases to the point where supply airflow reaches a minimum limit.

As the cooling load continues to decrease, the fan remains at minimum airflow, but now the SAT set point is gradually reset upward to avoid overcooling the zone. Depending on the outdoor conditions, as the SAT set point increases, eventually no mechanical cooling is needed to make this temperature, so the compressors are turned off. And even the airside economizer modulates back to bring in only the minimum outdoor airflow required for ventilation. When that happens, and the load continues to decrease, the zone temperature begins to drop below the zone cooling set point, into the dead band between cooling and heating set points. The fan continues to operate at minimum airflow, with no compressors or heaters operating, and the zone temperature is allowed to float within this dead band.

2.3.2 VAV with Reheat

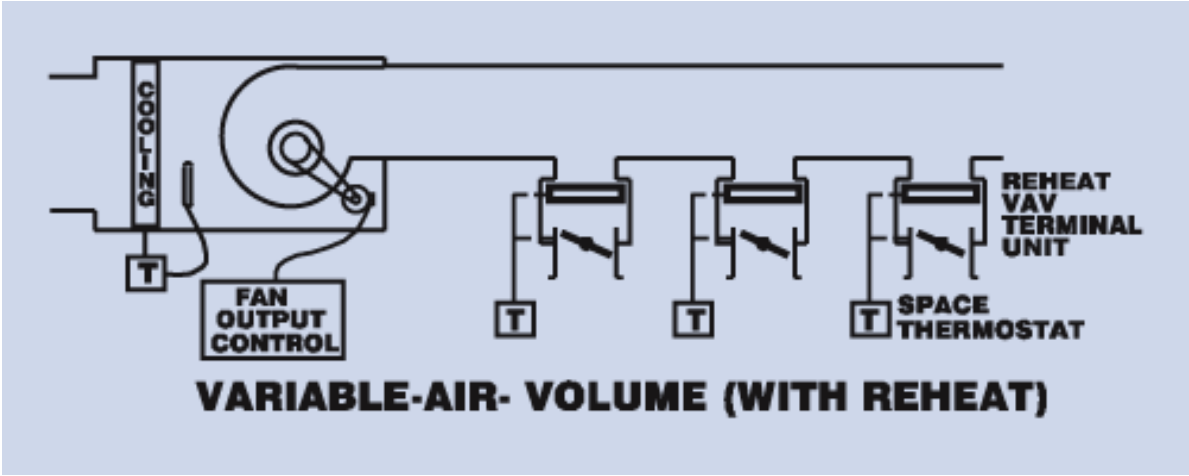


Figure 9: VAV With Reheat

Figure 9 shows a typical VAV system with reheat. In VAV systems, chilled air is distributed to spaces from an air handling unit, and the temperature of individual spaces is controlled by throttling the quantity of air into each space. The throttling is accomplished by terminal units that are controlled by the space thermostats.

VAV systems were originally introduced as a more efficient alternative to constant-volume reheat systems. The VAV concept offers two major efficiency improvements: (1) it reduces or eliminates reheat and (2) it minimizes fan power. Unfortunately, the full efficiency potential of many VAV systems has not been achieved in practice. In many systems, the terminal units contain reheating coils. These are used to provide space heating, or to reheat the chilled air to allow a minimum air flow to be maintained in the spaces, or for both purposes.

As VAV systems became widespread during the 1980's, it became apparent that they incur a number of conflicts with comfort and air quality. Indeed, VAV has become notorious for its comfort problems. Attempts to resolve the comfort problems and to reduce the installation cost of the systems often resulted in systems that squandered much of the efficiency potential of the VAV concept.

By far the largest opportunity for energy conservation in VAV reheat systems is minimizing the operation of the reheat coils. Also, you can save both reheat energy and fan energy by using accurate fan modulation to match cooling or heating load changes.

Energy conservation measures for optimizing the efficiency of VAV systems, must pay particular attention to reducing comfort and ventilation problems and include reducing reheat losses by adjusting the discharge temperature of the cooling coil automatically with supply air temperature reset controls. Changeover controls that can completely eliminate reheat for much of the time should also be installed.

VAV systems also allow you to use energy saving thermostatic controls, including dead band thermostats and temperature setback thermostats.

2.3.3 VAV with Dual Duct

The next common type of VAV system is double duct or dual duct systems [10]. VAV dual-duct systems have the potential of being efficient and comfortable, but they often have significant opportunities for improvement.

In dual-duct systems, the air handling unit has two coils, a continuously operating cooling coil and a continuously operating heating coil. The cooling coil feeds chilled air into a cold air duct. The heating coil feeds hot air into a hot air duct. The two ducts run in parallel throughout the building. At each space, air is tapped from the two ducts by a terminal unit. The terminal unit has a hot air damper and a cold air damper. When the space thermostat calls for heating, the hot air damper opens. When the thermostat calls for cooling, the cold air damper opens. This can be seen in Figure 10 shown below.

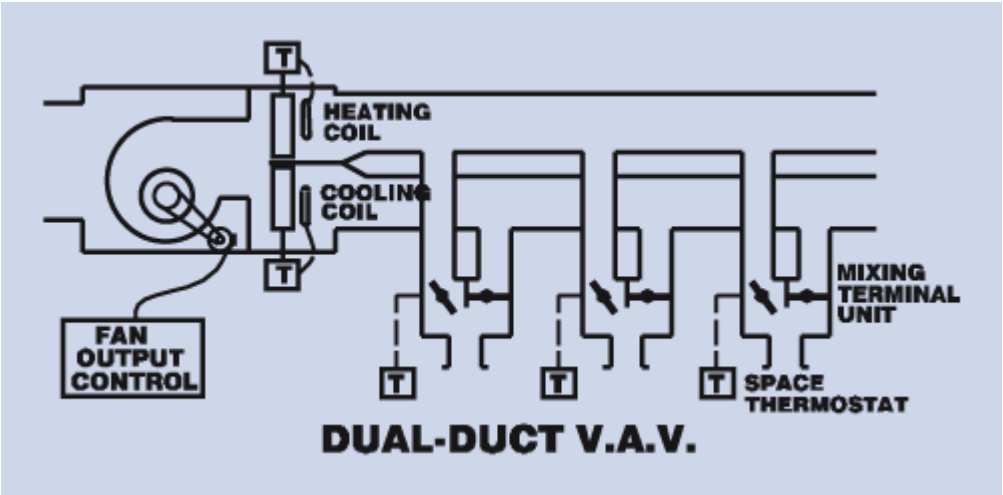


Figure 10: A Dual Duct VAV System

Efficiency suffers if a terminal unit mixes chilled air with heated air under any conditions. The system may be designed to do this deliberately under low heating or cooling loads to maintain a minimum air flow into the spaces.

"Triple-duct" systems avoid air mixing. They are similar in appearance to dual-duct systems. The main difference is a third duct that carries unconditioned air (a mixture of return air and outside air) for mixing with either the heated air or the chilled air. If properly installed, triple-duct systems have no mixing losses, except for leakage that occurs inside the terminal units.

The largest opportunity for energy conservation in VAV double duct systems is eliminating any mixing of heated air and chilled air. Also, fan energy can be saved by using accurate fan modulation to follow the cooling or heating loads. Variable-speed fans are the favored approach. VAV systems also allows the use of energy saving thermostatic controls, including dead band thermostats and temperature setback thermostats. The dual-duct VAV system design is especially favorable for exploiting the energy saving principle of the outside air economizer cycle.

2.3.4 VAV with Changeover Bypass

Changeover-bypass VAV is a comfort system developed for light commercial applications. A changeover-bypass VAV system responds to changing cooling or heating requirements by varying the quantity or volume of air delivered to each zone. Each zone has a thermostat for individual comfort control. An HVAC unit delivers a constant volume of air to the system. As the volume of air required by the zone changes, excess supply air is directed to the return duct via a bypass duct and damper. (See Figure 11 for typical system components.) A changeover-bypass VAV system combines the comfort benefits of VAV with the cost effectiveness and simplicity of packaged, constant volume unitary equipment.

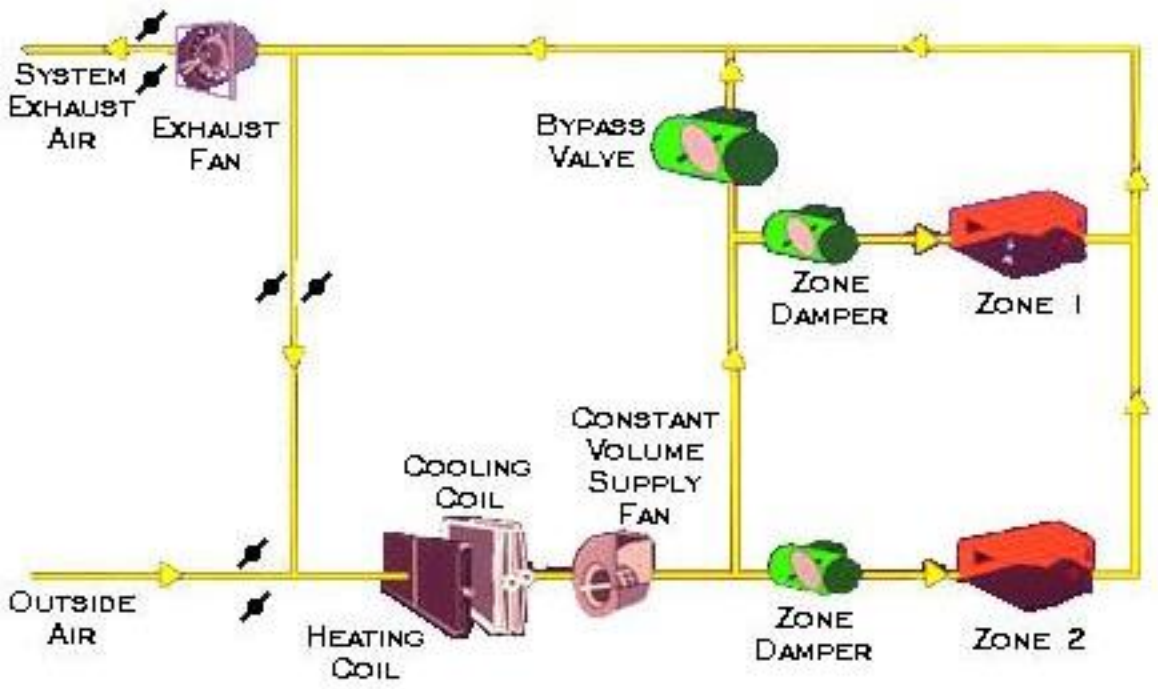


Figure 11: A Changeover-Bypass VAV System

Changeover bypass systems [11] use the practicality and cost effectiveness of constant volume unitary components like packaged rooftop units, split systems, or water-source heat pumps, and simply add dampers and a central control panel to coordinate the components. This allows for multiple (over 10) individual sensors (thermostats) for independent temperature control.

A changeover-bypass VAV system commonly consists of an HVAC unit with a constant-volume supply fan, and direct-expansion (DX) cooling. This combined system has the ability to “change” to the heating mode or cooling mode, depending on individual zone comfort requirements. A heating coil or a gas-fired heater and an outside air damper are possible options.

A temperature sensor in each zone communicates information to an electronic controller on the VAV terminal unit. The controller then modulates the zone damper open or closed, supplying heating or cooling air to the zone. The HVAC unit delivers a constant volume of supply air to the system. In order to maintain duct static pressure, a bypass duct and damper are required to bypass (detour) air not required in the zones.

The VAV terminal unit controller communicates zone temperature information to a central control panel (CCP). The CCP also gathers information from the system, including duct static pressure and supply-air temperature. The CCP determines zone heating or cooling needs using voting (or polling) logic, then requests heating or cooling from the HVAC unit. The CCP directs the HVAC unit to provide ventilation air to high-occupancy areas (demand control ventilation) or free cooling when the outside air temperature falls below the temperature set point (economizer control).

3 TOWN OF CARY FIRE STATION 8

3.1 Facility Overview

The Town of Cary has united its fire and police personnel under one roof with the construction of Fire Station 8. The building, located at 408 Mills Park Drive (Figure 12 [12]) on property owned by the Town of Cary, includes a Police Department substation.

Figure 12: Town of Cary – Fire Station 8

Fire Station 8 is housed in a modern 14,410 sq. ft. building that is LEED registered and has a low carbon footprint. Located at 408 Mills Park Drive, it is a good example of a green building. The facility was built & commissioned in 2013 and has a satellite Police response office attached to it.

The two storied building uses a host of energy efficient equipment including LED lighting, occupancy sensors, zone based thermostatic control for HVAC, solar lighting, solar water

heating and a staged roof top unit (RTU) that services the building's HVAC needs. The garage bay also uses natural gas fired infra-red heaters as shown below in Figure 13

Figure 13: Natural Gas Fired Infrared Heaters in the Garage Bay

The RTU has an integrated furnace that burns natural gas in order to provide primary heat during winter. The building itself is maintained using ten VAV boxes that control the volume of air being delivered to each zone in order to maintain temperature in that zone. The VAV boxes have electric strip heat built into them that are designed to provide secondary heating. The RTU is shown in Figure 14 below.

Figure 14: RTU at Fire Station 8

3.2 Energy Use

For analysis of the facility, energy use data from 2014 was used. Energy use is split into two categories, electricity and natural gas. Electricity is provided by Duke Progress Energy on the Medium General Service (MGS) rate schedule. Based on this rate schedule the facility is charged at the rate of $0.0712/kWh for energy use and $4.81/kW for demand however based on mutual understanding between the Town of Cary and Duke Progress Energy, the facility paid a combined rate of $0.0821/kWh for electricity. The facility also uses natural gas for which it is charged $8.97/MMBTU. The building is occupied between 8 AM and 5 PM, however occupancy levels are highly variable throughout the day.

3.2.1 Electricity

Table 2 shows electricity usage of the facility for the year 2014. It should be noted that electricity use remains fairly consistent during the summer months; this may indicate that there is not a lot of reheat occurring. Humidity is kept at 50% and any form of reheat is for dehumidification purposes only. However, the energy use numbers for the winter months indicate high electricity use which may be caused due to use of electric heating. This will be investigated in a later section.

Table 2: Electricity Usage for 2014

2014	Billed Electricity	Energy Cost	Demand	Demand Cost	Total Cost
Month	kWh	$	kW	$	$
Jan	44,920	$3,198	68	$385	$3,525
Feb	27,720	$1,974	73	$385	$2,325
Mar	24,840	$1,769	73	$385	$2,120
Apr	20,120	$1,433	67	$385	$1,755
May	22,840	$1,626	52	$385	$1,876
Jun	24,080	$1,714	58	$385	$1,993
Jul	28,400	$2,022	64	$308	$2,330
Aug	28,480	$2,028	63	$303	$2,331
Sep	32,080	$2,284	64	$308	$2,589
Oct	23,120	$1,646	58	$279	$1,925
Nov	30,200	$2,150	72	$346	$2,497
Dec	30,720	$2,187	67	$322	$2,510
Total	337,520	$24,031	868	$4,175	$27,710

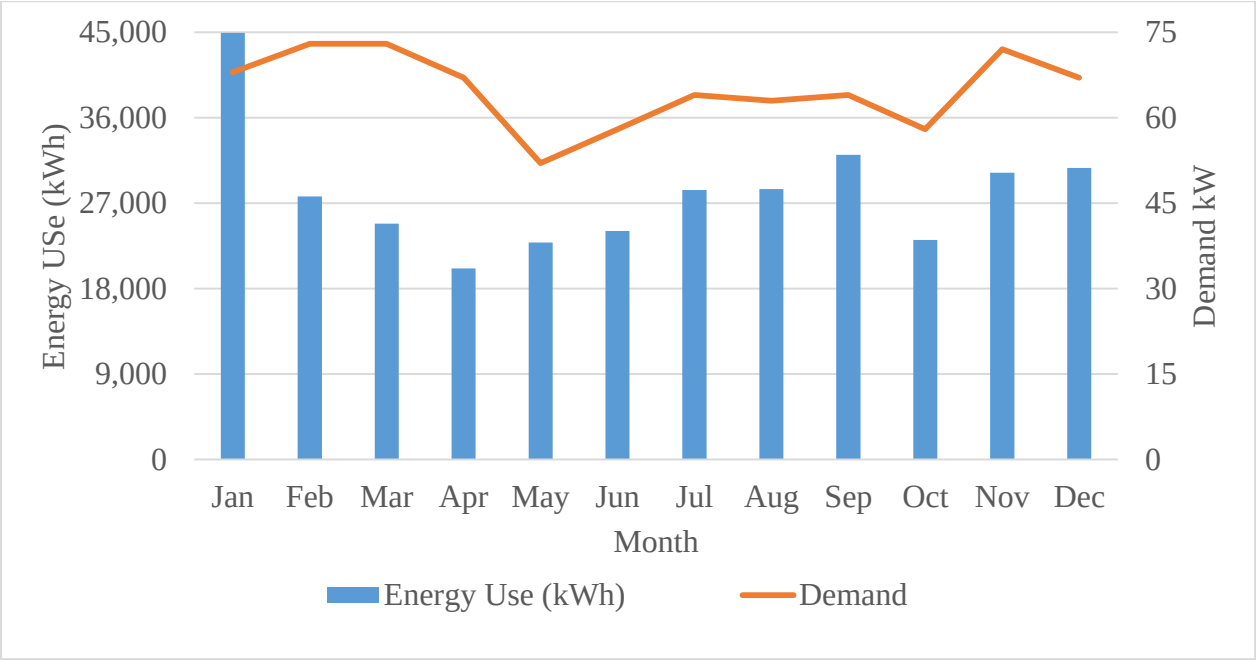


Figure 15: Electric Energy Usage and Demand

3.2.2 Natural gas

The following table (Table 3) details the actual natural gas consumption for the year 2014.

Table 3: Natural Gas Usage for 2014

Year	Natural Gas Use	Natural Gas Cost
2014	MMBTU	$
Jan	70.2	$630
Feb	65.8	$590
Mar	36.2	$325
Apr	11.2	$100
May	6.5	$58
Jun	11.7	$105
Jul	7.5	$67
Aug	8.2	$74
Sep	7.7	$69
Oct	8.3	$74
Nov	8.5	$76
Dec	13.1	$118
Total	254.9	$2,286

Natural Gas use remains is very low during the summer months as expected. The water heaters and gas stoves are the only things using it. During the winter months, energy use numbers increase slightly which indicates that the gas fired infrared heaters are being operated for cooling the garage bays. The natural gas usage is far less than what is expected for a building of this size. This observation will be analyzed in later sections. Energy usage trends are shown below in Figure 16 and Figure 17

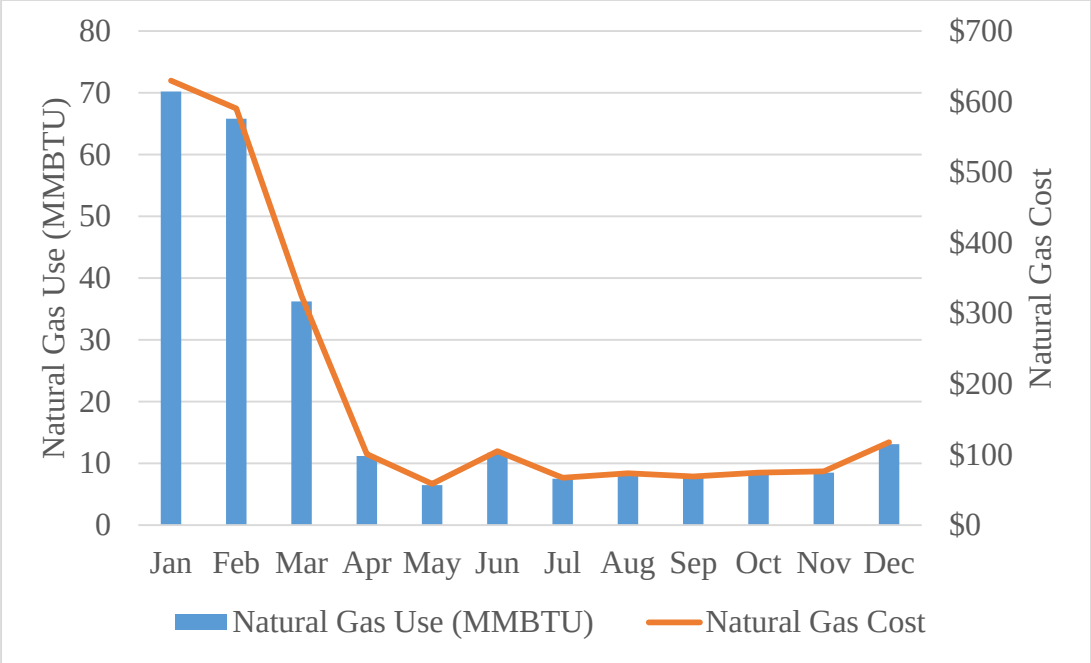


Figure 16: Natural Gas Usage and Cost

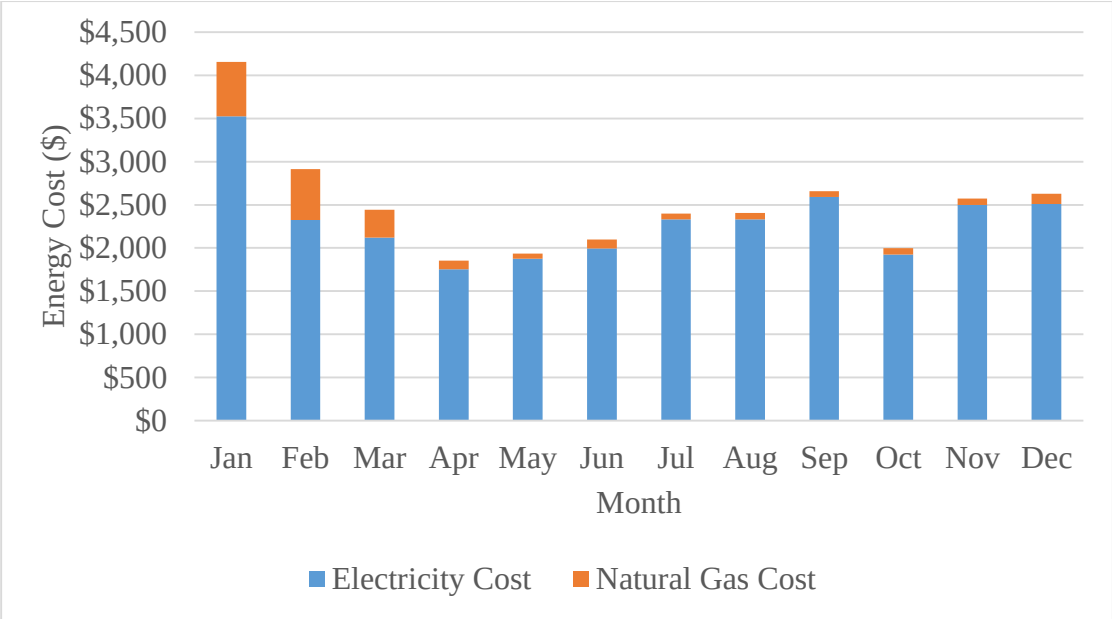


Figure 17: Total Energy Cost

Based on the figure above we can see that even though natural gas use goes up in the winter months, the electricity use remains high. This is the cause for concern and a focal point of investigation

3.3 Understanding the Problem

The recently built Town of Cary Fire Station 8 was designed to be a model of energy efficiency for all current and future fire stations in the Town of Cary. In particular, the HVAC system that was installed at the site was designed to be highly energy efficient and have a very low carbon footprint. To support the claims of energy efficiency, an energy model was created during the design realization phase to project the energy use of the facility over a period of one year. Initially, the projected energy use data from the model was very optimistic. However, after a year of operation the actual electrical energy use for the facility was found to be much more than the projected energy use.

The analysis of the HVAC system occurred in multiple stages, ranging from initial survey to data collection and finally design modelling. Over the course of the next few months, it was determined that the VAV reheat system within the HVAC seemed to be contributing to the additional electrical energy use. Data was collected by MAE Energy Solutions staff and students to assist in the analysis of the HVAC system. From the data collected, several energy models were created and analyzed to determine the root cause and determine alternative solutions that may be taken to help improve the system.

First, a walkthrough of the building was conducted by the assessment team to gather high level information and understand the HVAC system installed. The team also interacted with the building's original designers and HVAC consultants. Next, the roof top unit (RTU) at the fire station was data logged to determine the reason behind increased use of electricity during winter. Figure 18.

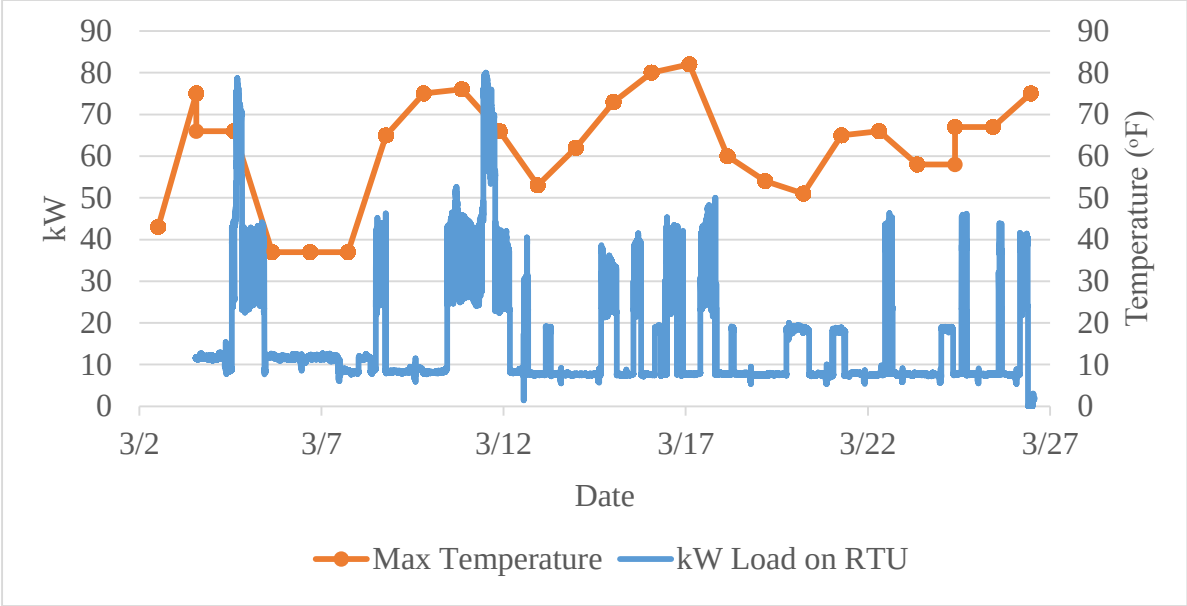


Figure 18: Logged Data for the RTU

The logged results showed that the maximum load on the RTU goes up 80 kW while the average load is about 15 kW. The 'spikes' in loads, as seen above, are likely due to increased use of air conditioning during peak cooling operation periods but only some during periods of very low temperature. Due to the fact that the logged data was from a period of very low temperatures, it might indicate that the increased loading may be due to the compressors running within the RTU. This was followed by an energy model using the eQuest modelling software created to verify that the system was performing as per the approved design.

Data gathered from this model concluded that the VAV system was using electric strip heat as primary during heating season instead of from the natural gas fired furnace. This was confirmed during a subsequent visit, when on a very cold morning with temperature in the low teens, the natural gas furnace stacks were not only cold but also showed no signs of paint wear (peeling) which would indicate use over time.

4 BUILDING LOAD MODELS

The investigation involved trial and error type modelling to determine the mode of operation that would cause the electric use to be high during the winter. Initial investigation pointed towards the compressors in the RTU, it was speculated that the RTU was running the compressor and using its bypass heat to provide reheat to the building instead of the natural gas. Thus putting extra load on the compressor and driving up the electrical usage during winter.

This conclusion was at odds with the logger data since we were not able to see a spike in RTU loading during very cold days. This led the investigation team to abandon this theory. It was only when the original commissioning plans were made available to the investigation team that it was determined that during the commissioning process [13], the design was changed in such a way that the electric strip heat acted as primary and the natural gas was barely used, if at all.

This required further investigation and hence, the original building design models (eQuest [14]) that were created during the initial stages of the project were retrieved. The analysis of the models confirmed this claim and it was determined that the natural gas heat was indeed replaced with electric strip heat as effective primary source of space heating.

To understand the problem better, two energy models that were originally created during the design phase of the building (by a third party) were fine-tuned and simulated for the purpose

of this analysis. First was an ASHRAE baseline model, created to determine the minimum design that will satisfy the ASHRAE building efficiency guidelines.

Second, a proposed pre-construction model that improved upon the ASHRAE model with lower energy costs, through the use of a single duct VAV system that used of natural gas fired heaters for the RTUs with backup electric strip heat.

After comparing the two models with the actual energy use profiles of the building it was determined that although the building was sized correctly, the HVAC system itself was designed in way that virtually eliminated the need for gas heat to ever activate. This placed the electric heat strips firmly as the primary source of heat thus producing a cascading effect on the energy use patterns of the building.

As a solution, a new energy model was created that utilized hot water coils at the VAV boxes instead of electric reheat. This resulted in a reduction in proposed electrical energy use during winter, conversely it increased the natural gas use. However, as natural gas is less expensive as a fuel and has a smaller carbon footprint, this model was found to be a better fit for the fire station.

The result for all the models were compared and it was concluded that while a hot water system was the better option, having two HVAC systems where one served the inner (core) rooms and the other served the outer (perimeter) rooms was the optimum design as it allowed for having better temperature control with minimal use of electric strip heat during winter.

To conclude, additional recommendations for lighting, thermostatic control and HVAC data reporting system were discussed in order to make the building as energy efficient as possible.

Some of the basic details of common to all the models such as the 3D layout and floor plans are shown below in Figure 19. These have been used to model both the pre-construction and post-construction designs and are very accurate.

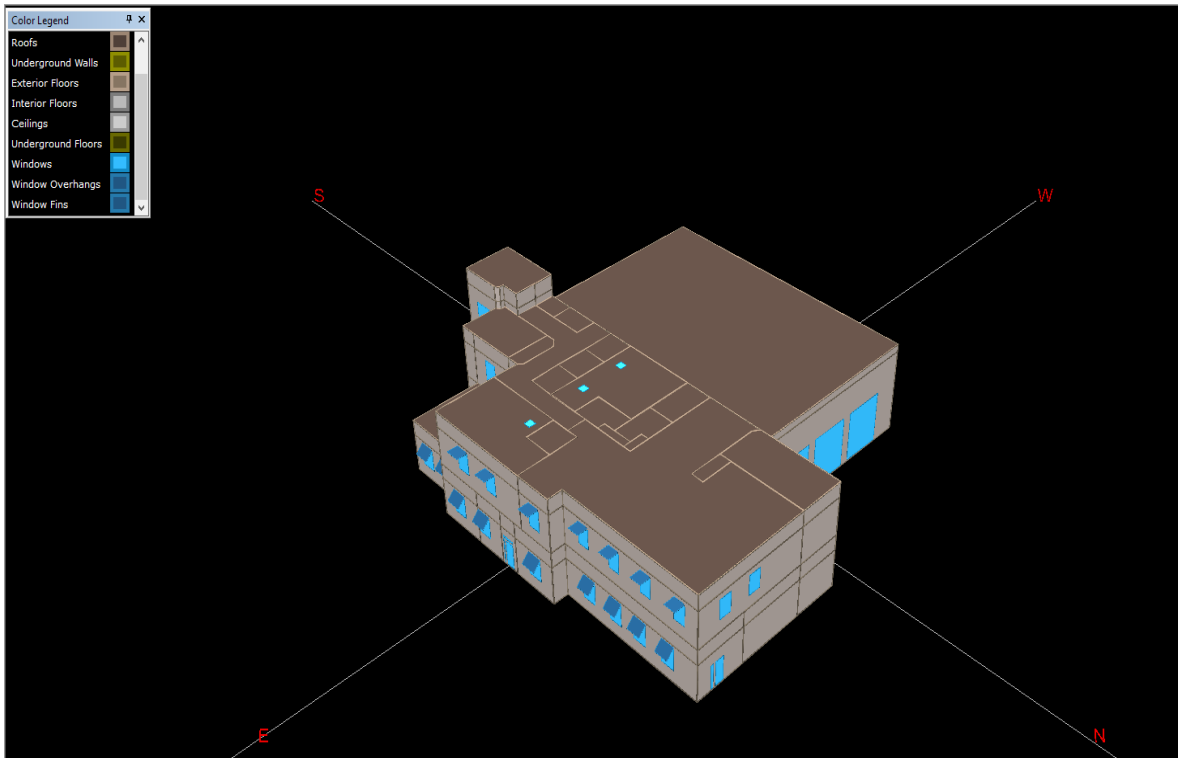


Figure 19: 3D Layout of Fire Station 8

This layout is based on the floor plans approved by the Public Works Department of Town of Cary which are shown in Figure 20 and Figure 21.

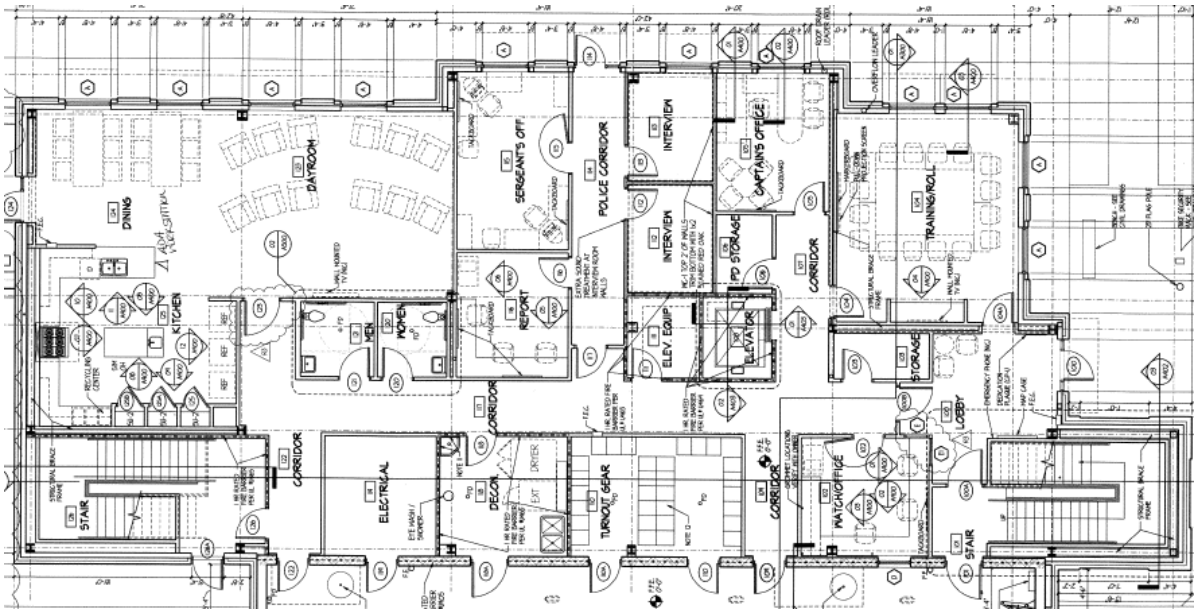


Figure 20: First Floor Plan

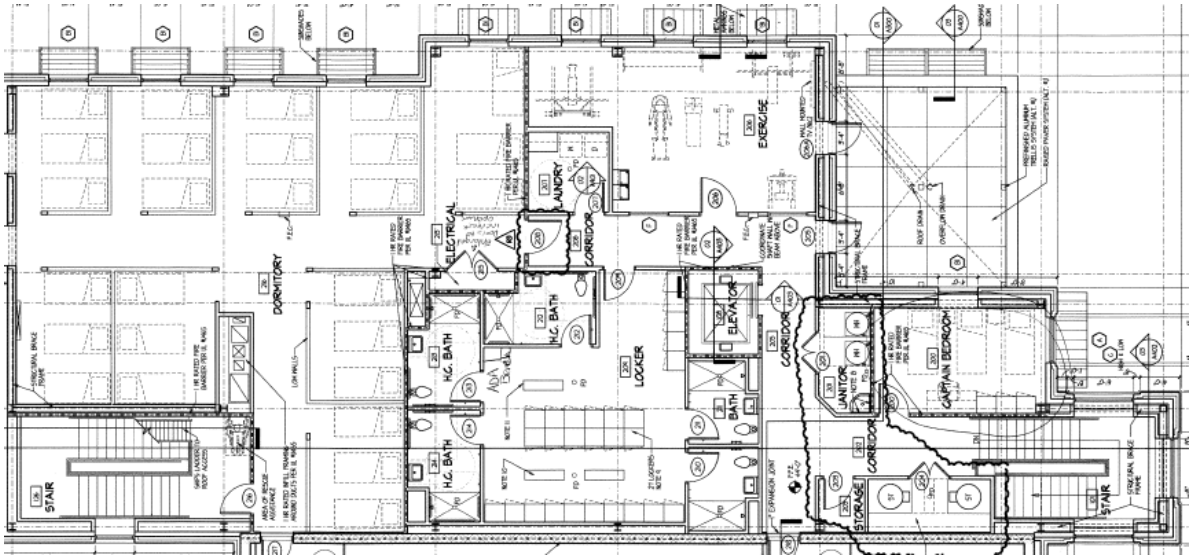


Figure 21: Second Floor Plan

4.1 Pre - Construction Design Models

The first phase of the design modelling involves analyzing and simulating models that were created during the design realization phase of the project, i.e. pre-construction phase. These models were created by a third party for the Town of Cary to determine the feasibility of the HVAC system and to verify whether they meet and exceed the minimum ASHRAE baseline requirements.

For this purpose, two models were created, the first model was meant to simulate the building's HVAC performance according to ASHRAE 90.1.2007 [15] baseline requirements. The second model was created to meet the 'Green Building' standards of the Public Works Department of Town of Cary and provide an improvement of at least 30% over the ASHRAE baseline.

Since both of these models were pre-construction, they may not account for changes in occupancy levels and weather pattern differences that may have occurred once the building commenced operation. Hence, for the purpose of this research both these models were suitable modified and simulated to achieve results that are discussed in the next section.

4.1.1 Baseline ASHRAE Model

The first step in creating the model for the building is to input its construction information. Since the exact construction details were known, these were used to ensure that the results are as accurate as possible. The ASHRAE baseline model for this building requires it to adhere to the ASHRAE 90.1.2007 standards. This standard is used to define all aspects of the building from an HVAC point of view. Hence, the model must be able to meet the minimum requirements for indoor air quality (IAQ), ventilation, occupant comfort, temperature ranges and infiltration.

The building operation schedule was set to be open 9 hours per day and 7 days per week but closed on holidays. The entire model was divided into ten zones each with its own indoor air quality (IAQ) and cooling/heating requirements. As an example Figure 22 shows the space properties for the 'Dining Area'. All the other zones were designed using the same methodology.

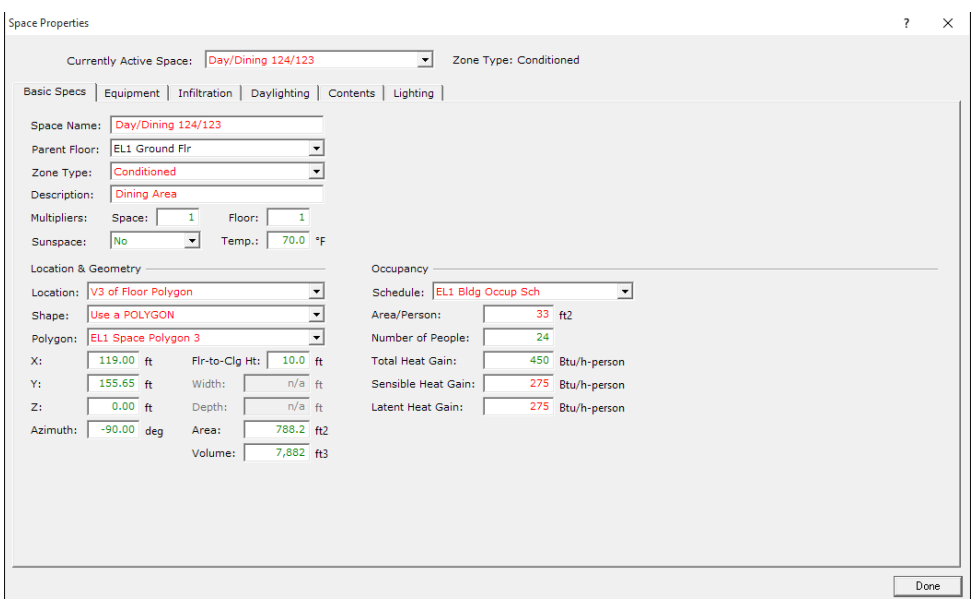


Figure 22: Space Properties for Dining Area

With the characteristics of the building entered, the heating and cooling systems can be input. The model uses an RTU with a direct expansion (DX) coil for cooling, a natural gas furnace for primary heating, and an air side economizer to reduce the amount of mechanical cooling required. The set points were chosen to be cooling at 70°F and heating at 68° F to maintain comfort. Finally, the humidity set point was chosen to be a maximum of 50% to keep the occupants comfortable. All sizing of equipment was automatically done by eQuest. With all of these variables set, the simulation could be run. For ease of understanding, the HVAC system layout is shown in the schematic below (Figure 23).

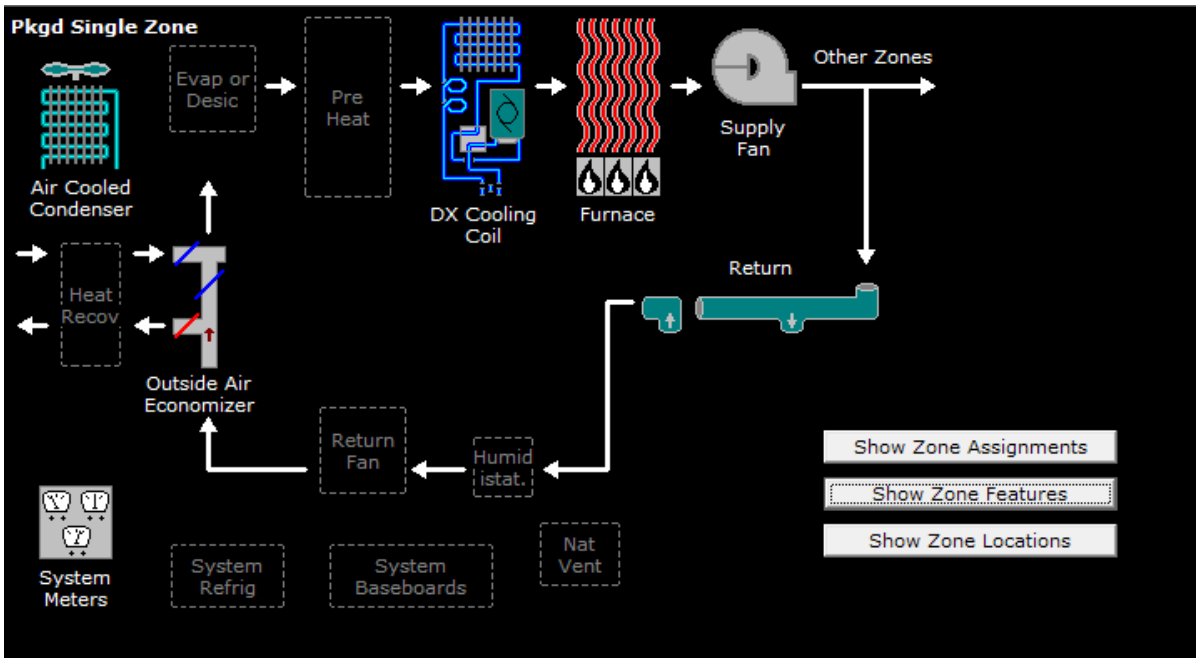


Figure 23: HVAC Layout for ASHRAE Baseline Model

Setup for the HVAC system in eQuest is shown below in Figure 24. The system is designed per zone and the return air path is ducted. The same HAVC design details are maintained for the 2nd floor as well.

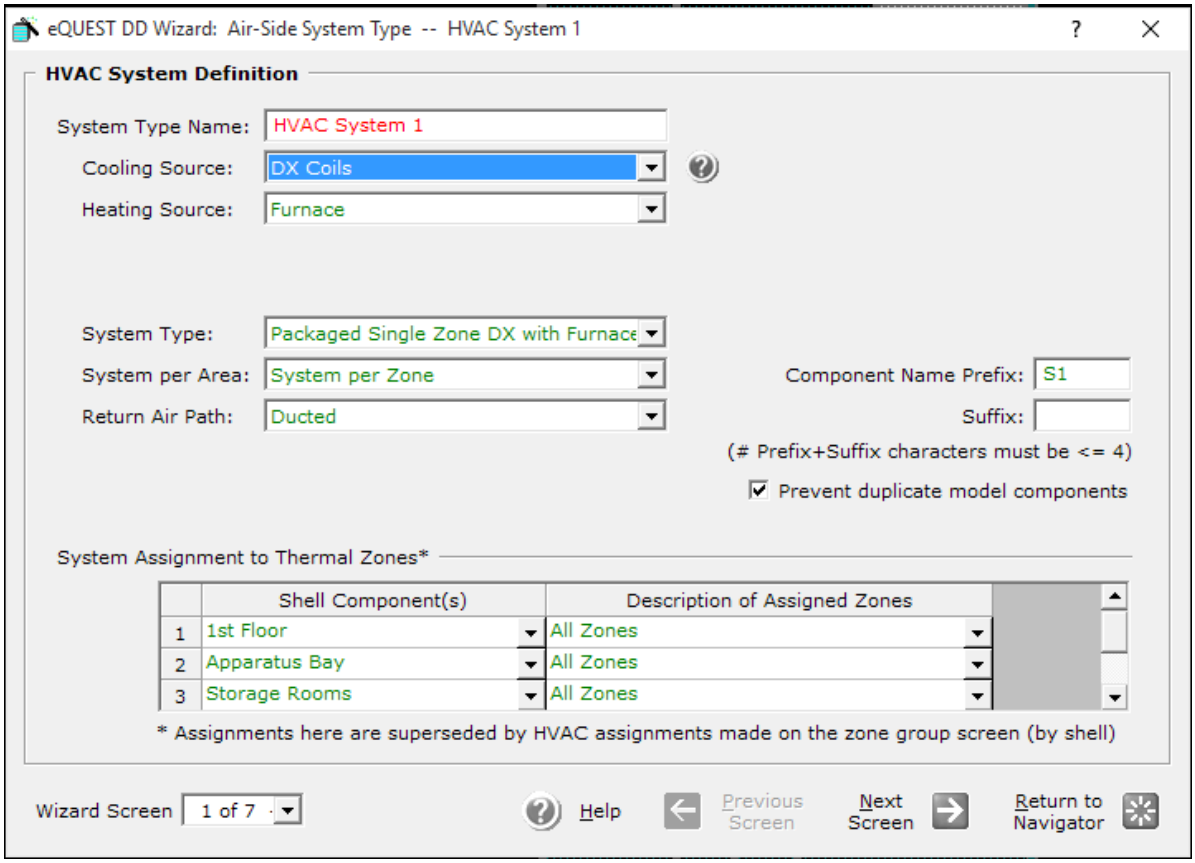
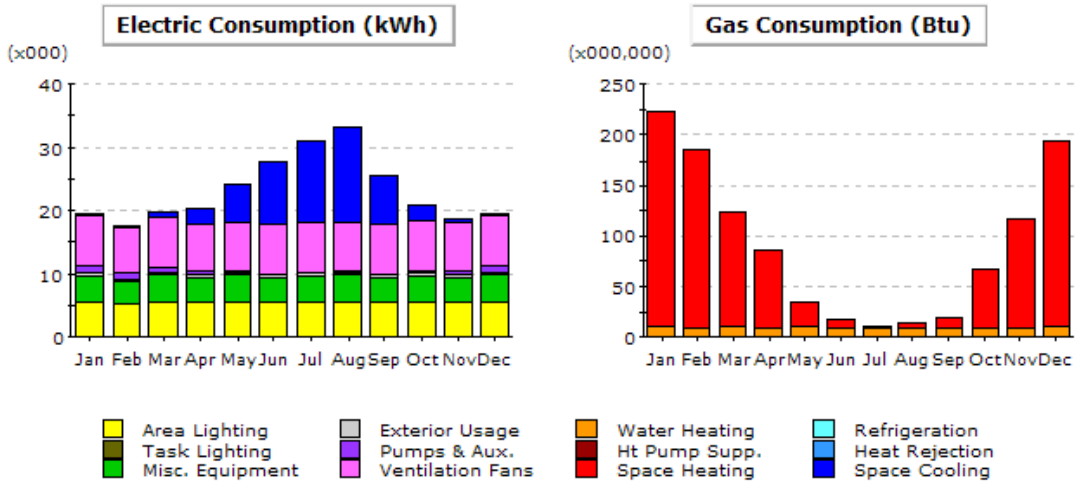


Figure 24: Air Side System for ASHRAE Baseline Model

As the electric and gas bills that were obtained were from 2014, the simulation was set to use that year's weather data for Raleigh. Shown below in

Project/Run: 2852 03-10-11 Cary Fire Station 8 (0deg - Baseline) - Baseline Design Run Date/Time: 10/11/15 @ 16:00



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.07	0.10	0.74	2.33	5.85	10.29	13.01	14.96	8.02	2.39	0.48	0.21	58.44
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	7.99	7.22	7.99	7.74	7.99	7.74	7.99	7.99	7.74	7.99	7.74	7.99	94.11
Pumps & Aux.	1.30	0.98	0.80	0.50	0.15	0.09	0.07	0.07	0.08	0.39	0.73	1.14	6.31
Ext. Usage	0.36	0.33	0.36	0.35	0.36	0.35	0.36	0.36	0.35	0.36	0.35	0.36	4.24
Misc. Equip.	4.02	3.68	4.12	3.93	4.06	3.98	4.02	4.12	3.97	4.02	3.97	4.08	47.98
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	5.71	5.16	5.71	5.53	5.71	5.53	5.71	5.71	5.53	5.71	5.53	5.71	67.24
Total	19.45	17.47	19.73	20.37	24.12	27.97	31.16	33.21	25.69	20.87	18.79	19.49	278.32

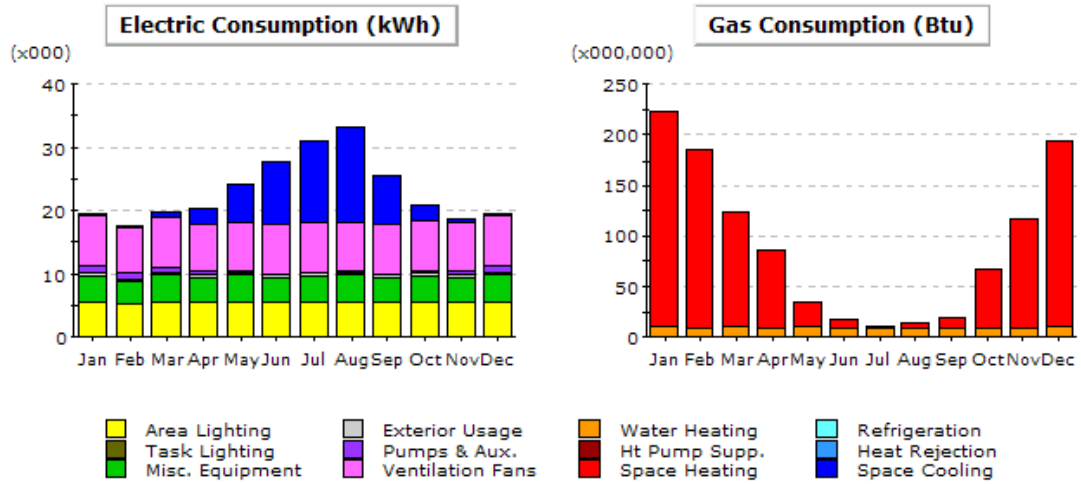
Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	213.7	176.5	113.6	76.1	24.3	8.0	1.6	3.5	11.3	57.9	108.0	184.6	979.2
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	10.2	9.3	10.4	9.9	10.1	9.7	9.8	9.9	9.5	9.8	9.7	10.1	118.6
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	223.8	185.9	124.0	86.1	34.5	17.7	11.5	13.4	20.8	67.8	117.7	194.8	1,097.8

eQUEST 3.65.7163 Monthly Energy Consumption by Enduse Page 1

Figure 25. are the energy consumption plots and tables generated by eQuest for the heating and cooling of the building.

Project/Run: 2852 03-10-11 Cary Fire Station 8 (0deg - Baseline) - Baseline Design Run Date/Time: 10/11/15 @ 16:00



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.07	0.10	0.74	2.33	5.85	10.29	13.01	14.96	8.02	2.39	0.48	0.21	58.44
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	7.99	7.22	7.99	7.74	7.99	7.74	7.99	7.99	7.74	7.99	7.74	7.99	94.11
Pumps & Aux.	1.30	0.98	0.80	0.50	0.15	0.09	0.07	0.07	0.08	0.39	0.73	1.14	6.31
Ext. Usage	0.36	0.33	0.36	0.35	0.36	0.35	0.36	0.36	0.35	0.36	0.35	0.36	4.24
Misc. Equip.	4.02	3.68	4.12	3.93	4.06	3.98	4.02	4.12	3.97	4.02	3.97	4.08	47.98
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	5.71	5.16	5.71	5.53	5.71	5.53	5.71	5.71	5.53	5.71	5.53	5.71	67.24
Total	19.45	17.47	19.73	20.37	24.12	27.97	31.16	33.21	25.69	20.87	18.79	19.49	278.32

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	213.7	176.5	113.6	76.1	24.3	8.0	1.6	3.5	11.3	57.9	108.0	184.6	979.2
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	10.2	9.3	10.4	9.9	10.1	9.7	9.8	9.9	9.5	9.8	9.7	10.1	118.6
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	223.8	185.9	124.0	86.1	34.5	17.7	11.5	13.4	20.8	67.8	117.7	194.8	1,097.8

eQUEST 3.65.7163 Monthly Energy Consumption by Enduse Page 1

Figure 25: eQuest Simulation Results for ASHRAE Baseline Model

It can be seen that since this model assumes natural gas furnace as the primary heat source with no strip heating, the natural gas use goes up proportionally. Additionally, the electricity consumption for the winter drops save for the fans that are modelled to run 24/7. This is projects the energy use for the building according to the AHRAE baseline.

While this model meets the ASHRAE minimum, it still does not produce optimum results. Keeping this in mind, a second, more robust model was created by the designers to get the desired '30%' improvement that was required.

4.1.2 Final Proposed Model

The second model was an improvement over the ASHRAE model and was intended to be the 'Final Proposed Model' for the HVAC system to be implemented at Fire Station 8. This model made a number of minor changes and modifications to the ASHRAE baseline model with the intent of achieving a 30% improvement in energy usage. The physical layout and zoning of the building was kept the same. The HVAC system continued to use a DX cooling coil with gas furnace heat as primary and economizer to recover heat but the ventilation requirements and occupancy levels were both altered to represent a real life scenario for the building.

This can be seen below for the same 'Dining Area' space that was discussed earlier. The new space properties for the room are shown in Figure 26. Here, we can see that the occupant density has changed in order to better reflect actual usage. The new values are 8 people with 100 ft^2 per occupant as compared to 24 persons with 33 ft^2 per occupant.

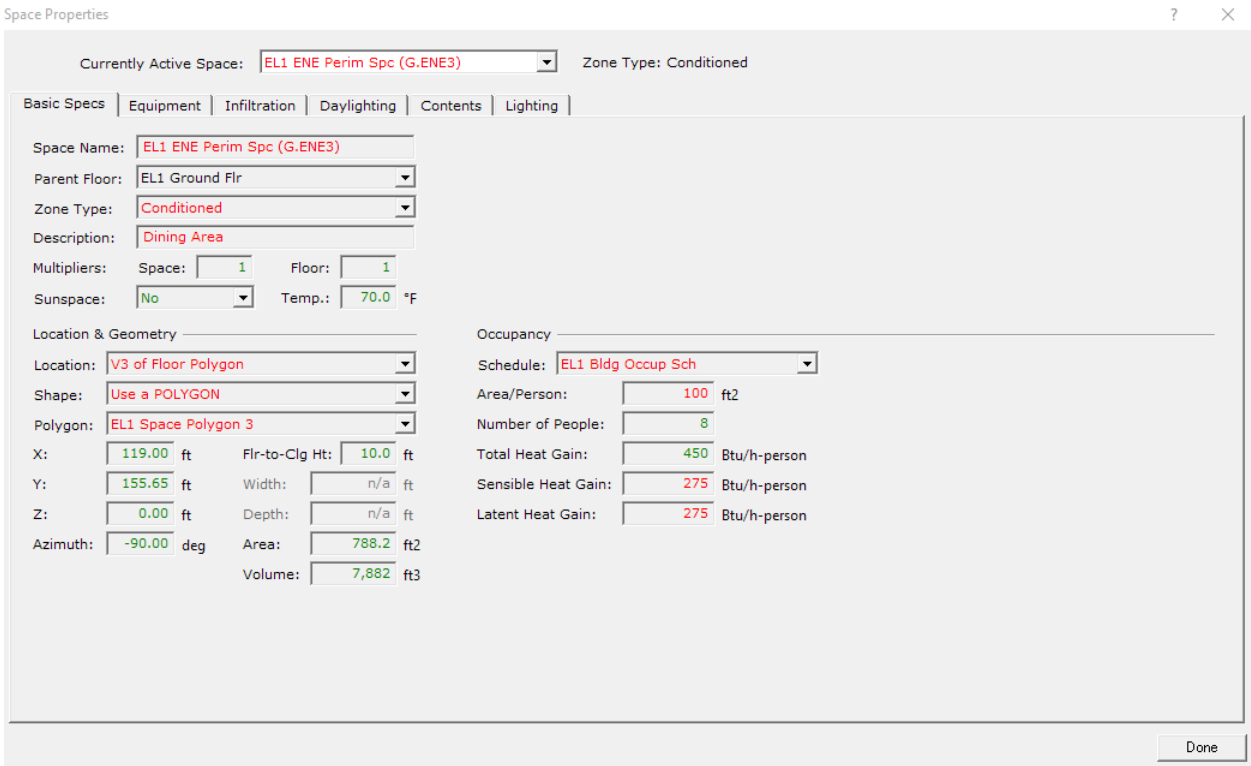


Figure 26: Space Properties for Dining Area for Final Proposed Model

This time the thermostat set points were chosen to be 76°F for cooling and 70°F for heating. Finally, the humidity set point was chosen to be a maximum of 50% as earlier to keep the occupants comfortable. The economizer was operated with a 65°F dry-bulb reset. Once again, all sizing of equipment was automatically done by eQuest. With all of these variables set, the simulation could be run. The set point details are shown in the screenshot below (Figure 27).

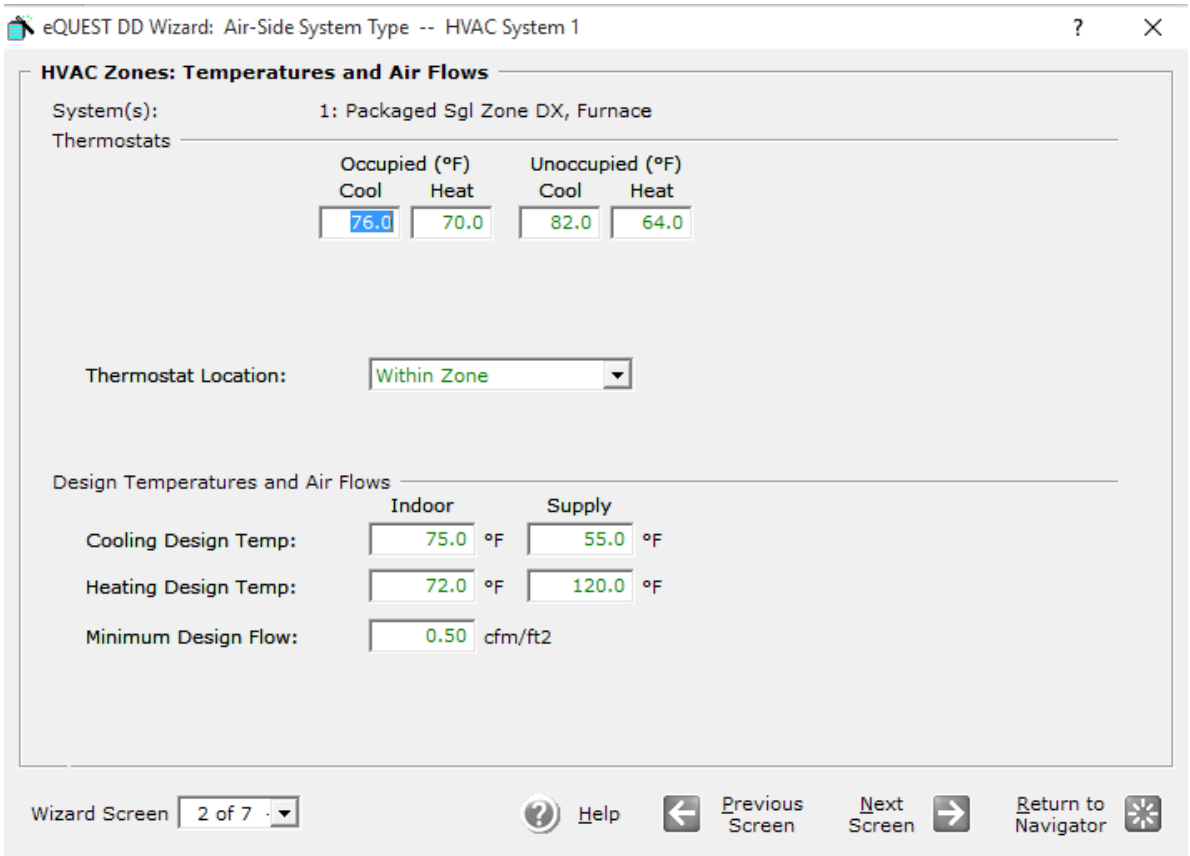
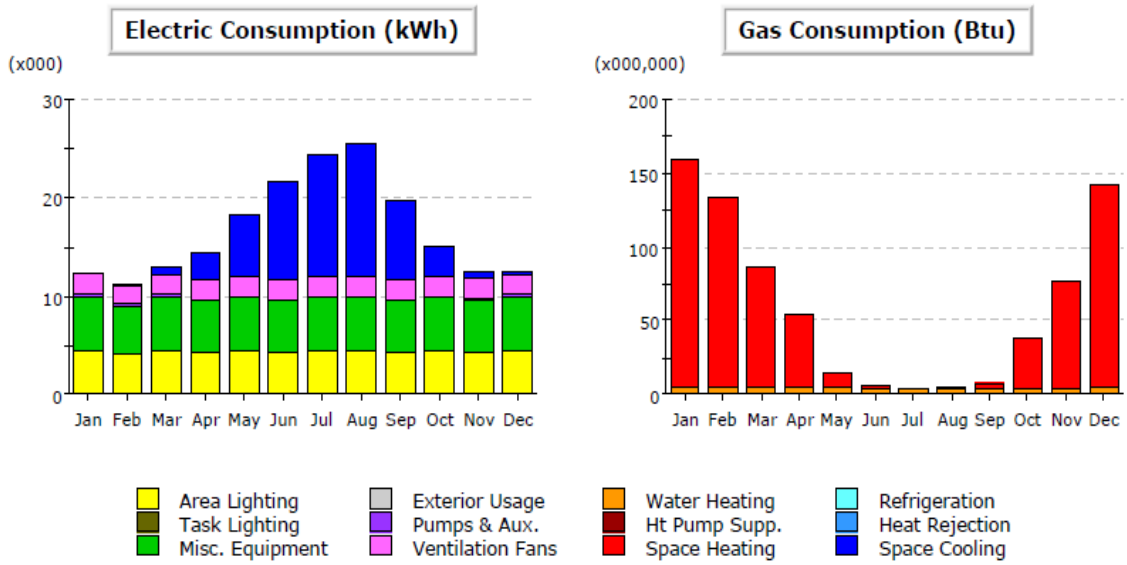


Figure 27: Temperature Set points for Final Proposed Model

With the necessary input provided, the simulation was run and the results were recorded. These are shown below in Figure 28. The results show a reduction in natural gas consumption by 34% over the ASHRAE baseline model. There is also a significant reduction in electric energy use which went down by 28%.

Reduction in natural gas	=	1,097.8 MMBTU – 200.02 MMBTU
	=	370.69 MMBTU ~ 34% less than ASHRAE Baseline
Reduction in electricity	=	278,320 kWh – 200,020 kWh
	=	78,300 kWh ~ 28% less than ASHRAE Baseline



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.08	0.25	0.82	2.71	6.19	9.90	12.40	13.41	8.05	3.10	0.77	0.25	57.94
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	2.10	1.89	2.10	2.03	2.10	2.03	2.10	2.10	2.03	2.10	2.03	2.10	24.68
Pumps & Aux.	0.31	0.23	0.18	0.11	0.02	0.01	-	-	0.00	0.08	0.17	0.27	1.37
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	5.44	4.92	5.44	5.27	5.44	5.27	5.44	5.44	5.27	5.44	5.27	5.44	64.11
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	4.41	3.98	4.41	4.27	4.41	4.27	4.41	4.41	4.27	4.41	4.27	4.41	51.92
Total	12.34	11.27	12.95	14.38	18.16	21.47	24.35	25.36	19.62	15.13	12.51	12.47	200.02

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	155.23	129.56	80.59	49.03	10.62	1.99	0.17	0.60	4.02	33.55	71.37	138.30	675.04
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	4.83	4.50	4.97	4.73	4.58	4.14	4.02	3.86	3.73	4.01	4.14	4.57	52.07
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	160.06	134.05	85.56	53.76	15.19	6.13	4.19	4.46	7.75	37.56	75.51	142.87	727.11

Figure 28: Simulation Results for Final Proposed Model

4.2 Post - Construction Design Models

With the initial data models created, the next step was to recreate the eQuest model with the data gathered during the investigative process. These post construction models were created with some basic assumptions such as:

- The Building dimension were kept at 14,410 sq. ft. of floor area and 12 ft. ceiling height.
- All other physical properties were automatically decided by eQuest for the given input.
- The Temperature set points were considered to be 72°F for cooling and 70°F for heating.
- The solar PV system was not modelled to keep the design simple
- Operating hours of 8 AM to 5 PM
- RTU fans were set to start at 7 AM and shut off at 6 PM
- Setback temperatures were kept at 78°F during summer and 50°F during winter

The first post construction model (hence forth referred to as 'test model') was designed to be noncomplex, intended only to replicate the actual energy use data from the building and provide comparison between different heating solutions. This was instrumental in proving the presence of electric strip heat as primary within the HVAC system.

4.2.1 Initial Verification Models

The initial verification model was created using the same parameters for construction of the building, however modifications were made to the HVAC system to better model the actual building as constructed and completed. The test model underwent several revisions to compare and contrast results with the pre-construction models and actual real life energy use data.

Some of the modeling parameters that were used for the test model can be seen below in Figure 29, Figure 30, and Figure 31. Note the changed fan operation and temperature set points.

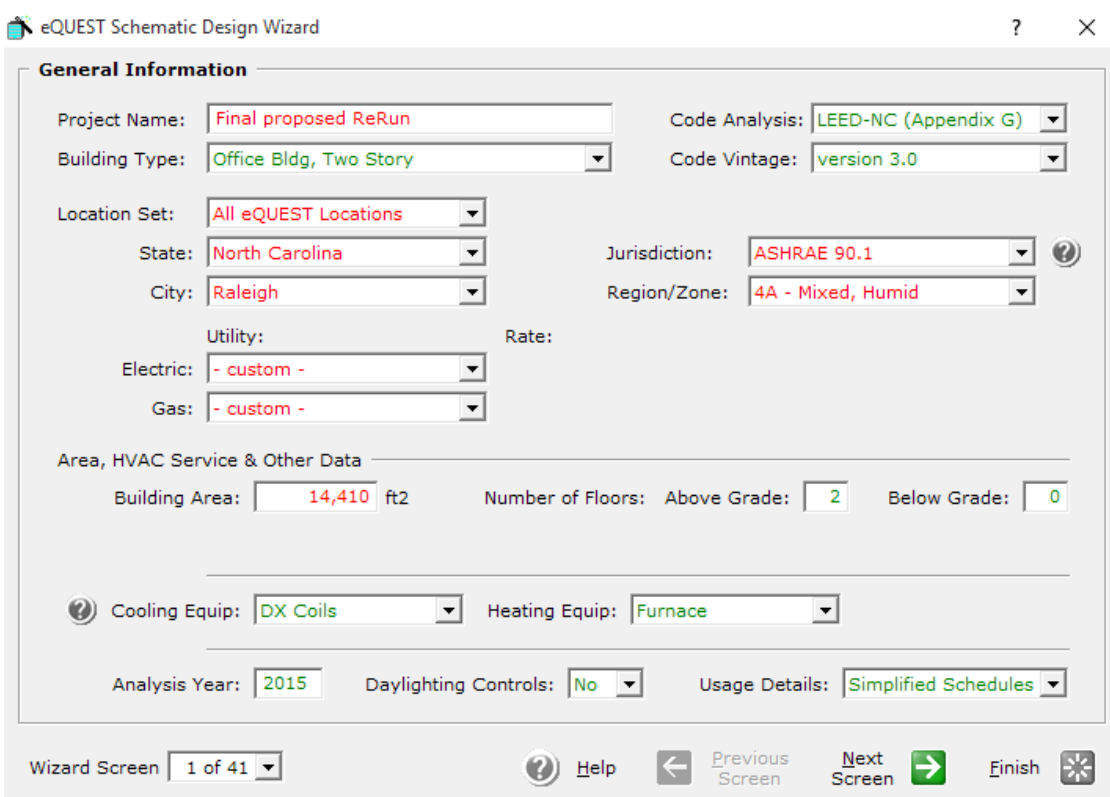


Figure 29: Test Model Basic Input

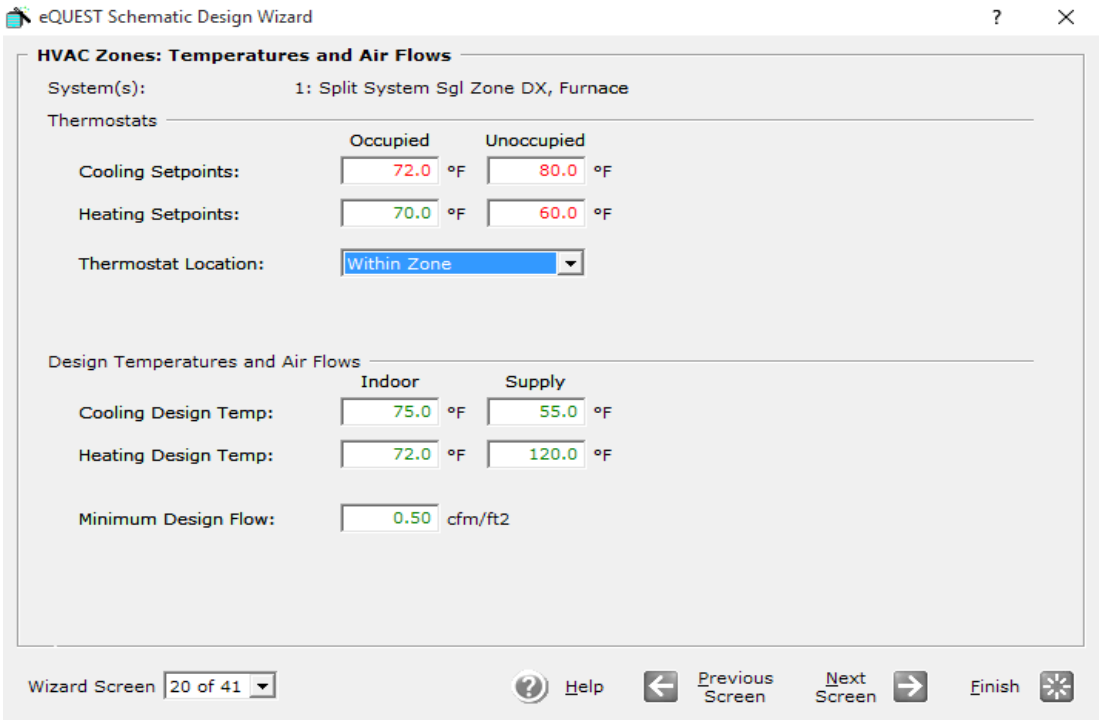


Figure 30: Temperature Settings for Test Models

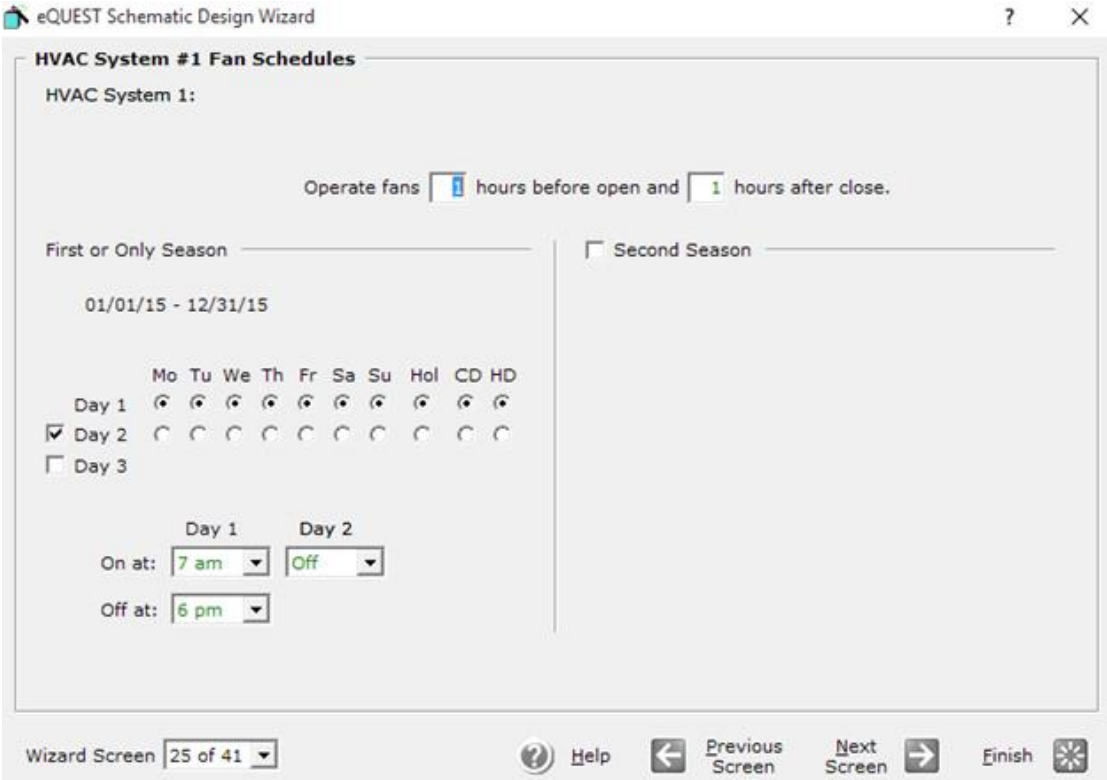


Figure 31: Fan Timings for Test Model

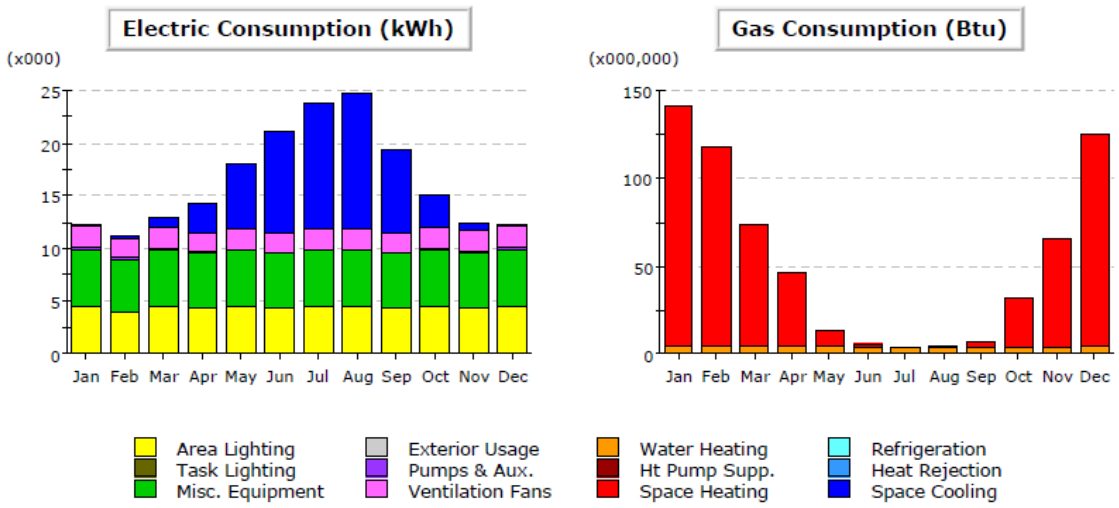
The test model was varied slightly over the course of the modelling process to achieve three different results based on the type of heating. The results for each one are shown below with their accompanying analysis (Figure 32, Figure 33, and Figure 34).

The first test model incorporated standard gas fired furnace heating with no electric strip heating while keeping all other design parameters as mentioned earlier. This model (hereby referred to as test1) was made early on during the analysis when sufficient data was not available regarding the building's performance. The result from this model was used to determine the building's HVAC performance based on the commissioning information which stated the use of a natural gas furnace within the RTU to provide primary heat.

The figure below (Figure 32) show similar numbers compared to the final proposed model. There is a slight difference in the energy use numbers between the two models but the overall trend is nearly same. This proves that our simulation of the building performance is valid. This sets a firm base for us to make changes to the model to simulate different VAV reheat systems.

This model, as predicted shows increased electricity use during the summer for cooling and reduced use during winter when cooling requirement is minimal if not completely absent. On the contrary, natural gas usage peaks during winter and drops off during summer (minimal use for water heating). This is to be expected from a system that uses natural gas furnace for heating instead of electric strip heat.

This model also confirms the results from the ASHRAE baseline model because of similar energy use trends. This implies that the model conforms to the basic system requirements mentioned during the commissioning process i.e. the HVAC system was supposed to use natural gas heating for primary heat.



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.08	0.26	0.84	2.67	6.06	9.55	11.93	12.83	7.84	3.09	0.80	0.26	56.21
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	1.96	1.77	1.96	1.90	1.96	1.90	1.96	1.96	1.90	1.96	1.90	1.96	23.07
Pumps & Aux.	0.31	0.23	0.18	0.11	0.02	0.01	-	-	0.00	0.08	0.17	0.27	1.37
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	5.44	4.92	5.44	5.27	5.44	5.27	5.44	5.44	5.27	5.44	5.27	5.44	64.11
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	4.41	3.98	4.41	4.27	4.41	4.27	4.41	4.41	4.27	4.41	4.27	4.41	51.92
Total	12.20	11.16	12.83	14.21	17.90	20.99	23.75	24.64	19.28	14.99	12.40	12.34	196.67

Gas Consumption (Btu x000,000)

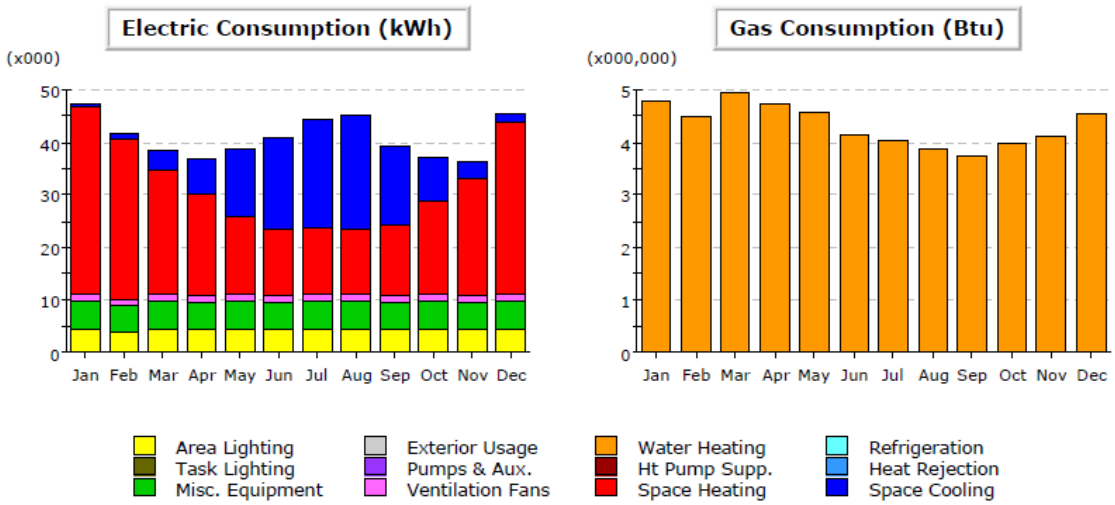
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	136.23	113.46	68.60	41.32	8.19	1.39	0.11	0.39	2.88	27.45	60.79	120.88	581.68
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	4.83	4.49	4.97	4.73	4.57	4.14	4.02	3.86	3.73	4.01	4.13	4.56	52.03
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	141.06	117.95	73.56	46.05	12.77	5.53	4.12	4.24	6.61	31.46	64.92	125.45	633.72

Figure 32: Results for Model Test1 with Gas Heating

The second model test model incorporated electric strip heating only with no natural gas furnace heating while keeping all other design parameters as mentioned earlier. This model (hereby referred to as test2) was made immediately after model test1. Since concrete evidence of building performance was yet to be determined at this time, the model was as of yet still experimental in nature. The result from this model (Figure 33) was used to determine the building's HVAC performance based on the assumption that the system was using electric heat in the VAV terminal boxes.

The results below show very different results when compared to the final proposed model. The difference in electricity use is significantly higher in this case, with winter usage increased by nearly 450%. The summer also sees an increased electricity use, likely due to frequent activation of the strip heaters to provide reheat for dehumidification. Natural gas use also changes significantly. In the previous two cases – test1 and final proposed; we see a marked increase in natural gas use in the winter months which tapers off during the summer months. This does not occur in test2 because natural gas use is restricted to water heating only which remains constant and negligible throughout the year. This model gives us an idea of how the building may perform if electric strips are the only source of heat.

This model has some similarities to the actual usage trends seen earlier (refers to section 3.2). The electricity usage during January is nearly identical. The trend also shows similarity during summer months. Differences arise due to the fact that the actual building also uses natural gas as a secondary source. Natural gas usage is also similar for nearly all months except the first three.



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.70	1.40	3.60	6.68	13.02	17.61	20.66	21.91	15.13	8.38	3.05	1.29	113.42
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	35.55	30.44	23.70	19.49	14.62	12.50	12.50	12.09	13.39	17.75	22.43	32.93	247.40
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	1.13	1.03	1.14	1.10	1.19	1.21	1.27	1.29	1.19	1.19	1.10	1.13	13.96
Pumps & Aux.	0.06	0.05	0.04	0.02	0.00	0.00	-	-	0.00	0.02	0.03	0.05	0.27
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	5.44	4.92	5.44	5.27	5.44	5.27	5.44	5.44	5.27	5.44	5.27	5.44	64.11
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	4.41	3.98	4.41	4.27	4.41	4.27	4.41	4.41	4.27	4.41	4.27	4.41	51.92
Total	47.30	41.82	38.33	36.83	38.68	40.86	44.28	45.14	39.25	37.19	36.15	45.25	491.08

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	4.80	4.47	4.95	4.71	4.57	4.14	4.02	3.86	3.73	4.00	4.11	4.54	51.91
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	4.80	4.47	4.95	4.71	4.57	4.14	4.02	3.86	3.73	4.00	4.11	4.54	51.91

Figure 33: Results for Model Test2 with Electric Strip Heating

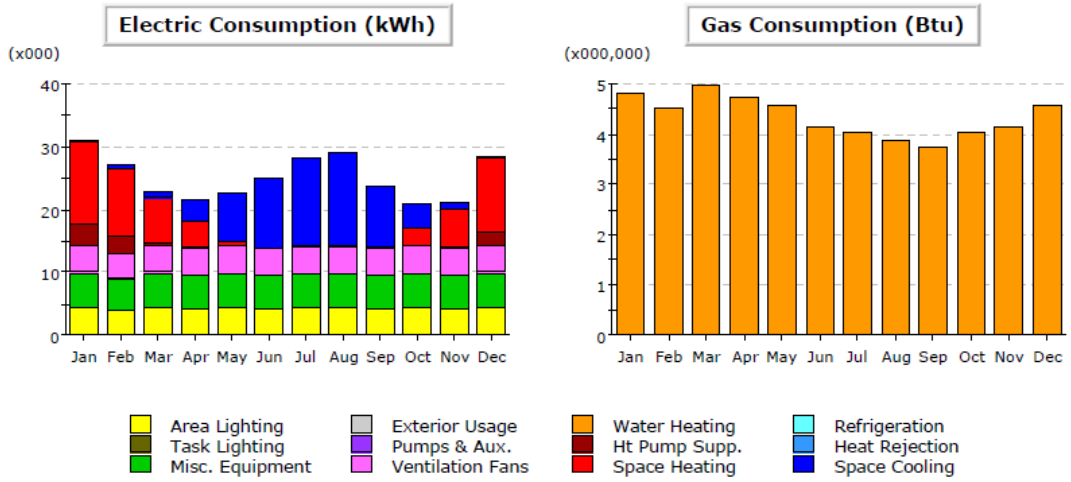
The numbers clearly show that the energy usage for the facility increases by nearly 250% over test model test1 which incorporated gas furnace heating. The analysis of this model will be dealt with in more detail in later sections. These similarities with the actual energy use patterns are enough to paint a basic picture of how the actual system might be performing.

The third model test model incorporated a heat pump within the terminal units while keeping all other design parameters as mentioned earlier. This model (hereby referred to as test3) was made immediately after model test2. By this time, some understanding of the building HVAC performance was achieved hence this model was used to simply understand the functioning of the building in a completely different mode of heating, i.e. with a heat pump.

The rationale behind developing this model (Figure 34) was to compare and show why the heat pump based system would still be preferable over one with electric heat for primary heating only. The model also presented an opportunity to compare the usefulness of such a system in place of a natural gas fire system.

The results below show similar trend compared to the test model test2 which used strip heat, but the heat pump model uses less power. Since a heat pump principally uses electricity as a power source the numbers will be in the same range. The only additional electric load is the heat pump supplement (auxiliary heat) which is small and relatively insignificant.

It is interesting to note that a heat pump uses less electricity for HVAC operation as compared to strip heating. This is because for humid subtropical climate conditions like in Cary, NC where the temperature rarely drops below freezing, a heat pump is more efficient. This is because a heat pump uses a thermos cycle to move heat at COP greater than unity. However, if the temperature does drop below freezing then a heat pump turns on its ‘auxiliary heat’ which is essentially electric strip heating. This may increase the energy usage in such situations. Hence heat pumps are recommended for places with moderate climate and not for colder regions as they cannot perform well at very low temperatures close to freezing.



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.10	0.30	1.00	3.27	7.41	11.29	14.05	14.89	9.51	3.79	0.93	0.32	66.86
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	13.07	10.94	7.03	4.30	0.95	0.18	0.02	0.06	0.36	2.86	6.15	11.77	57.69
HP Supp.	3.34	2.78	0.47	0.20	-	-	-	-	-	0.02	0.26	2.16	9.23
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	4.24	3.83	4.24	4.11	4.24	4.11	4.24	4.24	4.11	4.24	4.11	4.24	49.97
Pumps & Aux.	0.21	0.16	0.14	0.09	0.02	0.01	-	-	0.00	0.07	0.14	0.18	1.01
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	5.44	4.92	5.44	5.27	5.44	5.27	5.44	5.44	5.27	5.44	5.27	5.44	64.11
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	4.41	3.98	4.41	4.27	4.41	4.27	4.41	4.41	4.27	4.41	4.27	4.41	51.92
Total	30.82	26.91	22.73	21.50	22.48	25.11	28.16	29.06	23.52	20.84	21.13	28.53	300.80

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	4.83	4.50	4.97	4.73	4.58	4.14	4.02	3.86	3.73	4.01	4.13	4.57	52.06
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	4.83	4.50	4.97	4.73	4.58	4.14	4.02	3.86	3.73	4.01	4.13	4.57	52.06

Figure 34: Results for Model Test3 with Heat Pump

NOTE: Double peaks are visible during winter and summer months. This is because of air conditioning needs of the building during the summer and heating during winter. Natural gas numbers are still lower compared to test 1.

4.2.2 Optimized Model with Hot Water Loop

The final model was created after analyzing all the previous energy models and concluding that none of the earlier models were able to ensure efficient performance of the VAV system. There were two glaring flaws that became apparent from the previous models.

The first issue is the imbalance in the zone based temperature control. Even though a natural gas fired furnace system would provide necessary heat, it would be unable to provide optimum zone based air conditioning without requiring some form of electric heat as backup. The inner zones or the core zones would naturally be warmer while the outer zones near the walls (perimeter zones) would be colder.

A single RTU can only provide a single stream of conditioned air (usually 55°F) to all the zones and then the electric heaters in the VAV boxes would have to heat the air up to requisite temperatures in each zone individually. Since the interior zones rarely require heating and may require constant cooling, the gas fired heater may not be needed and the cooling system may run year round. This increases the energy required by the HVAC system to condition the space since majority of the spaces are cooled and then subsequently heated.

The perimeter zones are near the exterior walls and by extension on the perimeter of the building. The core zones are on the interior of the building and do not share any external walls. In each of the previous models that employed natural gas heating, accurate temperature control was not possible. This was because under regular operation, the natural gas furnace would provide the requisite heating to raise the temperature to the minimum

acceptable level based on the temperature of air needed by the zone with the lowest heating demand. Under normal operation, the hot air temperature requirement of the supply air for the perimeter zones would be very high, say 100°F.

However, since the inner rooms (core zones) would be better insulated from the cold, their heating requirement would be significantly lower, say 60°F or 70°F. So, the temperature in the inner zones would become too high to be comfortable. To counter this increased temperature in the core zones, the control system of the RTU would have to switch to cooling mode to provide cold air to the rooms in order to bring the temperature down to comfortable levels. Conversely, this action would have the effect of lowering the temperature of the outer perimeter zones to unacceptable levels. Under such conditions, the system would remain in constant state of flux and the cooling system would be competing against the heating system at the same time thus becoming inefficient in the process.

Second, if a pure electric system was used instead of the natural gas furnace, it would be able to provide better zone based heating using the VAV terminal boxes but at the same time drive up the air conditioning costs. The increase in costs can be attributed to the higher cost of electricity versus natural gas. To counter this, an optimized HVAC model was created with the intent of solving both these issues, the system has two main features:

First, the existing system has been split into two separate RTUs such that one services the inner core areas while the other services the outer rooms. Such a design eliminates the first issue of zone imbalance discussed earlier. Since the system has been divided into two, this

gives each RTU the freedom to control the temperature of its respective zone independently. The perimeter zone RTU can operate in heating mode at a higher temperature, and the core zone RTU can operate less aggressively. This allows better temperature control and also reduces energy usage.

Second, it uses natural gas furnace as primary heat and a hot water loop at VAV terminals to provide more efficient heating that electric strip heating. Since water is heated by burning natural gas, energy costs are lowered. Hot water reheat coils tend to last longer than electric strip heaters. Additionally, they do not pose a fire hazard unlike electric heaters.

It is important to note that this model is based off several assumptions as mentioned above. It is also not perfectly indicative of building performance due to the simplicity of design. It is likely that the actual building with such a system may have a higher energy use. This is because the model cannot accurately simulate variations that may occur after construction such as change in temperature set points based on personal comfort or use of personal electric heater by the residents. However, for the purpose of modelling results, this variation is being ignored

Figure 35 and Figure 36 show how the system is designed, the system includes a variable volume, single duct fan/distribution system serving multiple zones each with its own thermostatic control. Each zone is equipped with a variable volume terminal unit and hot water reheat coils.

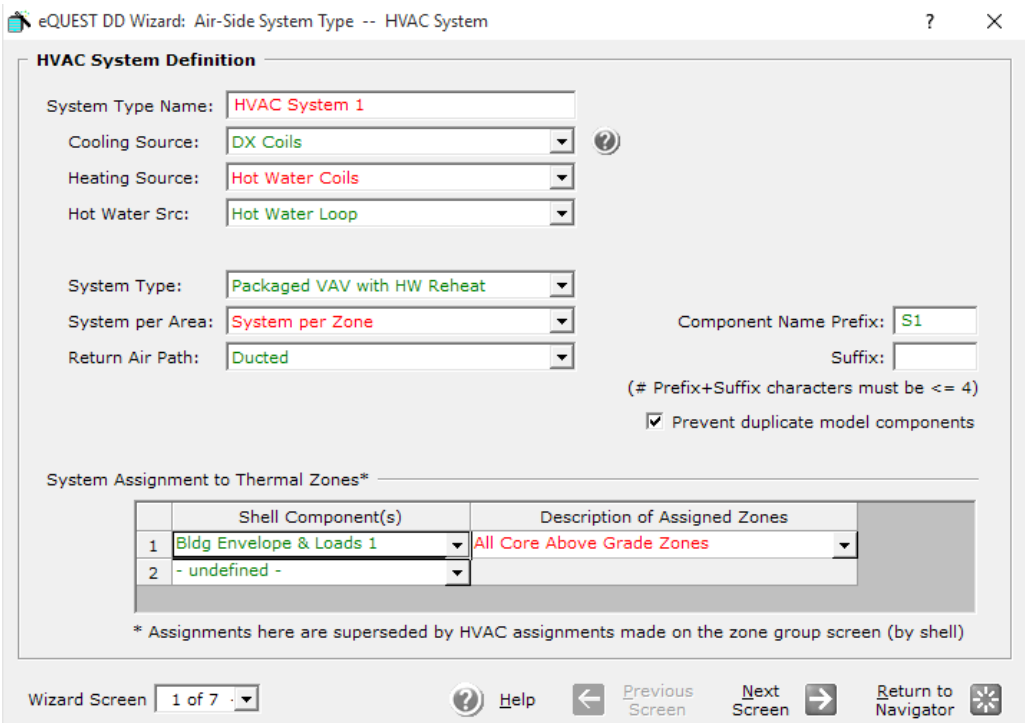


Figure 35: VAV with Hot Water Reheat for Core Zones

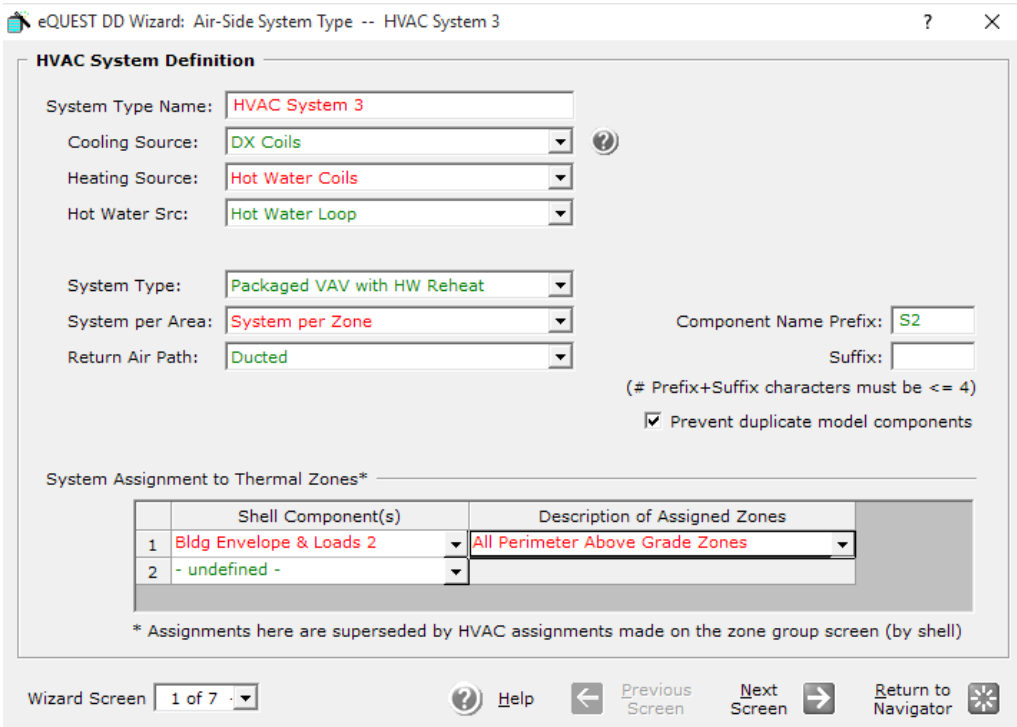


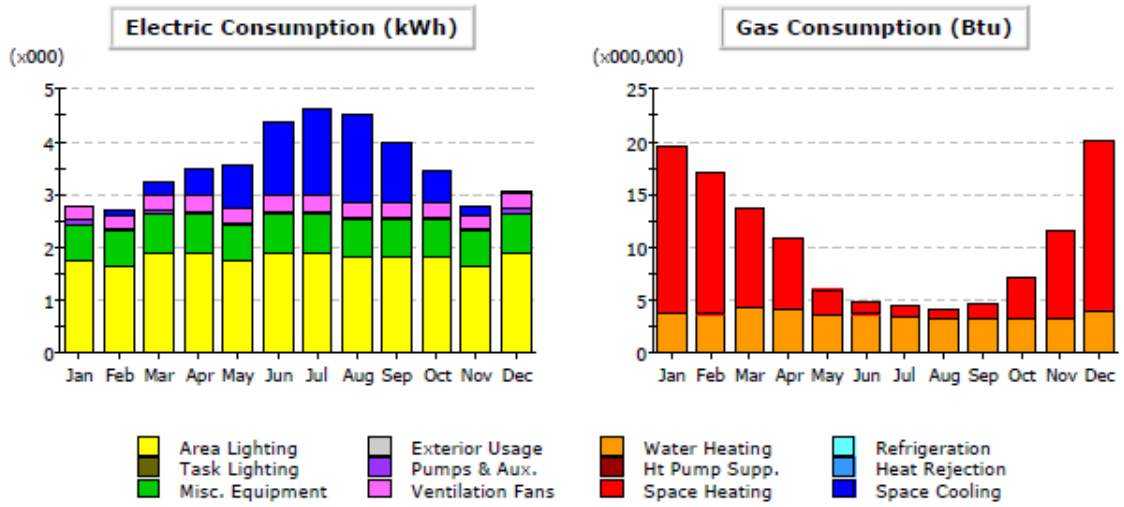
Figure 36: VAV with Hot Water Reheat for Perimeter Zones

The HVAC system shown above is for the core zones only while the HVAC system shown below is for the perimeter zones only. The model incorporated to separate RTUs; one for the core zones and one for the outer perimeter zones. It is expected that such a system would provide more uniform cooling and heating of the building as well as ensure better IAQ as per the ASHRAE 90.1. standards. The simulation results for both the RTU systems can be seen below in Figure 37 and Figure 38.

The results from the simulation appear to be different from each other. This is expected and is preferred. The core zone RTU has lower heating load during the winter due to the fact that the core zones are better insulated from the outside temperatures. The cooling load is also lowered for that same reason.

Conversely, the perimeter zone RTU has a higher heating load due to the fact that the perimeter zones have at least one side exposed to the surroundings. This results in higher ΔTs which causes the furnace to run for more prolonged periods.

Project/Run: optimum model perimeter zones - Baseline Design Run Date/Time: 10/26/15 @ 09:45



Electric Consumption (kWh x000)

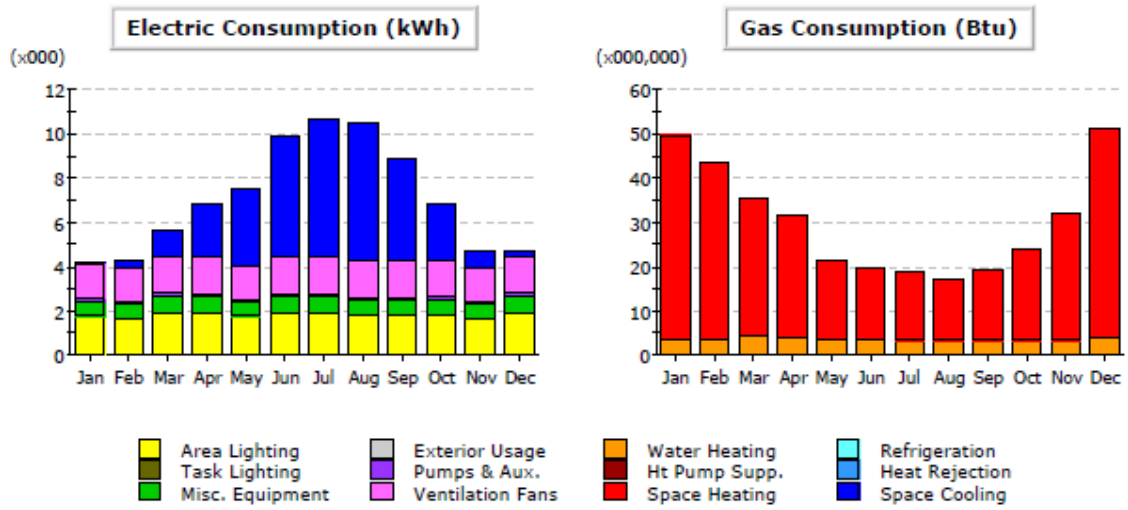
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.01	0.07	0.23	0.52	0.83	1.39	1.66	1.65	1.15	0.58	0.15	0.04	8.29
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	0.28	0.27	0.31	0.31	0.28	0.31	0.31	0.29	0.29	0.29	0.27	0.31	3.51
Pumps & Aux.	0.08	0.07	0.06	0.04	0.03	0.02	0.02	0.02	0.02	0.04	0.05	0.08	0.54
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	0.68	0.64	0.74	0.73	0.68	0.73	0.74	0.71	0.71	0.71	0.65	0.74	8.46
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	1.74	1.65	1.90	1.90	1.74	1.90	1.90	1.82	1.82	1.82	1.65	1.90	21.74
Total	2.80	2.69	3.24	3.51	3.55	4.36	4.63	4.50	3.99	3.44	2.77	3.07	42.54

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	15.61	13.34	9.32	6.79	2.33	1.27	1.04	1.03	1.47	3.82	8.32	16.10	80.45
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	3.84	3.73	4.27	4.17	3.59	3.63	3.42	3.15	3.15	3.30	3.22	3.94	43.42
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	19.45	17.07	13.59	10.96	5.93	4.90	4.46	4.18	4.61	7.12	11.54	20.04	123.86

Figure 37: Simulation Results for Core Zone RTU

Project/Run: optimum model core zones - Baseline Design Run Date/Time: 10/26/15 @ 09:48



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	0.09	0.38	1.17	2.37	3.44	5.42	6.23	6.22	4.64	2.62	0.83	0.22	33.62
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	1.56	1.48	1.72	1.72	1.56	1.72	1.72	1.64	1.64	1.64	1.48	1.72	19.58
Pumps & Aux.	0.14	0.12	0.12	0.11	0.08	0.09	0.08	0.08	0.08	0.10	0.11	0.14	1.24
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	0.68	0.64	0.74	0.73	0.68	0.73	0.74	0.71	0.71	0.71	0.65	0.74	8.46
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	1.74	1.65	1.90	1.90	1.74	1.90	1.90	1.82	1.82	1.82	1.65	1.90	21.74
Total	4.20	4.27	5.65	6.83	7.50	9.85	10.67	10.47	8.88	6.88	4.72	4.72	84.64

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	45.94	39.83	31.08	27.30	17.58	15.97	15.34	13.92	15.97	20.69	28.74	47.13	319.47
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	3.84	3.74	4.29	4.20	3.63	3.67	3.46	3.20	3.19	3.33	3.24	3.95	43.72
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	49.78	43.56	35.37	31.49	21.20	19.64	18.81	17.11	19.15	24.01	31.98	51.08	363.19

Figure 38: Simulation Results for Perimeter Zone RTU

The Figures below (Figure 39 and Figure 40) shows a comparative graph for the two zones.

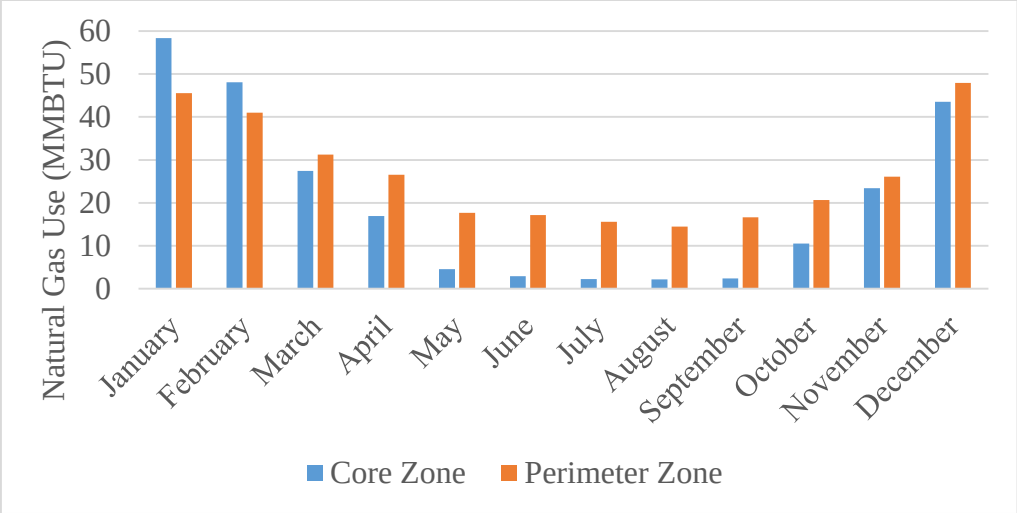


Figure 39: Natural Gas Use – Core Zone vs Perimeter Zone

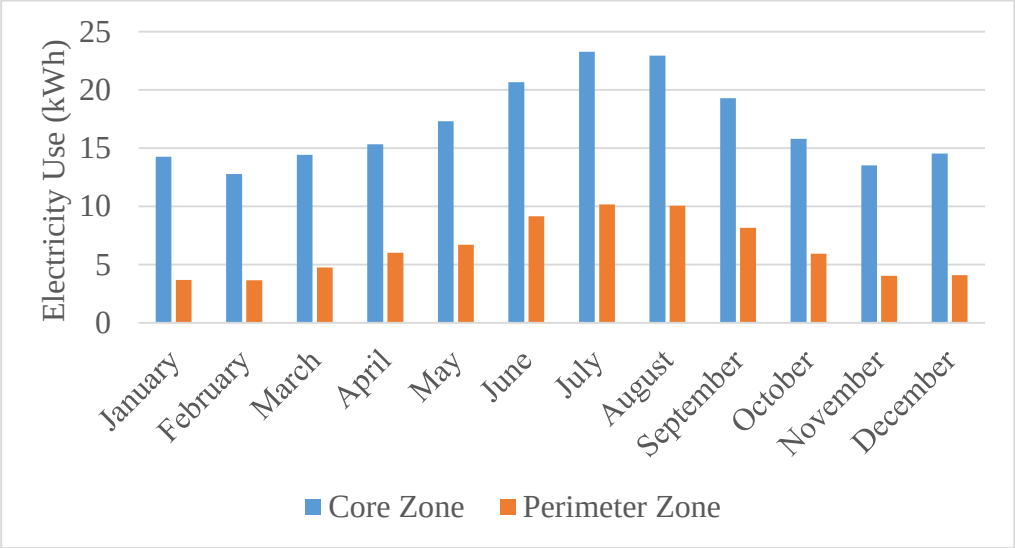


Figure 40: Electricity Use – Core Zone vs Perimeter Zone

Cost savings associated with this model over actual use is calculated using the energy use values for 2014 and the energy use values predicted by the model. The historical energy use and energy costs have been provided in chapter 3.

The costs associated with the actual energy use are:

Current Electricity Cost	=	337,520 kWh x $0.0821/kWh
	=	$27,710/yr.
Current Natural Gas Cost	=	1097.8 MMBTU x $8.97/MMBTU
	=	$2,286/yr.
Current Total Energy Cost	=	$27,710/yr. + $2,286/yr.
	=	$29,996/yr.

Similarly, the energy costs associated with the model are:

Model Electricity Cost	=	127,200 kWh x $0.0821/kWh
	=	$10,443/yr.
Model Natural Gas Cost	=	487.03 MMBTU x $8.97/MMBTU
	=	$4,368/yr.
Current Total Energy Cost	=	$10,443/yr. + $4,368/yr.
	=	$14,811/yr.

Hence the projected cost savings for this model over actual use are:

Projected Cost Savings	=	$29,996/yr. - $14,811/yr.
	=	$15,185/yr.

The greatest hurdle in the realization of this model would be the implementation. A project of this scope would require not only installation of a new second RTU but also involve re-ducting of the entire HVAC system and installation of hot water piping. There would also be an additional cost for controls and setting up of a new hot water loop at the terminal boxes. The cost of such a project is estimated to be around $150,000 including cost of labor. This gives an expected payback of 10 years. This would prove to be cost prohibitive, even with the projected savings of $15,185/yr. over current estimates.

5 RESULTS AND COMPARISON WITH REAL WORLD DATA

The results from the ASHRAE baseline model, final proposed model and the post construction optimized model with hot water loop are compared with real world usage data to determine the performance of the building using each type of HVAC system mentioned above.

It was determined that the actual building Energy Use Intensity (EUI) [16] is approximately 30% lower than ASHRAE code. EUI expresses a building's energy use as a function of its size or other characteristics. For most property types in EUI is expressed as energy per square foot per year. It's calculated by dividing the total energy consumed by the building in one year (measured in kBtu or GJ) by the total gross floor area of the building. However, this does not necessarily dictate the cost per square foot. In fact, the actual cost per square foot does not align with the EUI because of the cost difference between natural gas and electricity.

To calculate the EUI we will first consider the energy use for the year 2014 and compare it with the energy use based on each model. This can be seen in Figure 41 below. Energy use is split into two categories, electricity and natural gas. Table 4 details the baseline ASHRAE data, the proposed model data, the optimized model, the billed electricity, PV production, and the total electricity consumption. The total consumption is calculated by adding the billed electricity to the PV production. Table. 4 shows energy use comparison between the models and real world energy bills.

NOTE: Energy Star [16] ™ recommends using source energy for determining EUI however based on design information used during construction, building energy use was used.

Table 4: Energy Use Comparison for Each model with Actual Data

2014	ASHRAE Baseline Model	Final Proposed Model	Optimized Model (Combined)	Actual Billed Electricity	Actual PV Output	Total Consumption
Month	kWh	kWh	kWh	kWh	kWh	kWh
Jan	19,450	12,340	7,000	44,920	1500	46,420
Feb	17,470	11,270	6,960	27,720	1500	29,220
Mar	19,730	12,950	8,890	24,840	2200	27,040
Apr	20,370	14,380	10,340	20,120	1800	21,920
May	24,120	18,160	11,050	22,840	500	23,340
Jun	27,970	21,470	14,210	24,080	0	24,080
Jul	31,160	24,350	15,300	28,400	50	28,450
Aug	33,210	25,360	14,970	28,480	100	28,580
Sep	25,690	19,620	12,870	32,080	1500	33,580
Oct	20,870	15,130	10,320	23,120	1000	24,120
Nov	18,790	12,510	7,490	30,200	0	30,200
Dec	19,490	12,470	7,800	30,720	100	30,820
Total	278,320	200,020	127,200	337,520	10,250	347,770

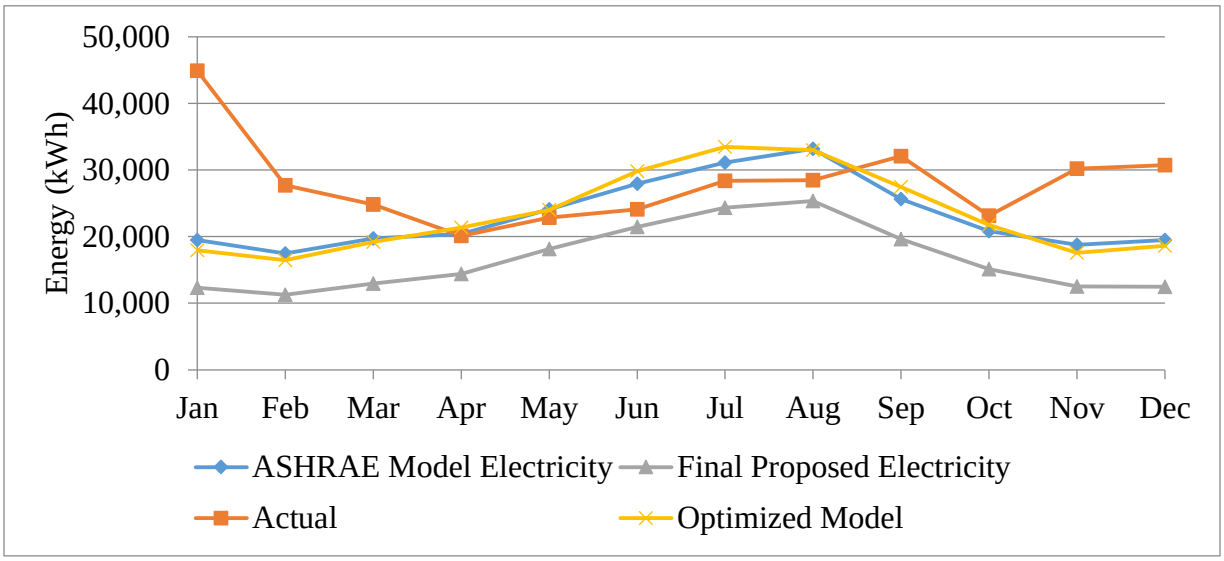


Figure 41: Electricity Use Comparison of Models with Actual Data

Figure 42 clearly demonstrates that the final proposed model does indeed consume minimum electricity, the optimized model also consumes nearly 70% less electrical energy than actual building. This fairly significant reduction in the optimized model is likely due to better handling of air loads and tighter temperature control of each zone.

Table 5 details the ASHRAE baseline, the final proposed model natural gas consumption, the optimized model natural gas consumption, and the actual consumption for the year 2014.

Table 5: Natural Gas Use

Year	ASHRAE Baseline	Final Proposed	Optimized Model	Actual
2014	MMBTU	MMBTU	MMBTU	MMBTU
Jan	223.8	160.06	69.23	70.2
Feb	185.9	134.35	60.63	65.8
Mar	124	85.56	48.96	36.2
Apr	86.1	53.76	42.45	11.2
May	34.5	15.19	27.13	6.5
Jun	17.7	6.13	24.54	11.7
Jul	11.5	4.19	23.27	7.5
Aug	13.4	4.46	21.29	8.2
Sep	20.8	7.75	23.76	7.7
Oct	67.8	37.56	31.13	8.3
Nov	117.7	75.27	43.52	8.5
Dec	194.8	142.87	71.12	13.1
Total	1,097.80	727.11	487.03	254.9

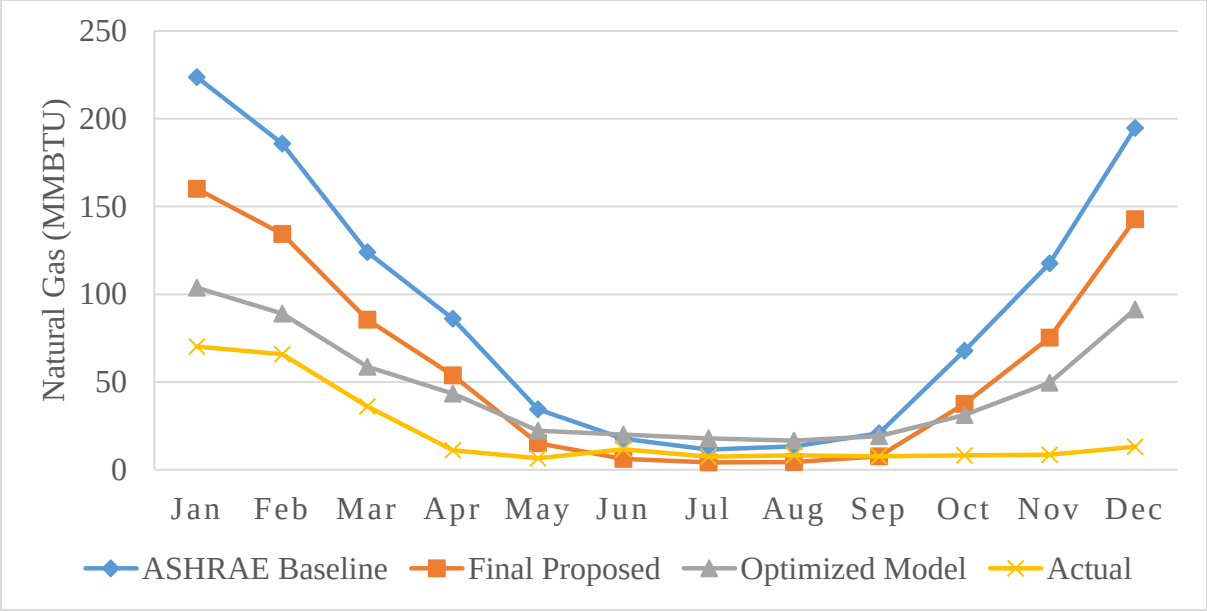


Figure 42: Natural Gas Use Comparison of Models with Actual Data

Figure 42 shows that the optimized model consumes less natural gas than both the final proposed model and the ASHRAE baseline model. While the use as compared to real world bills is more because the actual design uses electric strip heat.

This leads us to the Energy Use Intensity (EUI) calculations for this facility. The building baseline energy use was based upon ASHRAE 90.1 -2007 code, and then improved on for 31% better performance for the final proposed model. The optimized model improved upon the ASHRAE baseline by 55%. The total (natural gas and electricity) modeled energy use for the facility, in kBtu/yr. is calculated below.

Energy use Intensity

For ASHRAE baseline Model:

Proposed Model Use	=	Modeled Electric Use + Modeled Nat Gas Use
	=	278,250 kWh/yr. x 3.413 kBtu/kWh + 1,097 kBtu/yr.
	=	2,046,667 kBtu/yr.

Therefore, the total proposed EUI is:

Proposed Model EUI	=	2,046,667 kBtu/yr. / 14,410 sqft
	=	142 kBtu/sqft –yr.

For Final Proposed Model:

Proposed Model Use	=	Modeled Electric Use + Modeled Nat Gas Use
	=	200,020 kWh/yr. x 3.413 kBtu/kWh + 727,110 kBtu/yr.
	=	1,409,778 kBtu/yr.

Therefore, the total proposed EUI is:

Proposed Model EUI	=	1,409,778 kBtu/yr. / 14,410 sqft
	=	98 kBtu/sqft –yr.

For Optimized Model:

Optimized Model Use	=	Modeled Electric Use + Modeled Nat Gas Use
	=	127,200 kWh/yr. x 3.413 kBtu/kWh + 487,030 kBtu/yr.
	=	921,163 kBtu/yr.

Therefore, the total proposed EUI is:

Optimized Model EUI	=	921,163 kBtu/yr. / 14,410 sqft
	=	64 kBtu/sqft –yr.

For Actual Energy Used:

Total Actual Energy	=	Actual Electricity Use+ Actual Natural Gas Use
	=	347,770 kWh/yr. x 3.413 kBtu/kWh + 254,900 kBtu/yr.
	=	1,441,839 kBtu/yr.

Thus, the Total Actual EUI is:

Total Actual EUI	=	1,441,839 kBtu/yr. / 14,410 sqft
	=	100 kBtu/sqft – yr.

Based on the calculations above, the ASHRAE Model is supposed to have an EUI of 142 kBtu/sqft – yr. This implies that the final proposed design is 31% more efficient than the original baseline ASHRAE model. Additionally, the Optimized model is 55% more efficient than the ASHRAE model.

Final Proposed EUI Change	=	(142 – 98) / 142 x 100
	=	31% better
Optimized Model EUI Change	=	(142 – 64) / 142 x 100
	=	55% better
Actual Energy Use EUI Change	=	(142 – 100) / 142 x 100
	=	30% better

This means that the actual building performance is approximately 30% better than if designed with ASHRAE 90.1 – 2007 code. But the actual performance is still not as efficient as if it were based on the Optimized model with hot water loop. The actual model is in fact 56% less efficient than it. This difference may be slightly exaggerated due to difference in model design and actual operation however, it is likely that even as a conservative estimate the optimized model would still be at least 25% more efficient.

Actual Energy Use EUI vs Optimized Model	=	(64 – 100) / 64 x 100
	=	-56 % (worse)

Table 6 summarizes the EUI values for the categories of total proposed model EUI, ASHRAE code model EUI, and Total Actual EUI.

Table 6: EUI Comparison

	Energy Use Intensity (kBtu/sqft-yr.)	Improvement Compared to Code
ASHRAE Baseline	142	-
Proposed Model	98	34%
Optimized Model	64	55%
Actual	100	30%

The actual EUI performance (30% improvement) is lower than the proposed model's 34% and the optimized model's 55% improvement compared to ASHRAE code respectively. This can be explained in two ways. First, the model did not factor in the solar hot water heating when calculating natural gas consumption. Additionally, the model assumes the natural gas heating instead of electricity. Natural gas heating ensures burner efficiency losses close to 20% (depending on type of water heater). Additionally, there are other systems in the building that may influence overall energy use. For example, lighting may consume fewer kWh than expected.

However, EUI does not tell the whole story. In the next section we will look at cost per square foot for the ASHRAE model, the proposed model, the optimized model and the actual usage.

Cost Per Square Foot

The cost per square foot is based on the mixture of natural gas use and electricity use. If electricity is used for heating, the facility will incur higher energy costs, as electricity is 2 to 3 times more expensive for heating.

Table 7 summarized the cost per square foot for the proposed model, the code model, and the actual energy use. Average energy rates for electricity and natural gas were used to calculate costs.

Table 7: Overall Energy Cost Per Square Foot

	Cost per Square Foot ($/sqft)	Improvement Compared to Code
ASHRAE Code	1.65	-
Proposed Model	1.08	35%
Optimized Model	1.03	38%
Actual	1.98	-20%

Despite the better than expected Actual EUI compared to the ASHRAE Code EUI, the actual cost per square foot is a 20% increase as compared to the cost per square foot with the building designed to the code. Worse still, it is 88% more expensive compared to the optimized model.

Carbon Footprint

The Carbon footprint of a building can be estimated based on its CO_2 emissions (tons/yr.). According to the Environmental Protection Agency (EPA) [17], North Carolina as a state has the 14th highest carbon footprint in the USA. In 2014, North Carolina emitted 123 million metric tons of CO_2. It can be calculated based on the metrics provided by EPA eGRID [18] database. The Emissions & Generation Resource Integrated Database (eGRID) is a comprehensive source of data on the environmental characteristics of almost all electric power generated in the United States. Based on the model data and actual data, we can tabulate the emission values as follows (Table 8):

Table 8: Overall Greenhouse Gas Emissions Per Year

	Natural Gas GHG (tons CO2/yr.)	Electricity GHG (tons CO2/yr.)	Total (tons CO2/yr.)	% Improvement from ASHRAE	Blended Energy Footprint (MMBTU/yr.)
AHSRAE	64	149	214	-	2051
Proposed	42	98	140	34%	1348
Optimized	28	68	97	55%	921
Actual	15	181	196	8%	1441

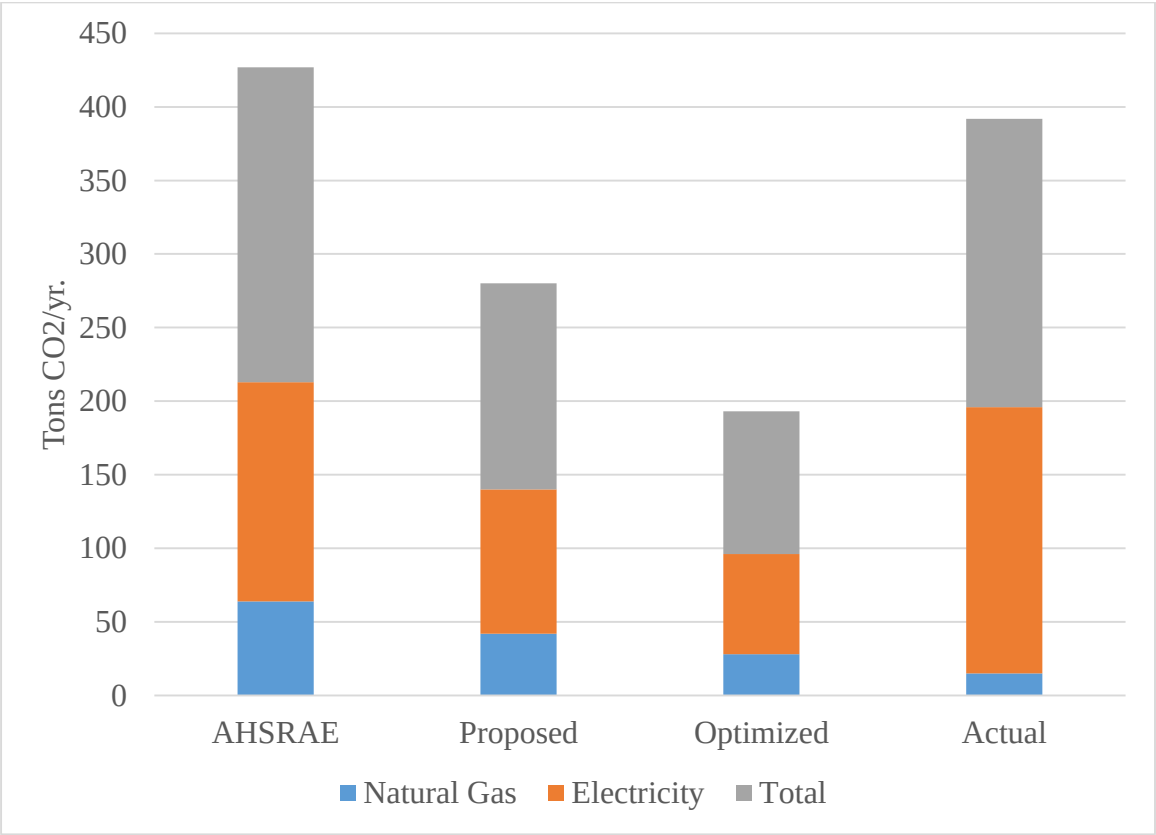


Figure 43: CO_2 Emissions Comparison of Models with Actual Data

Despite having lower actual CO_2 emissions compared to the ASHRAE baseline model (8% lower), it is 40% higher than final proposed model and a staggering 105% more than the optimized model. This is true because the actual system uses far more electricity than natural gas. The process of generation of electricity is highly inefficient and only 30% of the input fuel energy is converted to electric energy, the rest is lost as heat. In case of natural gas, the energy conversion efficiency is nearly 80%, hence a higher electricity use may make a system look efficient in terms of kWh usage but increases the cost per ft^2 and CO emissions.

Based purely on blended electricity and natural gas kBtu/yr., the facility is operating with a 30% improvement over the code. But this comes with a caveat. The cost per square foot does not align with the EUI improvement because electric heat is much costlier than gas heat.

Finally, in terms of energy costs, the actual energy costs are 8% lower than ASHRAE baseline, but 30% higher than final proposed model and 102% higher than the optimized model. The primary cause behind this being the cost of natural gas being lower than electricity. Table 9 shows this in detail.

Table 9: Energy Cost Comparison

	Natural Gas Cost ($/yr.)	Electricity Cost ($/yr.)	Total Cost ($/yr.)	% Improvement from ASHRAE
AHSRAE	$9,847	$22,850	$32,697	-
Proposed	$16,422	$6,522	$22,944	29%
Optimized	$10,443	$4,369	$14,812	55%
Actual	$27,710	$2,286	$29,997	8%

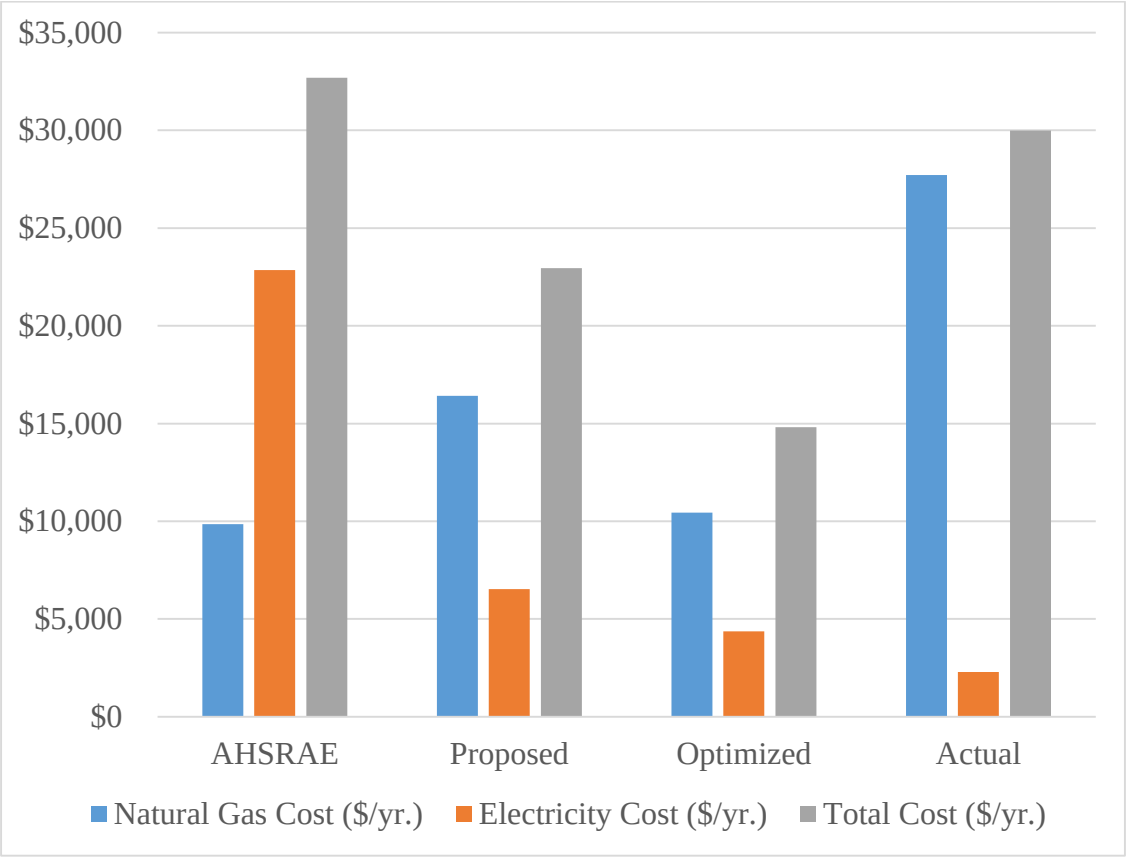


Figure 44: Energy Cost Comparison of Models vs Actual Use

Figure 44 encapsulates the comparison of costs between the models and actual use detailed in Table. 9. Note that despite the lower energy use and EUI, the energy costs for the actual HVAC system are not much lower than the ASHRAE baseline.

5.1 Additional Recommendations

In View of improving energy efficiency and overall energy use of the building, other minor recommendations are compiled in this section. These recommendations may have low or no energy savings potential but are easier to implement and verify.

5.1.1 Ensuring that thermostats are set at 72°F

The building has thermostatic control provided in each of the rooms to allow the residents to change the temperature at will. It is recommended that the thermostats be maintained at 72°F during the summer months to ensure occupant comfort and energy efficiency. It has been observed that the thermostats are kept at very low temperatures, about 68°F by the occupants during the summer months. This is a personal preference of the occupants for their own comfort however it increases the energy usage of the building.

In order to gauge the difference in performance of the HVAC system by changing the set point from 68°F to 72°F, a simple eQuest model (separate from the detailed models discussed earlier) was created to reflect the changes suggested. The objective of this model was to simply show a decrease in energy use by employing proper thermostatic control.

The facility was modeled using a floor area of 14,410 ft^2. Weather data for Raleigh, NC was used for the simulation. A one shift operation (8 am – 5 pm) was assumed. The model replicated the thermostat condition and on weekends, the space was considered unoccupied. Table 10 below shows the proposed energy usage and cost savings for the current model and

the model with changed temperature set points. The months of June to September were considered as they represent the month with highest cooling requirements.

Table 10: Expected Energy Savings from Thermostat Control

Month	Cooling Load (kWh)		Savings	
	Existing	Proposed	KWH	Cost Savings
June	10,290	9,475	815	$67
July	13,017	12,075	942	$77
August	14,961	13,922	1,039	$85
September	8,023	7,598	425	$35
Total	46,291	43,070	3,221	$264

The expected energy savings due to this energy conservation measure will be:

Cost Savings = 3,221 kWh/yr. x $0.0821/kWh
= $264/yr.

Figure 45 illustrates the difference in energy use in the two cases.

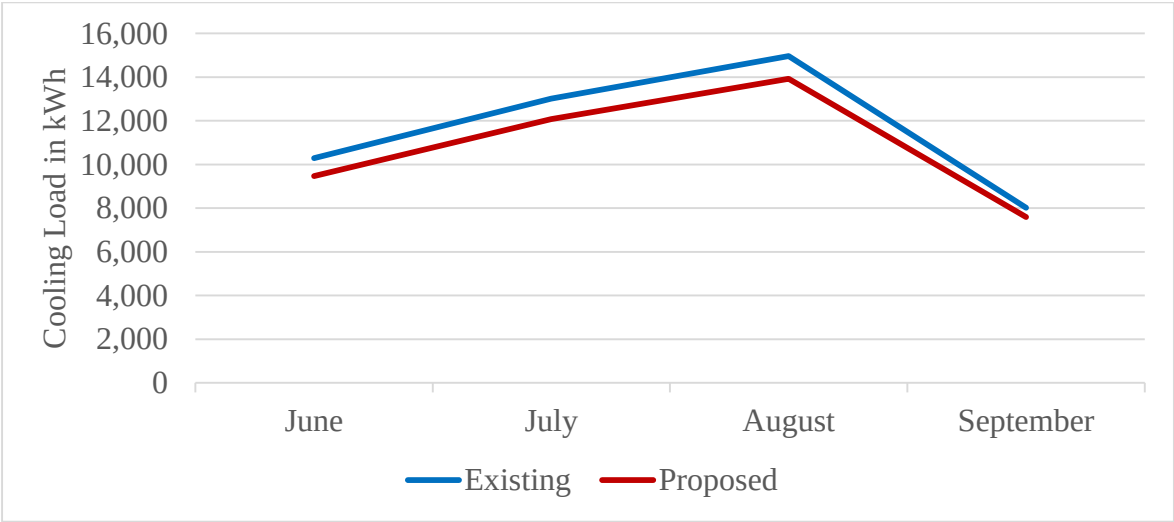


Figure 45: Change in Energy Use due to Thermostatic Control

The implementation for such a change would require a simple setting change within the thermostats. Following this suggestion, the changes were implemented. The old thermostat and new thermostat readings can be seen below in Figure 46.

Figure 46: Comparison of the Old Thermostat Setting vs New Thermostat Setting

5.1.2 Remove strip heat in stairwell

There are two small wall mounted electric heaters placed in the two stairwells of the building. The two heaters have been provided to maintain similar temperatures in the stairwells as the inner rooms to make the entire building more comfortable for the occupants. However, the stairwells are rarely occupied and heating the space using the electric heaters is both unnecessary and wasteful. It is recommended that the small electric heaters in the stairwell be removed.

Based on the specifications available online [19], each of the two heaters are 1.5 kW each and operate for 8 hours per day, 5 days a week during the winter months of December to

February. This total to approximately 500 hours a year. The energy savings expected from their removal is:

Current Energy use	=	2 x 1.5 kW x 500 hr./yr.
	=	1,500 kWh/yr.
Expected Energy Savings	=	1,500 kWh/yr. x $ 0.0821/kWh
	=	$123/yr.

Although the energy savings and cost savings may appear small, the implementation cost is virtually zero hence the payback for such an energy conservation measure (ECM) is minimal. The wall heaters being used in the building are shown below (Figure 47).

Figure 47: Wall Heater in the Stairwell

5.1.3 Improve Building Performance Data Reporting

The building employs data monitoring system to get real-time data about the building's energy performance.. The software has been created to monitor all the electrical subsystems within the building, generate real time data and publish energy usage reports. The system in use is robust and useful but it does have a few shortcomings. The user interface has some issues with report creation and data presentation. This issue can be seen below in Figure 48.

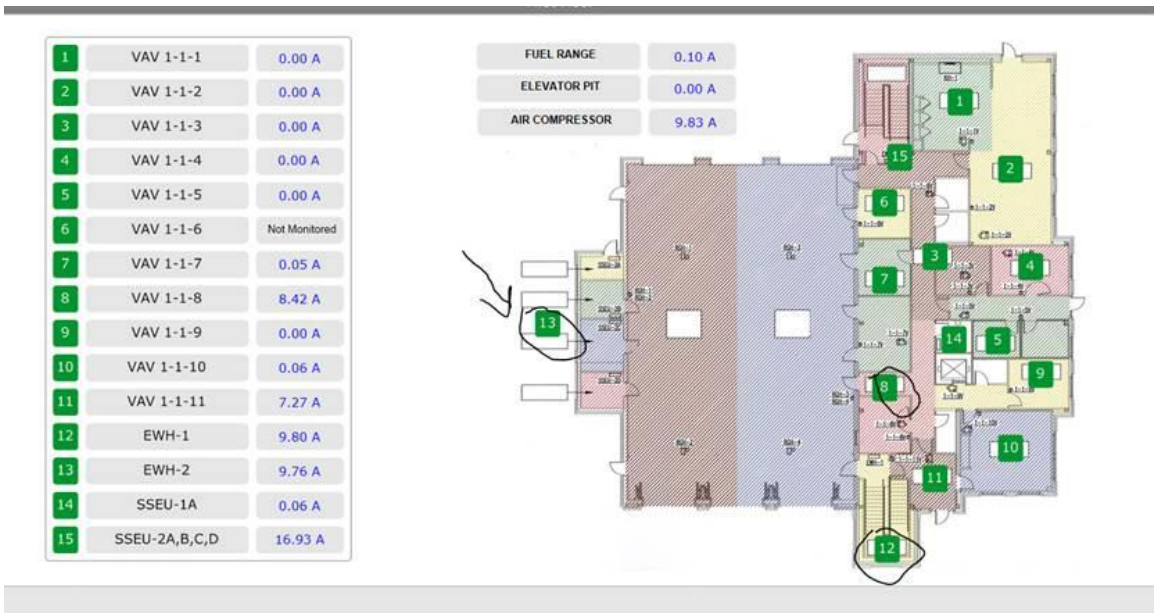


Figure 48: Ion Enterprise UI with Missing Hyperlinks

The links in all of the green boxes (note circles above) are supposed to be hyperlinked to new views but the hyperlinks do not work. This creates an issue while attempting to access different areas within the building within the software. It also results in broken links that give incorrect readings.

Another issue with the software is the lack of customized information for the end user. The naming schemes used to represent various areas within the facility in the electrical map are

not user friendly and can only be understood by a legacy (expert) user. In order to make it more user friendly, it is recommended that these names be changed to reflect the actual locations instead of code words. Figure 49 illustrates this point.

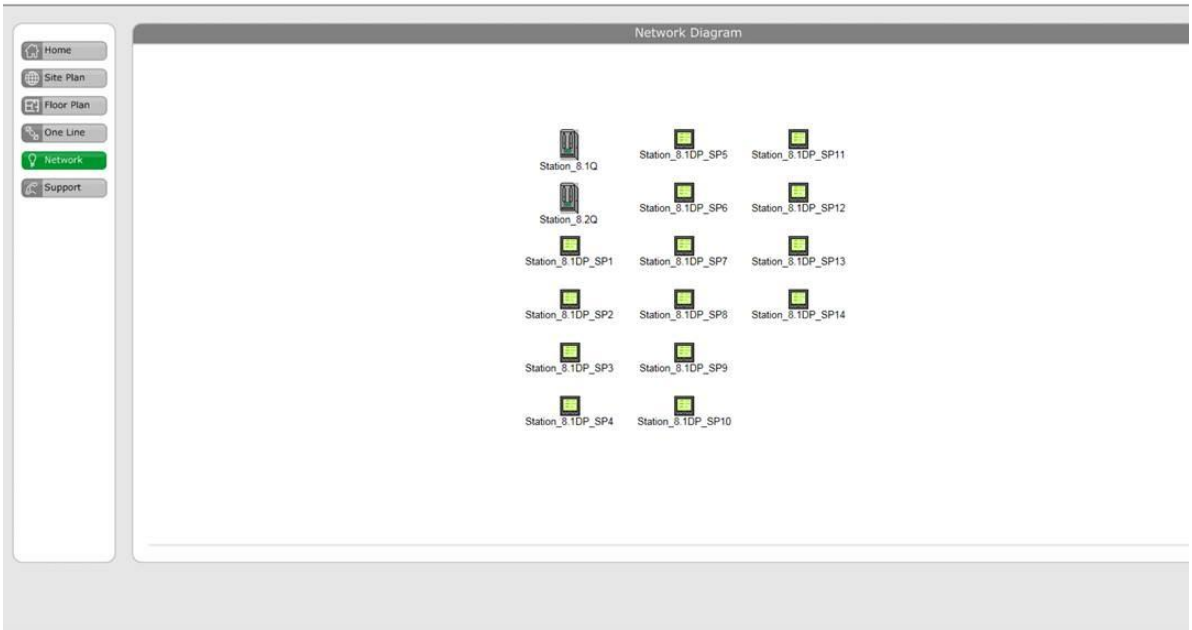


Figure 49: Naming Scheme for Electrical Circuits for Each Area in the Facility

The implementation of a new and improved Ion Enterprise Version 8 is underway and it should eliminate the problems mentioned above. Additionally, it is recommended that proper training be provided to end users of the system so that they may be able to use the new system more efficiently and use its robust features to help monitor and improve energy usage of the building.

5.2 Lessons Learned

The entire project required a concerted effort from day one. During the course of the project it became apparent that even though a system may be designed in a certain way, it is not guaranteed to work in the same way. This can be because of many reasons including change in operational parameters during actual use, increase occupant density than predicted etc.

While software modelling of a building is essential from an engineering point of view, it does not completely encapsulate the construction process. The process of commissioning is an equally important aspect of building construction. Through the course of the actual project multiple design changes are suggested that meet the performance requirements of the building but must also be cost effective. In such cases, it likely that the team designing and constructing the building might opt for “equivalency” which means the use of a material or equipment that performs in equal capacity to the material or equipment stated in the tender but at a much lower cost. This may result in designs that look good on paper but may have adverse effect on the performance of the building in the long run. In the case of Fire Station 8, the use of an electric strip heater was a consequence of such a design choice.

The entire project was also valuable in providing more in-depth understanding of the entire design and analysis process. The importance of having separate RTUs to cater to the inner and outer rooms cannot be emphasized enough. Most importantly, it brought forth the importance of using correct fuel for heating based on not just energy use but also energy cost i.e. natural gas instead of electricity.

6 CONCLUSION

From the analysis in the previous sections it can be understood that while the HVAC system underwent multiple design iterations, it could never perform optimally. The actual building, though efficient was not able to lower its EUI to the lowest possible level. The ASHRAE baseline model presented a useful baseline to develop the design. The final proposed model improved upon the design by making changes to the heating system in order to better capitalize on the low cost furnace heat over the more expensive electric heat options. However, both the models failed to produce optimum results.

The analysis of the building also led to the creation of multiple ‘post construction’ energy models that were used to test the validity of the pre-construction models and demonstrate the benefits and drawbacks of the different types of heating methods in the HVAC system such as furnace, electric strip heat and heat pumps.

The end result from the analysis proved that a two zone HVAC system with two separate RTUs was the optimal solution for the building. Such a system would have one RTU dedicated to serve the inner core zones and the other would serve the perimeter zones. This system would benefit from reduced HVAC load on each individual system and provide correct temperature air to each of the two zones. Since the RTUs would be independent of each other, the problem of constant switching between heating and cooling modes present in a single RTU would be avoided.

The analysis and simulation of this model also proved that such a design would have a much lower EUI of 64 as compared to 142 for the ASHRAE baseline and lower than the actual EUI of 100. The optimized model utilized a hot water loop which enabled it lower energy costs by 50% over the actual use data and 55% from the ASHRAE baseline model. The carbon footprint also reduced by 55% from the ASHRAE baseline and 50% from the actual use data. Finally, the cost of energy per square foot was lowered by 38% from ASHRAE code and 88% from actual use data.

The optimized model predicts that the HVAC with such a design would perform efficiently for the building and produce energy savings worth $15,185/yr. but with an implementation cost of $150,000, such a system would prove too costly for a retrofit. However, it can be considered as a viable option for any fire stations or mixed use buildings that may be constructed at a later date under consideration

7 FUTURE WORK

This project was conducted in part to analyzed suggested solutions for some of the issues that the Town of Cary fire station 8 is facing. Pursuing some of the ideas recommended in this report will help to increase the energy efficiency of the facility. Additionally, researching alternative options could be beneficial in helping the facility resolve some of their problems.

Going beyond this report, it is recommended that the Town of Cary managers work with other external companies to determine the most cost effective options to resolve the problems at hand. While the HVAC system for this building may not be changed as it would be cost prohibitive, lessons learned from the project should be used to design and develop future fire stations in the city.

This project lays the groundwork for more detailed study of similar VAV based systems for use in mixed use buildings. The results from the analysis may be used as basis for analyzing and understanding similar type buildings across North Carolina and maybe even the entire country. VAV systems are suitable choice for larger buildings with multiple floors and rooms. Their usefulness comes in to question with smaller buildings though. Proper controls and efficient HVAC system design is essential in ensuring that the system performs as per requirement. Hence a lot more research and case studies would be required to ensure that VAV systems are better understood and correctly implemented

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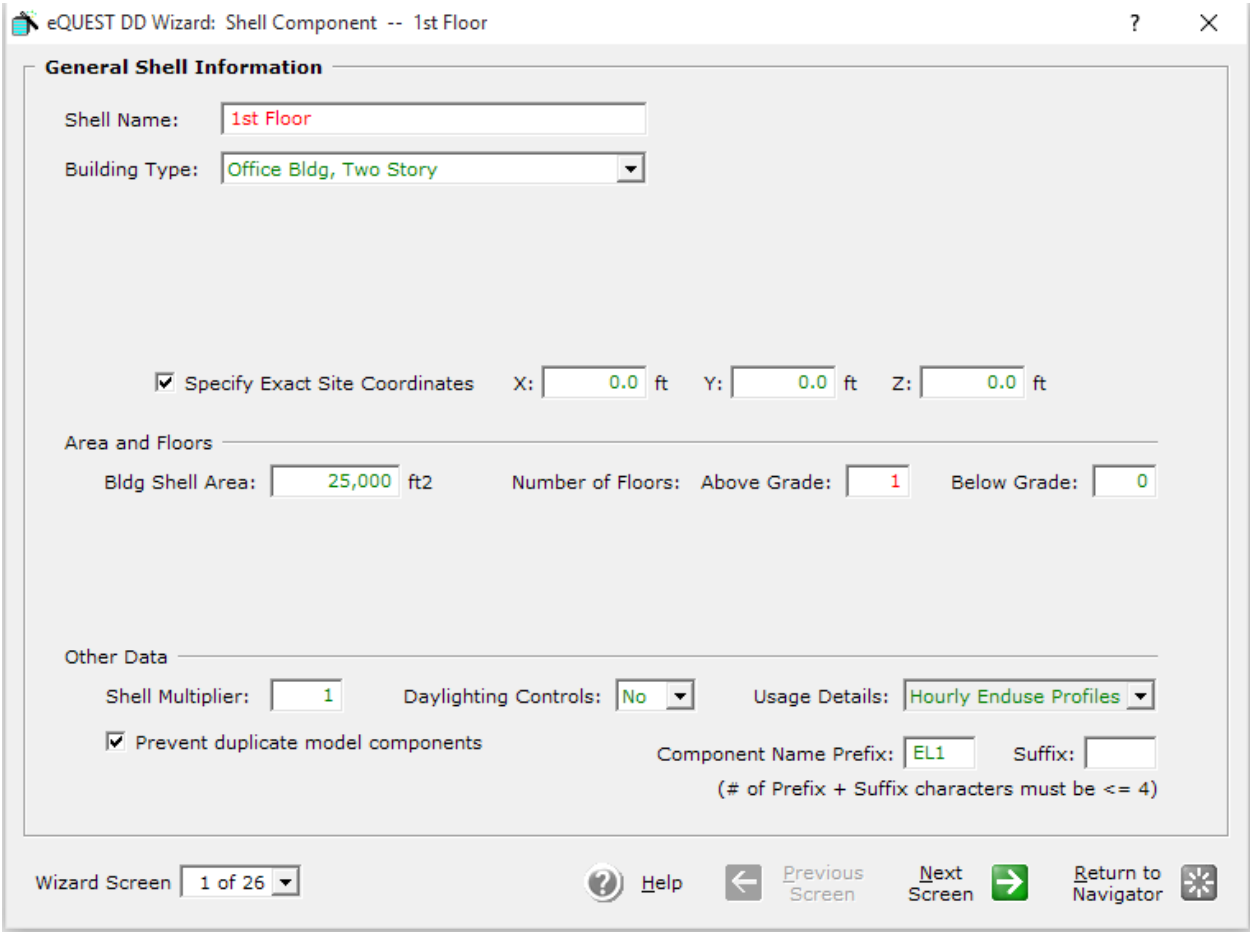
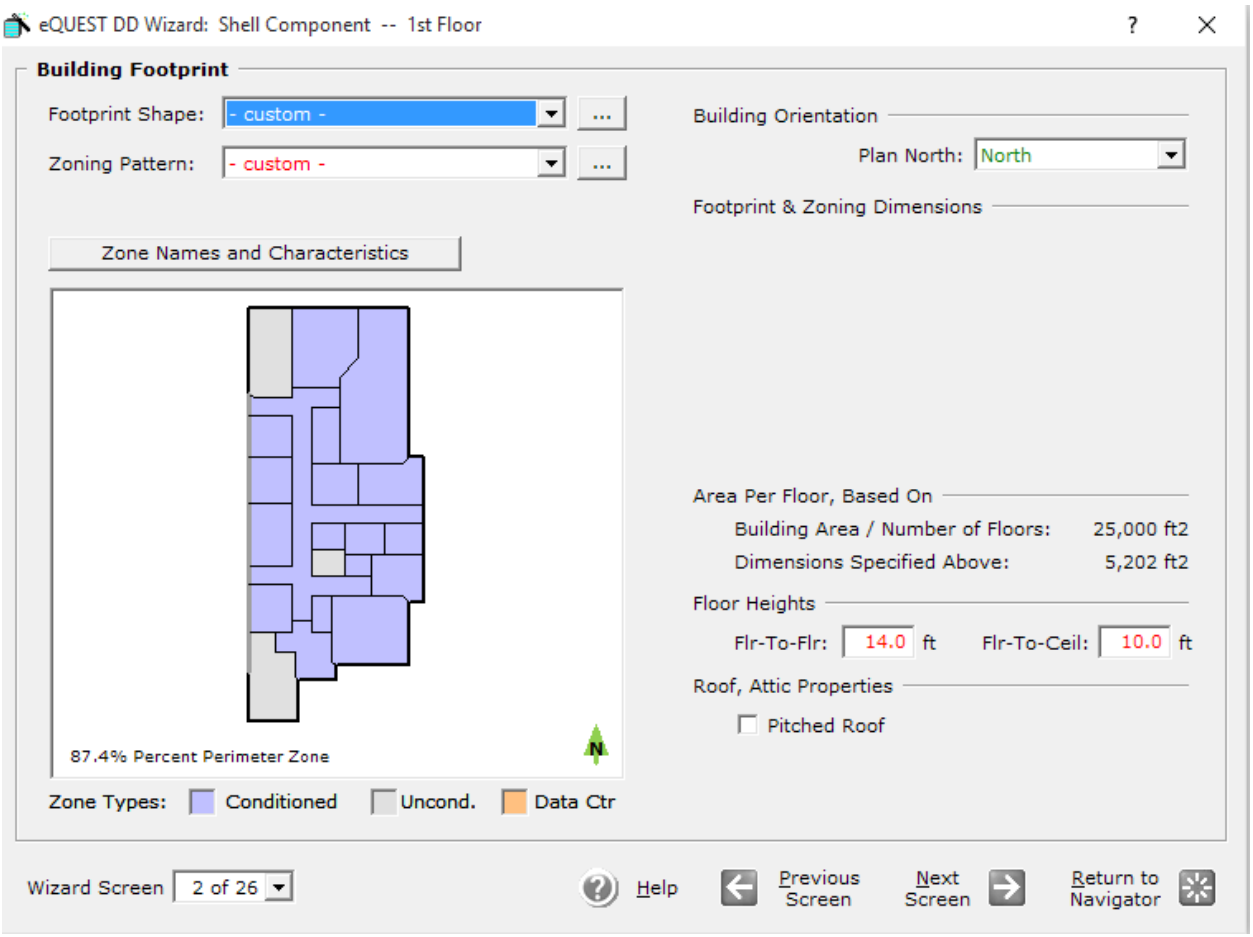
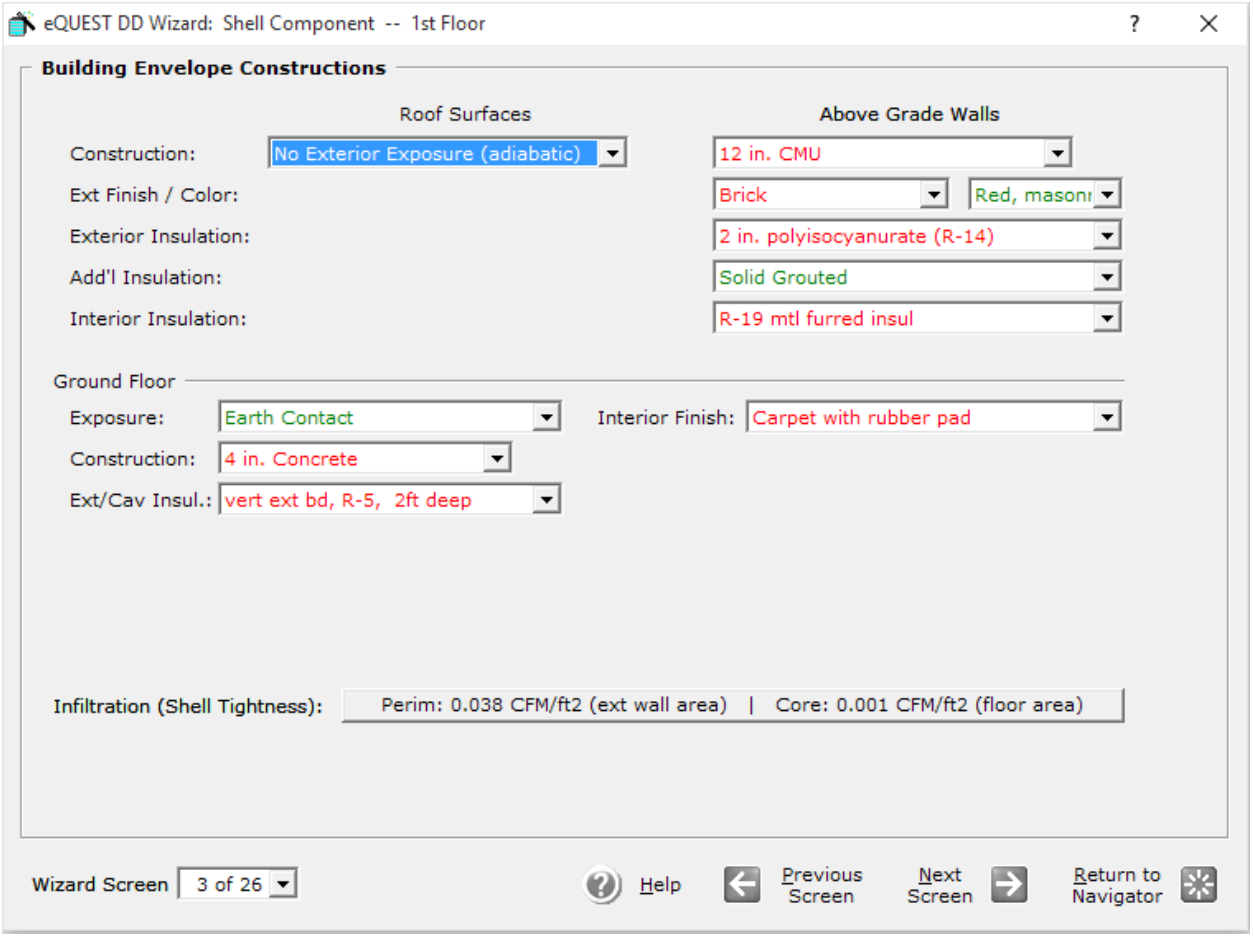
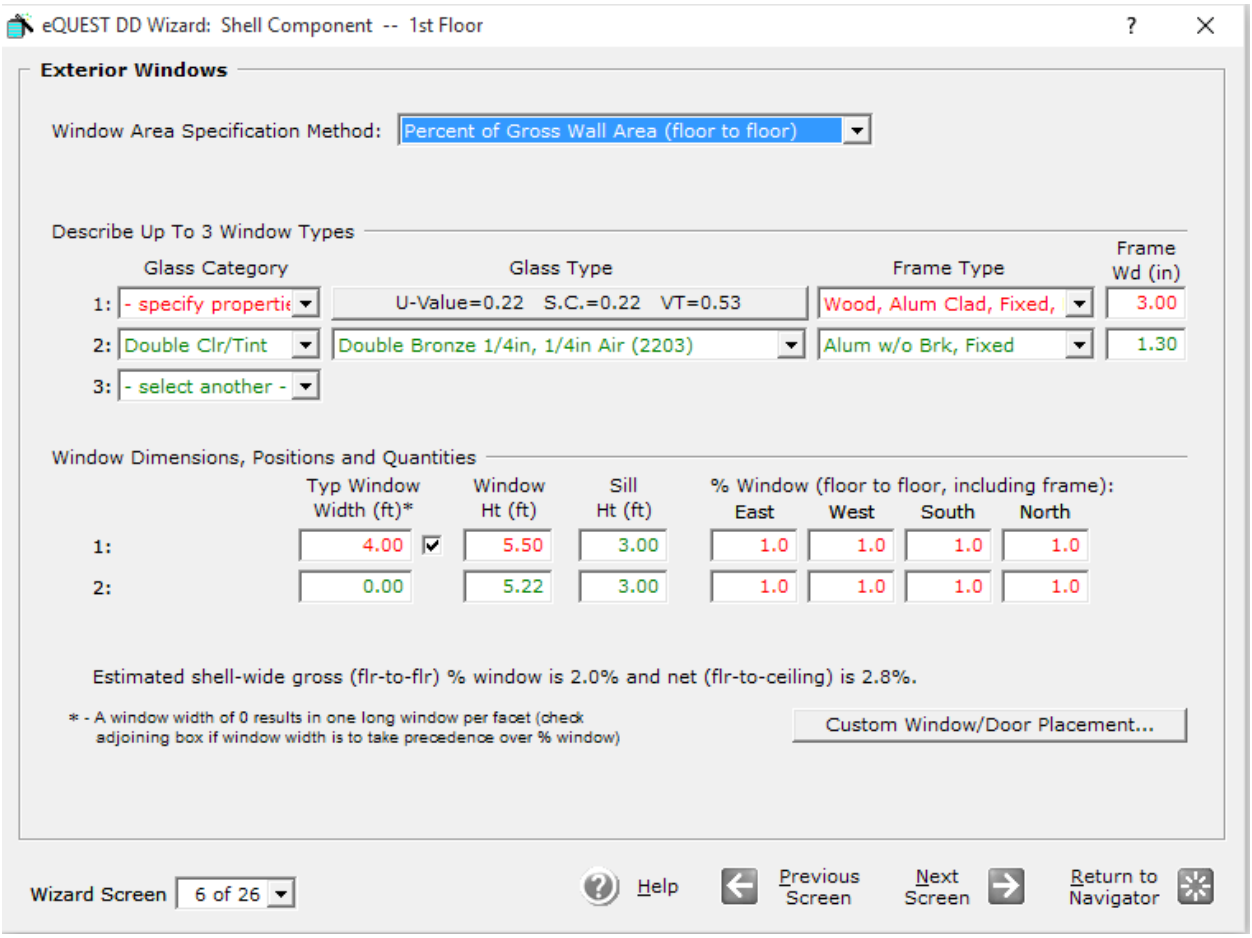
8 APPENDIX

8.1 APPENDIX A: Supplementary Data




eQUEST DD Wizard: Shell Component -- 1st Floor
Building Footprint
Footprint Shape:
- custom -
Zoning Pattern:
- custom -
Zone Names and Characteristics
87.4% Percent Perimeter Zone
N
Zone Types:
Conditioned
Uncond.
Data Ctr
Building Orientation
Plan North:
North
Footprint & Zoning Dimensions
Area Per Floor, Based On
Building Area / Number of Floors:
25,000 ft2
Dimensions Specified Above:
5,202 ft2
Floor Heights
Flr-To-Flr:
14.0
ft
Flr-To-Ceil:
10.0
ft
Roof, Attic Properties
Pitched Roof
Wizard Screen
2 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 1st Floor
Building Envelope Constructions
Roof Surfaces
Above Grade Walls
Construction:
No Exterior Exposure (adiabatic)
12 in. CMU
Ext Finish / Color:
Brick
Red, masonr
Exterior Insulation:
2 in. polyisocyanurate (R-14)
Add'l Insulation:
Solid Grouted
Interior Insulation:
R-19 mtl furred insul
Ground Floor
Exposure:
Earth Contact
Interior Finish:
Carpet with rubber pad
Construction:
4 in. Concrete
Ext/Cav Insul.:
vert ext bd, R-5, 2ft deep
Infiltration (Shell Tightness):
Perim: 0.038 CFM/ft2 (ext wall area) | Core: 0.001 CFM/ft2 (floor area)
Wizard Screen
3 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 1st Floor
Exterior Windows
Window Area Specification Method: Percent of Gross Wall Area (floor to floor)
Describe Up To 3 Window Types
Glass Category
Glass Type
Frame Type
Frame Wd (in)
1: - specify propertie
U-Value=0.22 S.C.=0.22 VT=0.53
Wood, Alum Clad, Fixed,
3.00
2: Double Clr/Tint
Double Bronze 1/4in, 1/4in Air (2203)
Alum w/o Brk, Fixed
1.30
3: - select another -
Window Dimensions, Positions and Quantities
Typ Window Width (ft)*
Window Ht (ft)
Sill Ht (ft)
% Window (floor to floor, including frame):
East
West
South
North
1: 4.00 5.50 3.00 1.0 1.0 1.0 1.0
2: 0.00 5.22 3.00 1.0 1.0 1.0 1.0
Estimated shell-wide gross (flr-to-flr) % window is 2.0% and net (flr-to-ceiling) is 2.8%.
* - A window width of 0 results in one long window per facet (check adjoining box if window width is to take precedence over % window)
Custom Window/Door Placement...
Wizard Screen 6 of 26
Help
Previous Screen
Next Screen
Return to Navigator

eQUEST DD Wizard: Shell Component -- 1st Floor

Interior Lighting Loads and Profiles

Area Type	Percent Area (%)	Lighting (W/SqFt)	Task Lt (W/SqFt)
1: Office (Executive/Private)	55.0	1.49	0.00
2: Corridor	15.0	0.57	0.00
3: Lobby (Office Reception/Waiting)	5.0	1.52	0.00
4: Restrooms	5.0	0.77	0.00
5: Conference Room	5.0	0.92	0.00
6: Mechanical/Electrical Room	5.0	0.81	0.00
7: Kitchen and Food Preparation	5.0	1.19	0.00
8: Dining Area	5.0	1.85	0.00
Multipliers on above intensities:		1.00	1.00

Interior Lighting Hourly Profiles by Season

Entire Year

Ambient: EL1 InsLtg Profile (S1)

Task: EL1 TskLtg Profile (S1)

Wizard Screen 16 of 26 Help Previous Screen Next Screen Return to Navigator

eQUEST DD Wizard: Shell Component -- 1st Floor

Miscellaneous Loads and Profiles

Area Type	Percent Area (%)	------ Electric ------ Load (W/SqFt)	Sensible Ht (frac)	---- Natural Gas ---- Load (Btuh/SF)	Sensible Ht (frac)
1: Office (Executive/Private)	55.0	0.75	1.00	0.00	1.00
2: Corridor	15.0	0.00	1.00	0.00	1.00
3: Lobby (Office Reception/Waiting)	5.0	0.25	1.00	0.00	1.00
4: Restrooms	5.0	0.10	1.00	0.00	1.00
5: Conference Room	5.0	0.10	1.00	0.00	1.00
6: Mechanical/Electrical Room	5.0	0.10	1.00	0.00	1.00
7: Kitchen and Food Preparation	5.0	1.00	1.00	0.00	1.00
8: Dining Area	5.0	0.10	1.00	0.00	1.00

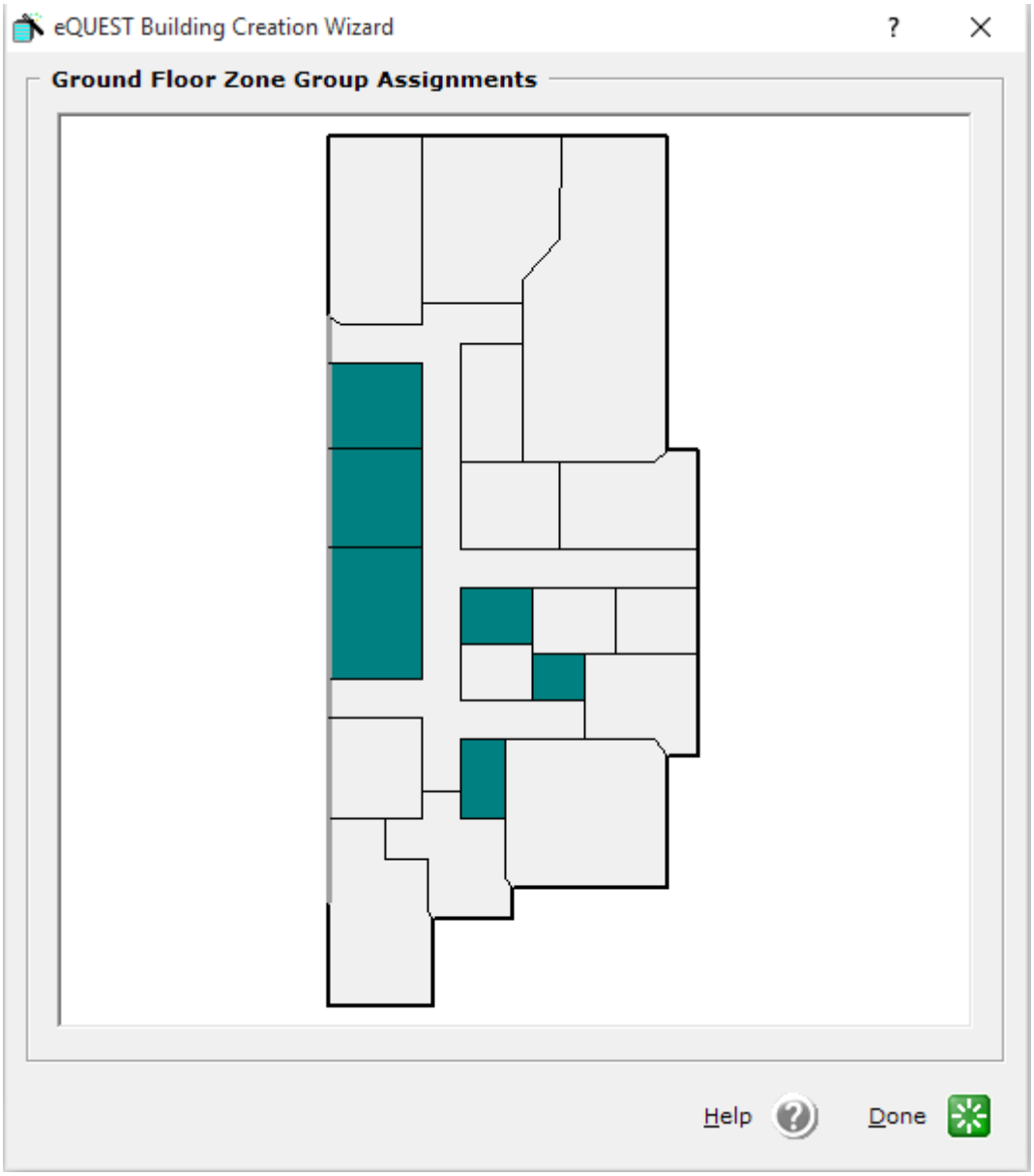
Miscellaneous Equipment Hourly Profiles by Season

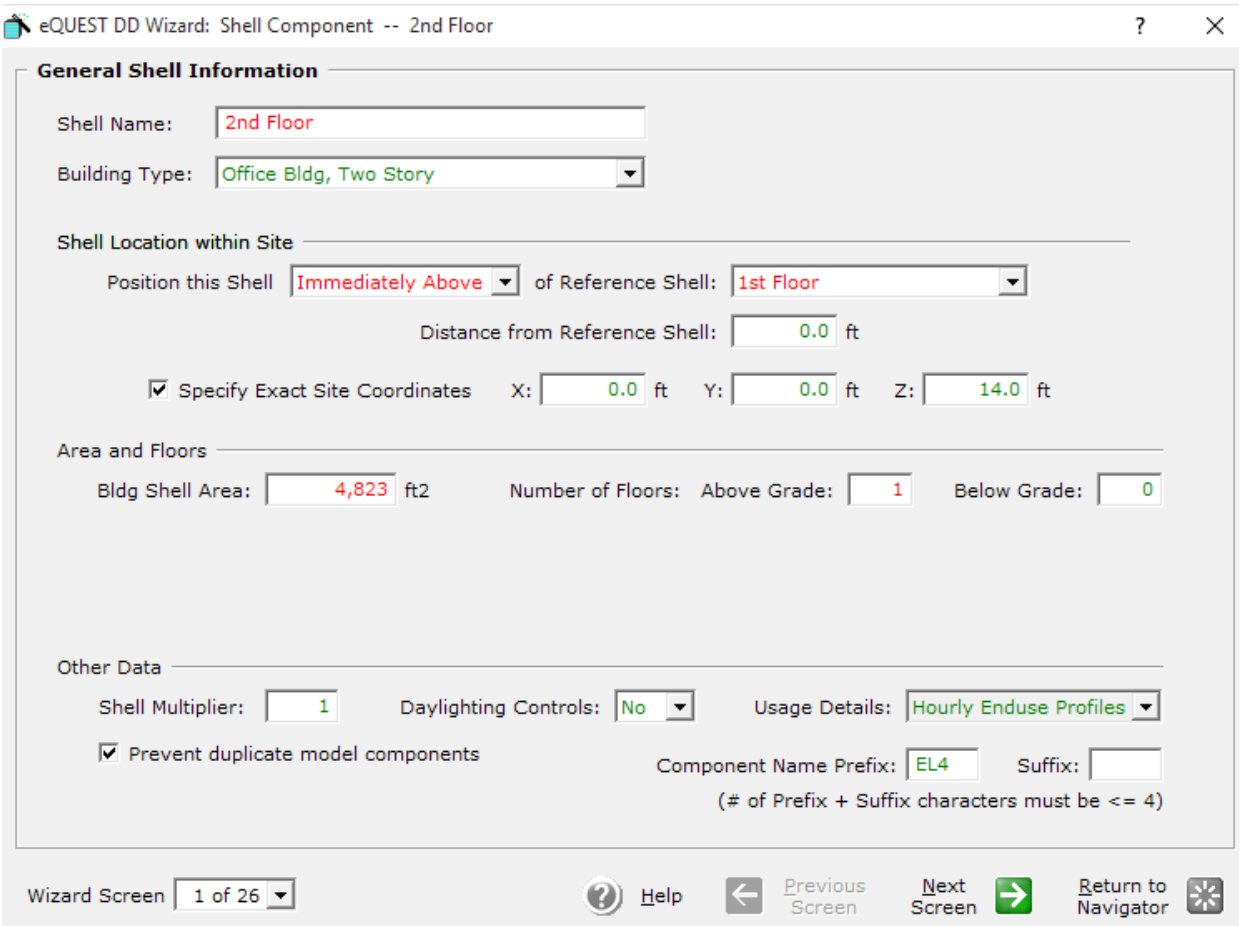
Entire Year

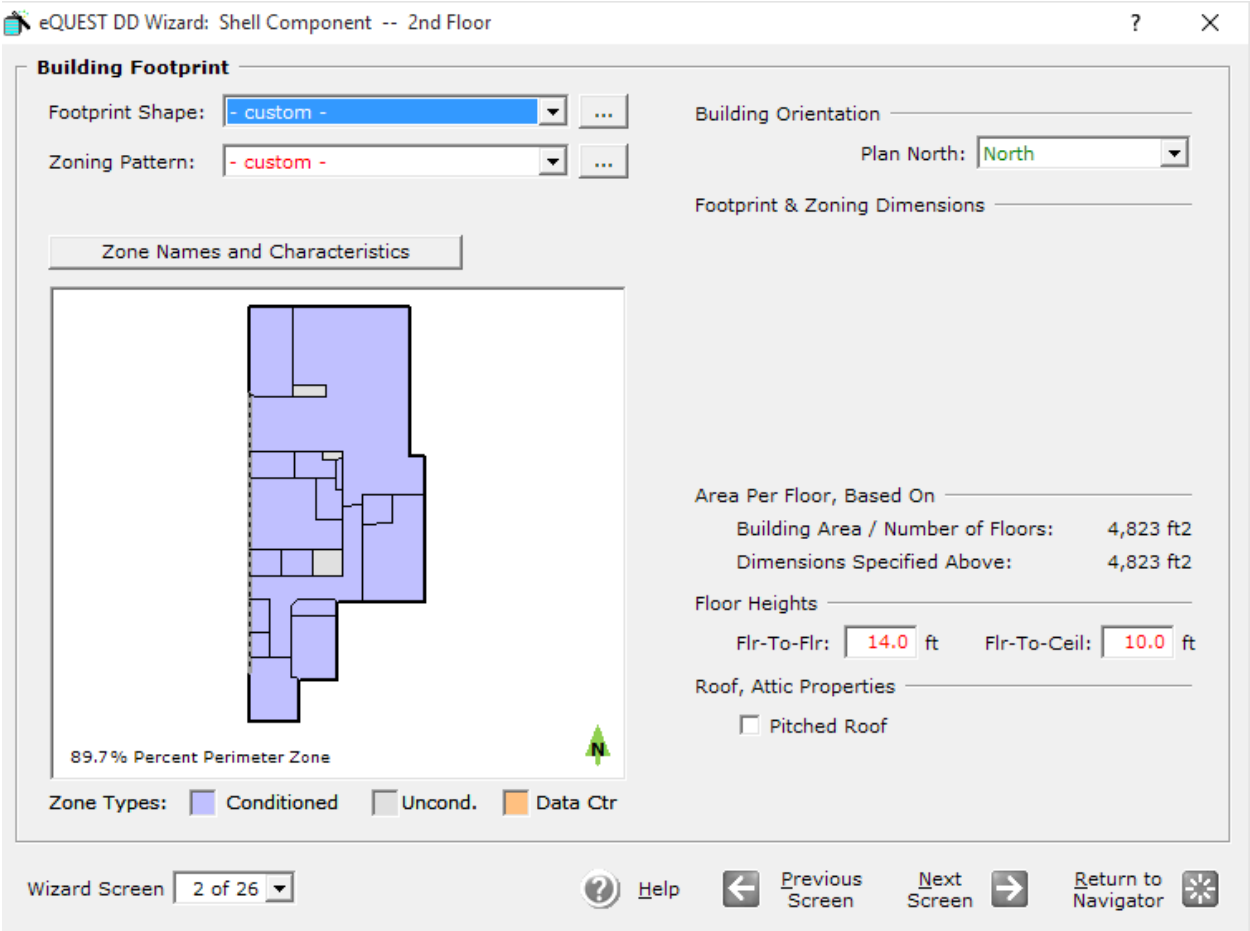
EL1 Misc Profile (S1)

...

Wizard Screen 20 of 26 Help Previous Screen Next Screen Return to Navigator


eQUEST Building Creation Wizard
Ground Floor Zone Group Assignments
Help
Done


eQUEST DD Wizard: Shell Component -- 2nd Floor
General Shell Information
Shell Name: 2nd Floor
Building Type: Office Bldg, Two Story
Shell Location within Site
Position this Shell Immediately Above of Reference Shell: 1st Floor
Distance from Reference Shell: 0.0 ft
Specify Exact Site Coordinates X: 0.0 ft Y: 0.0 ft Z: 14.0 ft
Area and Floors
Bldg Shell Area: 4,823 ft2 Number of Floors: Above Grade: 1 Below Grade: 0
Other Data
Shell Multiplier: 1 Daylighting Controls: No Usage Details: Hourly Enduse Profiles
Prevent duplicate model components
Component Name Prefix: EL4 Suffix:
(# of Prefix + Suffix characters must be <= 4)
Wizard Screen 1 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Building Footprint
Footprint Shape:
- custom -
Zoning Pattern:
- custom -
Zone Names and Characteristics
89.7% Percent Perimeter Zone
N
Zone Types:
Conditioned
Uncond.
Data Ctr
Building Orientation
Plan North:
North
Footprint & Zoning Dimensions
Area Per Floor, Based On
Building Area / Number of Floors:
4,823 ft2
Dimensions Specified Above:
4,823 ft2
Floor Heights
Flr-To-Flr:
14.0
ft
Flr-To-Ceil:
10.0
ft
Roof, Attic Properties
Pitched Roof
Wizard Screen
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Help
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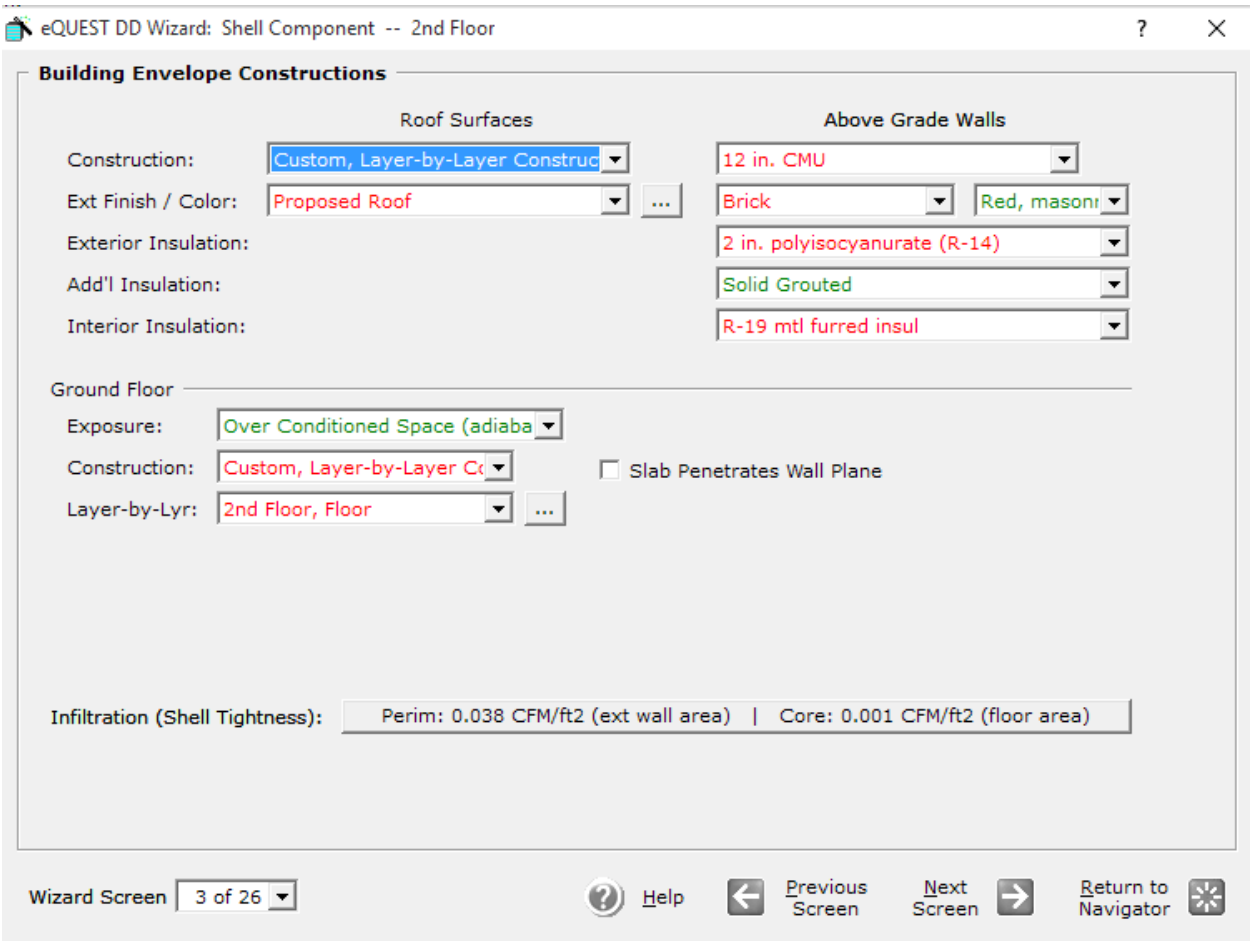
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Building Envelope Constructions
Roof Surfaces
Above Grade Walls
Construction:
Custom, Layer-by-Layer Construc
12 in. CMU
Ext Finish / Color:
Proposed Roof
Brick
Red, masonr
Exterior Insulation:
2 in. polyisocyanurate (R-14)
Add'l Insulation:
Solid Grouted
Interior Insulation:
R-19 mtl furred insul
Ground Floor
Exposure:
Over Conditioned Space (adiaba
Construction:
Custom, Layer-by-Layer Co
Slab Penetrates Wall Plane
Layer-by-Lyr:
2nd Floor, Floor
Infiltration (Shell Tightness):
Perim: 0.038 CFM/ft2 (ext wall area) | Core: 0.001 CFM/ft2 (floor area)
Wizard Screen
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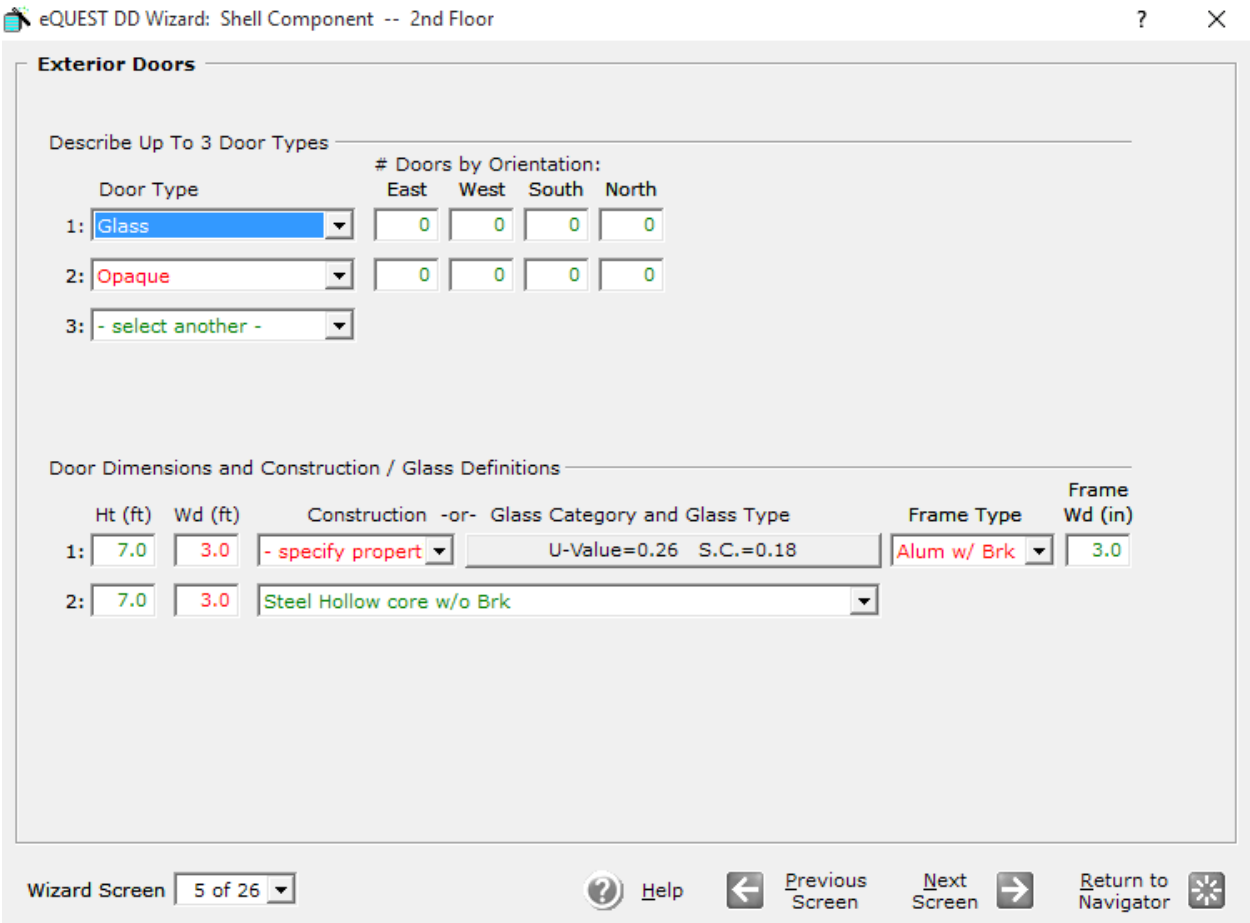
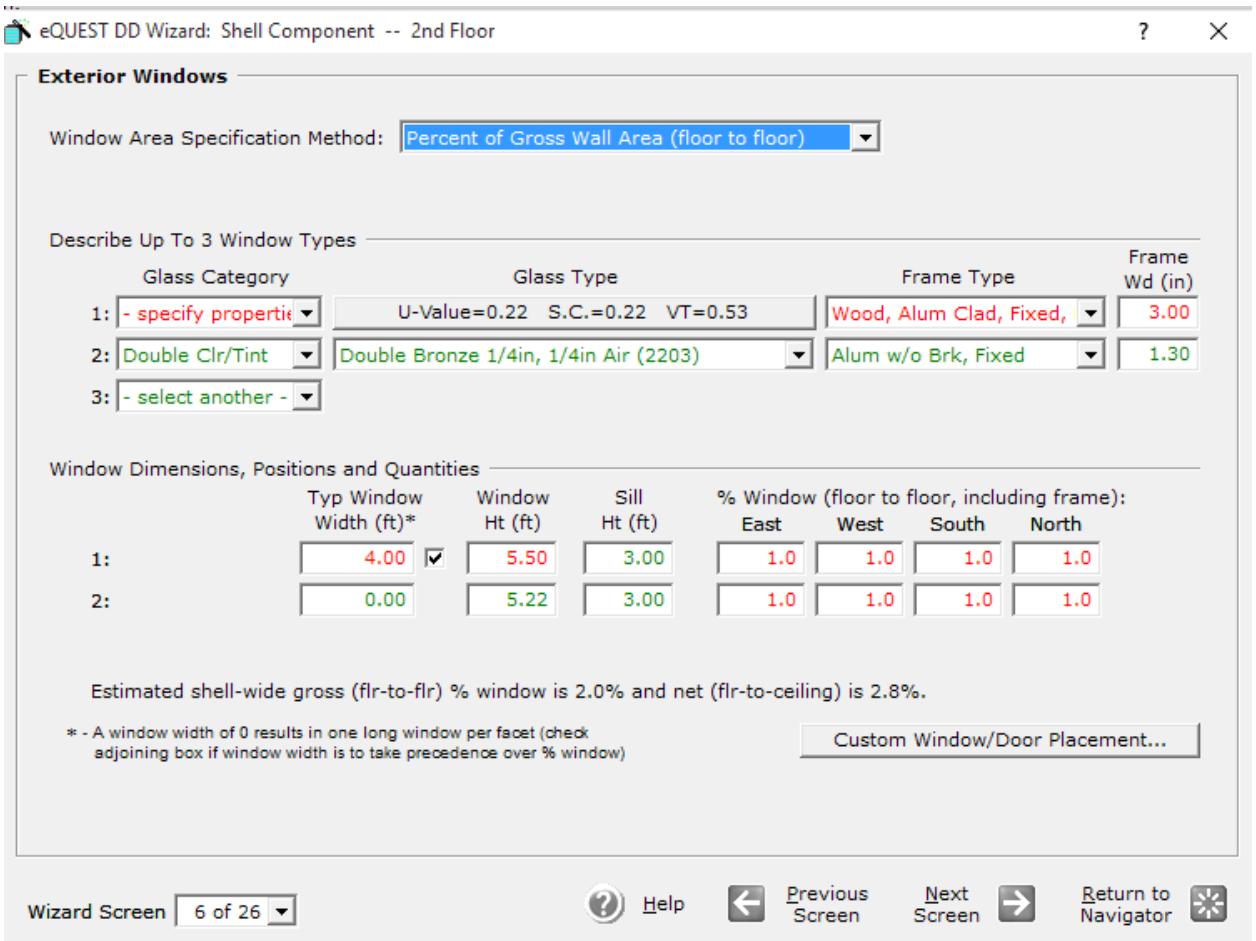
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Exterior Doors
Describe Up To 3 Door Types
Doors by Orientation:
Door Type
East
West
South
North
1: Glass 0 0 0 0
2: Opaque 0 0 0 0
3: - select another -
Door Dimensions and Construction / Glass Definitions
Ht (ft)
Wd (ft)
Construction -or- Glass Category and Glass Type
Frame Type
Frame Wd (in)
1: 7.0 3.0 - specify propert U-Value=0.26 S.C.=0.18 Alum w/ Brk 3.0
2: 7.0 3.0 Steel Hollow core w/o Brk
Wizard Screen 5 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Exterior Windows
Window Area Specification Method:
Percent of Gross Wall Area (floor to floor)
Describe Up To 3 Window Types
Glass Category
Glass Type
Frame Type
Frame Wd (in)
1: - specify propertie
U-Value=0.22 S.C.=0.22 VT=0.53
Wood, Alum Clad, Fixed,
3.00
2: Double Clr/Tint
Double Bronze 1/4in, 1/4in Air (2203)
Alum w/o Brk, Fixed
1.30
3: - select another -
Window Dimensions, Positions and Quantities
Typ Window Width (ft)*
Window Ht (ft)
Sill Ht (ft)
% Window (floor to floor, including frame):
East
West
South
North
1: 4.00 5.50 3.00 1.0 1.0 1.0 1.0
2: 0.00 5.22 3.00 1.0 1.0 1.0 1.0
Estimated shell-wide gross (flr-to-flr) % window is 2.0% and net (flr-to-ceiling) is 2.8%.
* - A window width of 0 results in one long window per facet (check adjoining box if window width is to take precedence over % window)
Custom Window/Door Placement...
Wizard Screen
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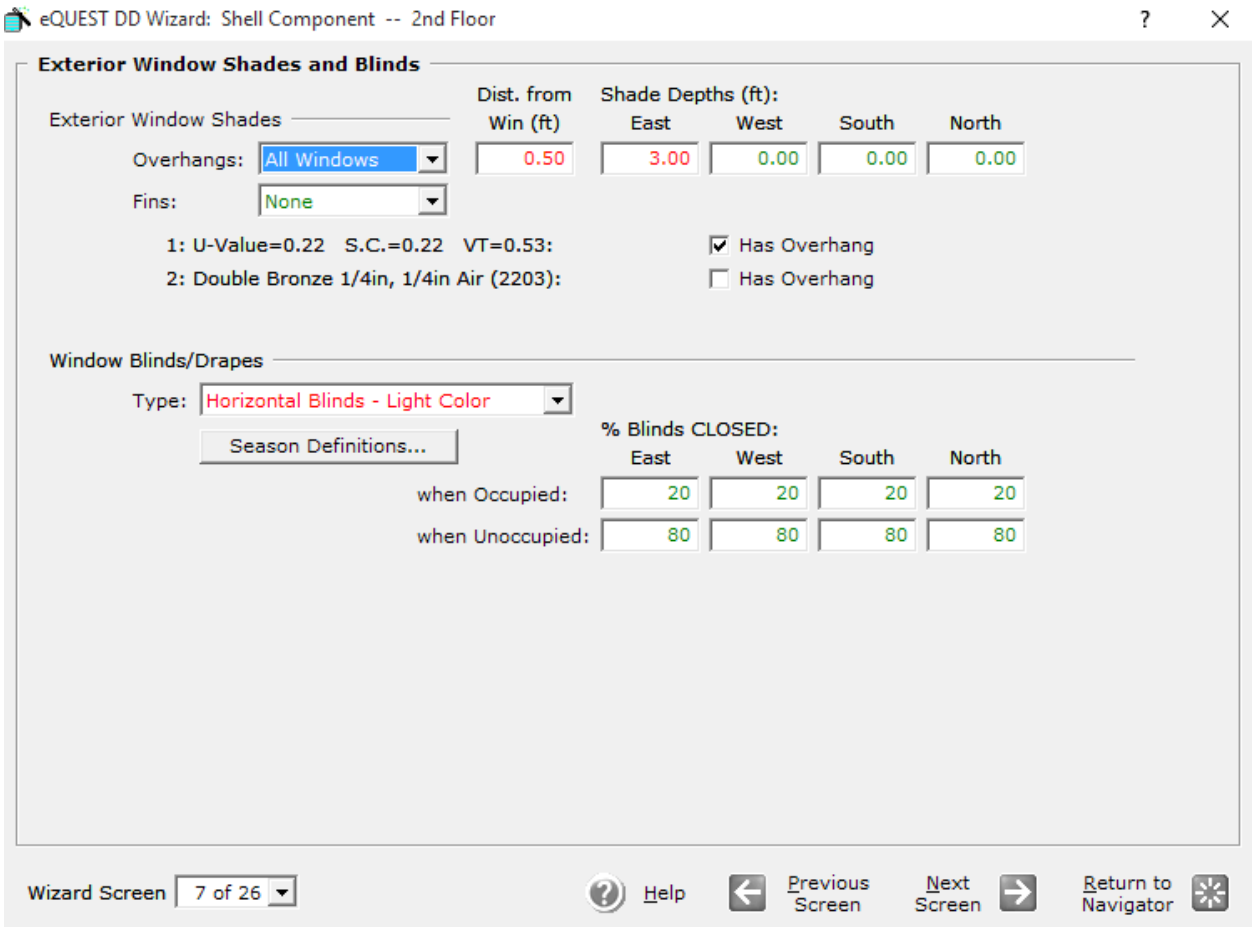
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Exterior Window Shades and Blinds
Exterior Window Shades
Dist. from Win (ft)
Shade Depths (ft):
East
West
South
North
Overhangs:
All Windows
0.50
3.00
0.00
0.00
0.00
Fins:
None
1: U-Value=0.22 S.C.=0.22 VT=0.53:
Has Overhang
2: Double Bronze 1/4in, 1/4in Air (2203):
Has Overhang
Window Blinds/Drapes
Type:
Horizontal Blinds - Light Color
Season Definitions...
% Blinds CLOSED:
East
West
South
North
when Occupied:
20
20
20
20
when Unoccupied:
80
80
80
80
Wizard Screen
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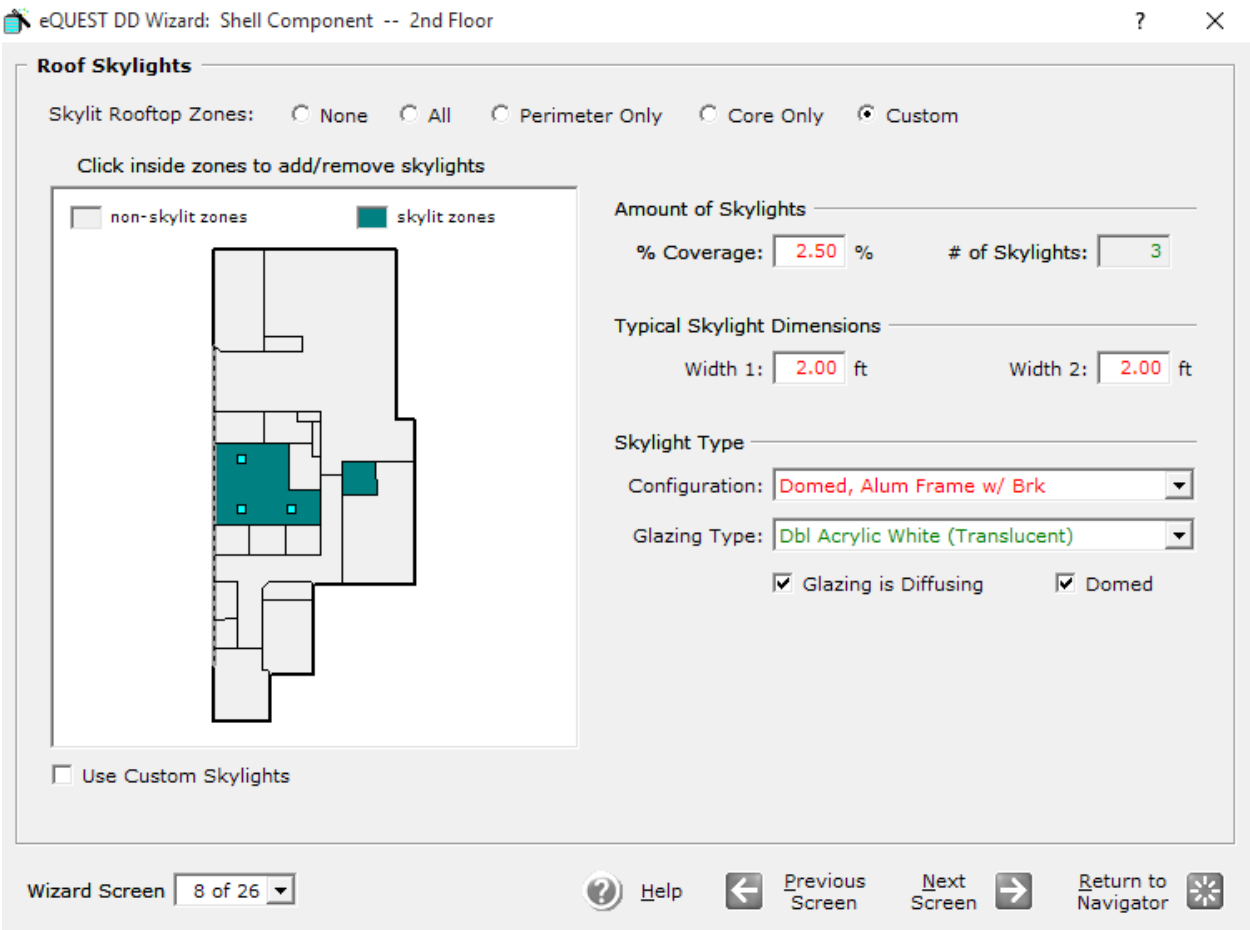
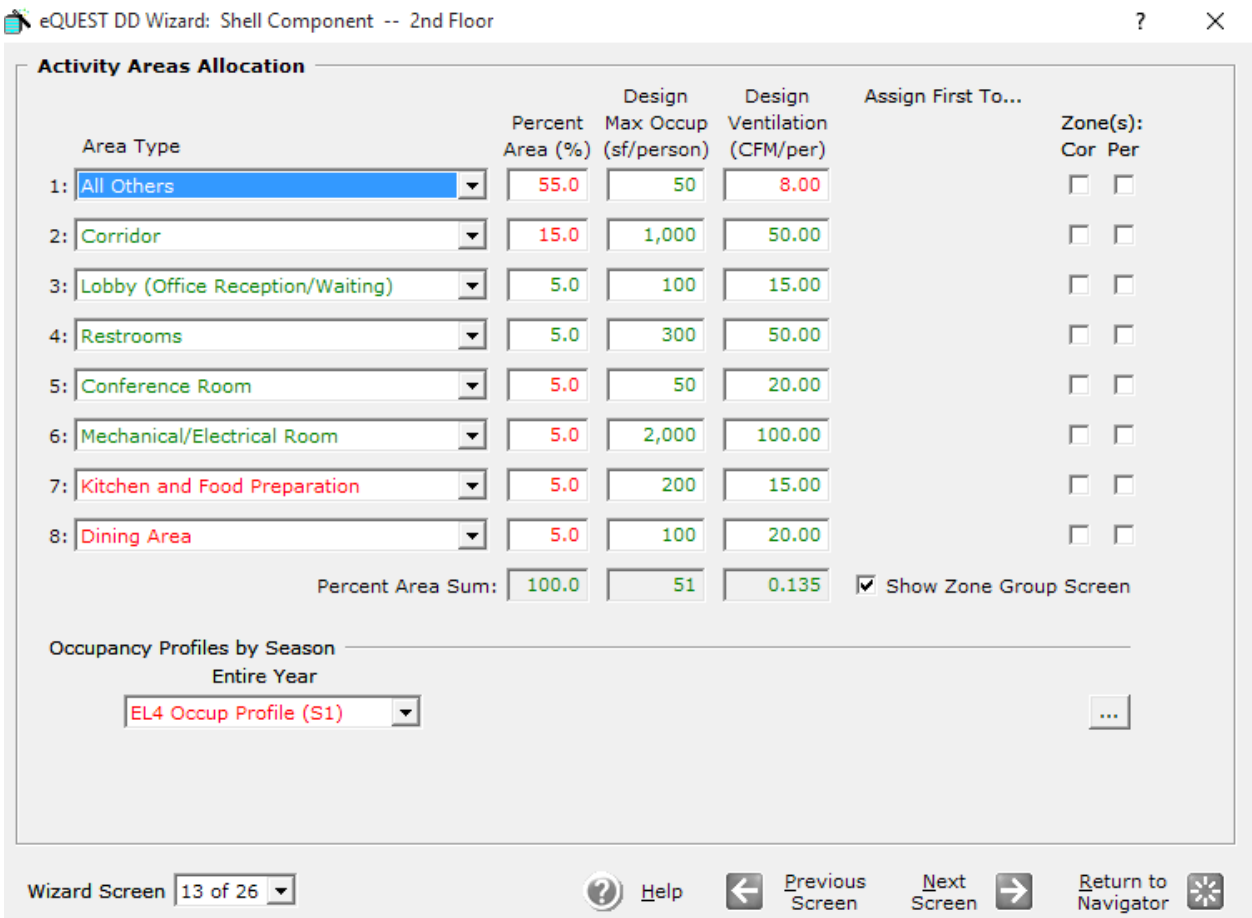
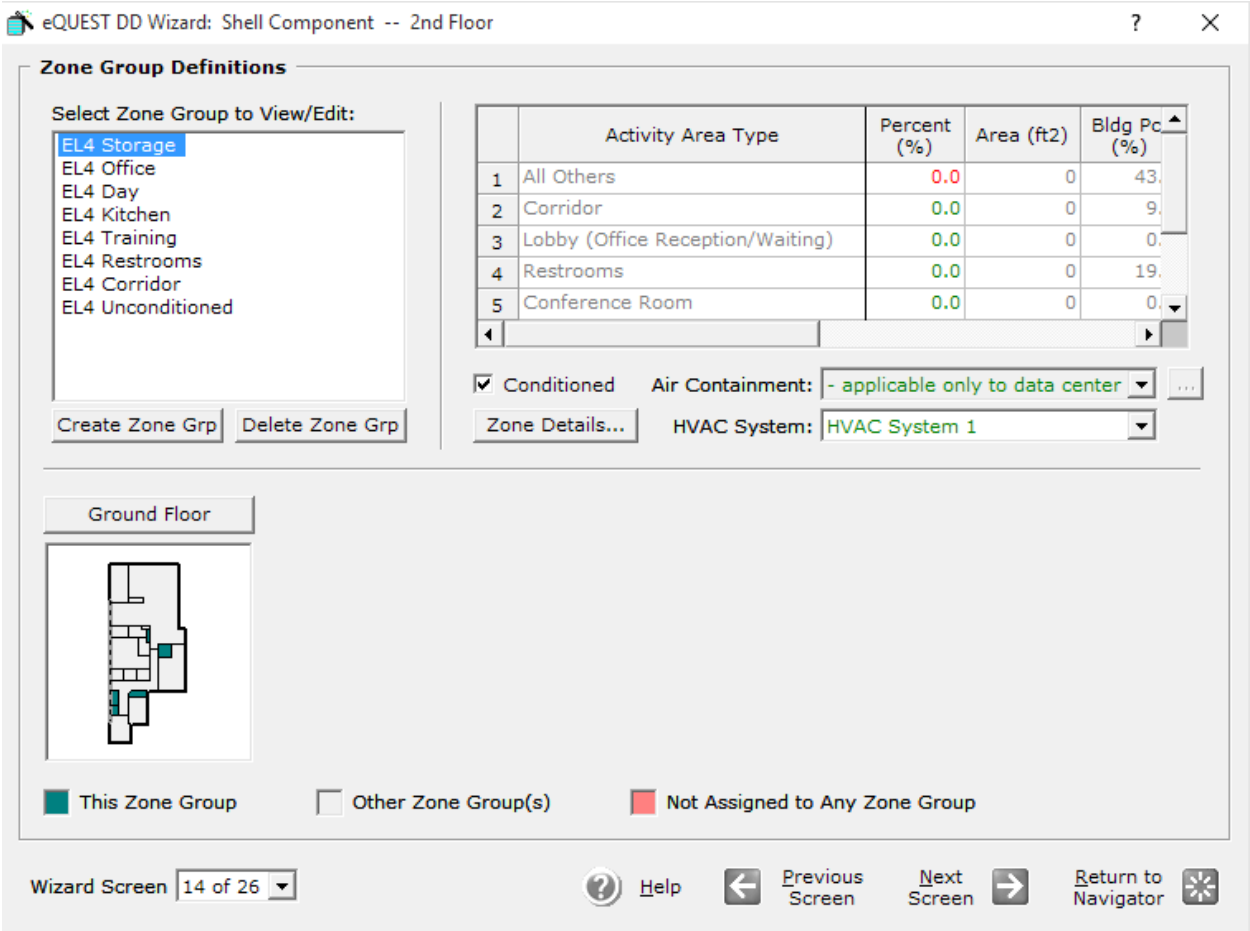
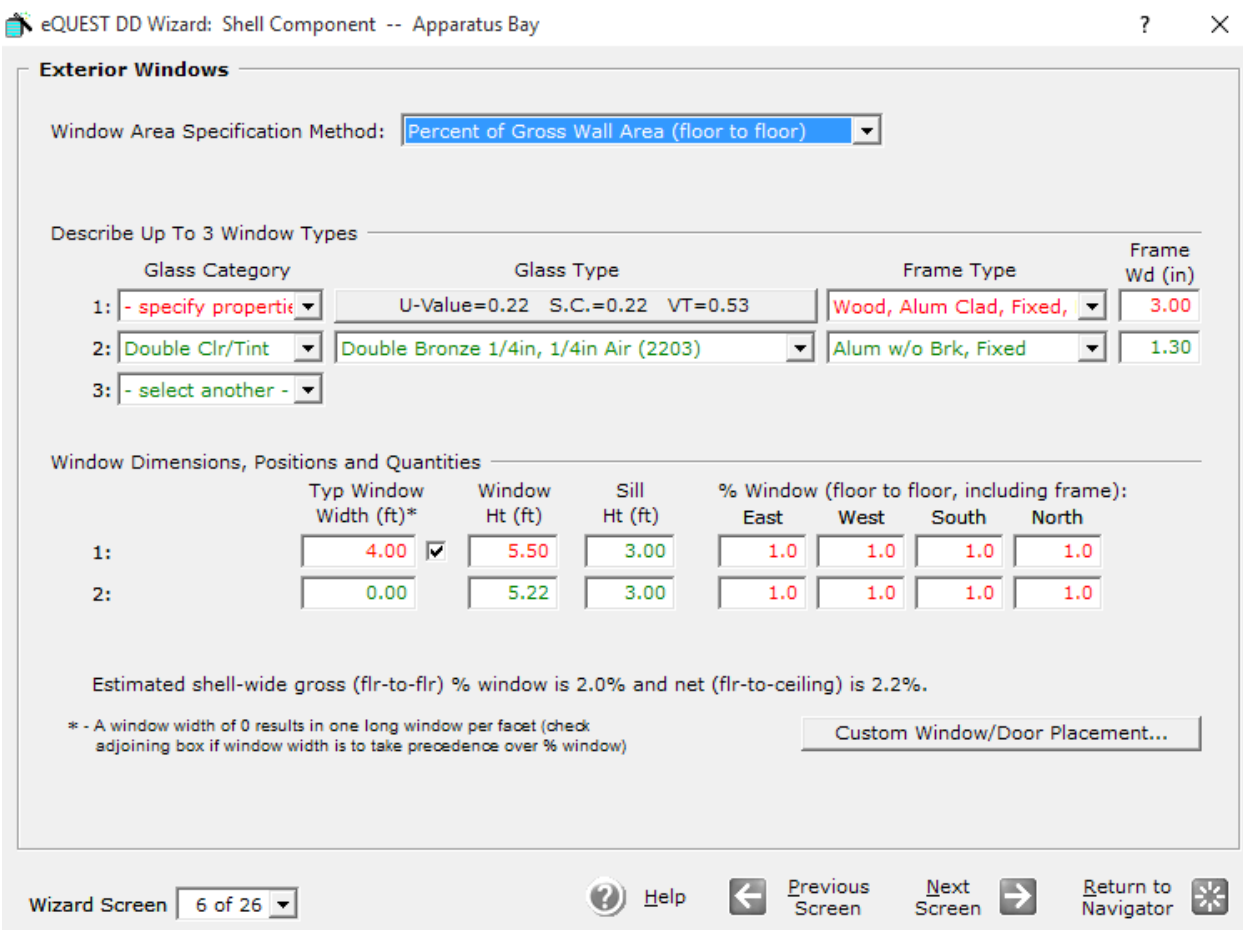
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Roof Skylights
Skylit Rooftop Zones:
None
All
Perimeter Only
Core Only
Custom
Click inside zones to add/remove skylights
non-skylit zones
skylit zones
Amount of Skylights
% Coverage: 2.50 %
# of Skylights: 3
Typical Skylight Dimensions
Width 1: 2.00 ft
Width 2: 2.00 ft
Skylight Type
Configuration: Domed, Alum Frame w/ Brk
Glazing Type: Dbl Acrylic White (Translucent)
Glazing is Diffusing
Domed
Use Custom Skylights
Wizard Screen 8 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Activity Areas Allocation
Area Type
Percent Area (%)
Design Max Occup (sf/person)
Design Ventilation (CFM/per)
Assign First To...
Zone(s): Cor Per
1: All Others 55.0 50 8.00
2: Corridor 15.0 1,000 50.00
3: Lobby (Office Reception/Waiting) 5.0 100 15.00
4: Restrooms 5.0 300 50.00
5: Conference Room 5.0 50 20.00
6: Mechanical/Electrical Room 5.0 2,000 100.00
7: Kitchen and Food Preparation 5.0 200 15.00
8: Dining Area 5.0 100 20.00
Percent Area Sum: 100.0 51 0.135
Show Zone Group Screen
Occupancy Profiles by Season
Entire Year
EL4 Occup Profile (S1)
Wizard Screen 13 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 2nd Floor
Zone Group Definitions
Select Zone Group to View/Edit:
EL4 Storage
EL4 Office
EL4 Day
EL4 Kitchen
EL4 Training
EL4 Restrooms
EL4 Corridor
EL4 Unconditioned
Create Zone Grp
Delete Zone Grp
Activity Area Type
Percent (%)
Area (ft2)
Bldg Pc (%)
1 All Others 0.0 0 43.
2 Corridor 0.0 0 9.
3 Lobby (Office Reception/Waiting) 0.0 0 0.
4 Restrooms 0.0 0 19.
5 Conference Room 0.0 0 0.
Conditioned
Air Containment: - applicable only to data center
Zone Details...
HVAC System: HVAC System 1
Ground Floor
This Zone Group
Other Zone Group(s)
Not Assigned to Any Zone Group
Wizard Screen 14 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- Apparatus Bay
Exterior Windows
Window Area Specification Method:
Percent of Gross Wall Area (floor to floor)
Describe Up To 3 Window Types
Glass Category
Glass Type
Frame Type
Frame Wd (in)
1: - specify propertie
U-Value=0.22 S.C.=0.22 VT=0.53
Wood, Alum Clad, Fixed,
3.00
2: Double Clr/Tint
Double Bronze 1/4in, 1/4in Air (2203)
Alum w/o Brk, Fixed
1.30
3: - select another -
Window Dimensions, Positions and Quantities
Typ Window Width (ft)*
Window Ht (ft)
Sill Ht (ft)
% Window (floor to floor, including frame):
East
West
South
North
1: 4.00 5.50 3.00 1.0 1.0 1.0 1.0
2: 0.00 5.22 3.00 1.0 1.0 1.0 1.0
Estimated shell-wide gross (flr-to-flr) % window is 2.0% and net (flr-to-ceiling) is 2.2%.
* - A window width of 0 results in one long window per facet (check adjoining box if window width is to take precedence over % window)
Custom Window/Door Placement...
Wizard Screen
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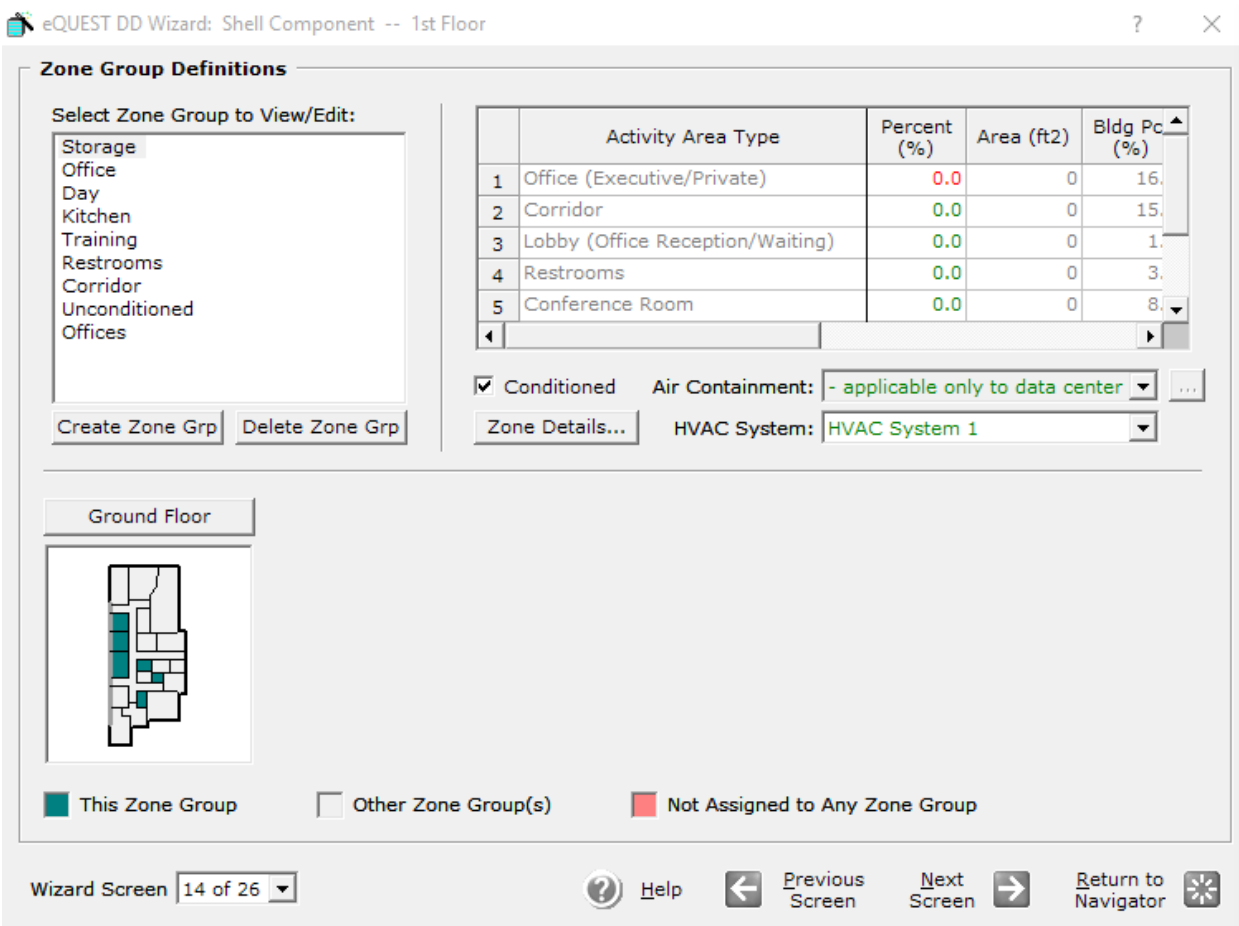
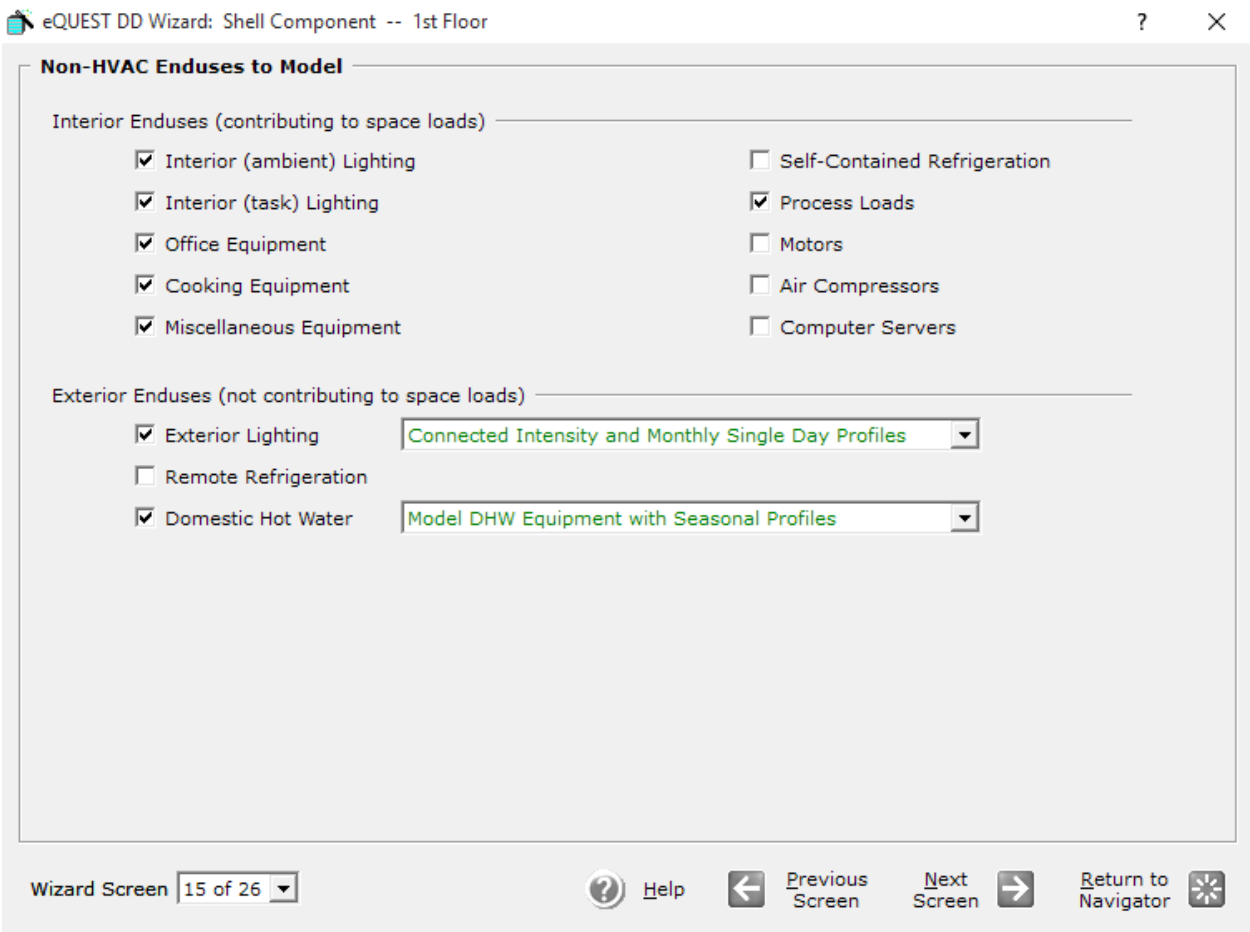
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 1st Floor
Zone Group Definitions
Select Zone Group to View/Edit:
Storage
Office
Day
Kitchen
Training
Restrooms
Corridor
Unconditioned
Offices
Create Zone Grp
Delete Zone Grp
Activity Area Type
Percent (%)
Area (ft2)
Bldg Pc (%)
1 Office (Executive/Private) 0.0 0 16.
2 Corridor 0.0 0 15.
3 Lobby (Office Reception/Waiting) 0.0 0 1.
4 Restrooms 0.0 0 3.
5 Conference Room 0.0 0 8.
Conditioned
Air Containment: - applicable only to data center
Zone Details...
HVAC System: HVAC System 1
Ground Floor
This Zone Group
Other Zone Group(s)
Not Assigned to Any Zone Group
Wizard Screen 14 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Shell Component -- 1st Floor
Non-HVAC Enduses to Model
Interior Enduses (contributing to space loads)
Interior (ambient) Lighting
Interior (task) Lighting
Office Equipment
Cooking Equipment
Miscellaneous Equipment
Self-Contained Refrigeration
Process Loads
Motors
Air Compressors
Computer Servers
Exterior Enduses (not contributing to space loads)
Exterior Lighting
Connected Intensity and Monthly Single Day Profiles
Remote Refrigeration
Domestic Hot Water
Model DHW Equipment with Seasonal Profiles
Wizard Screen
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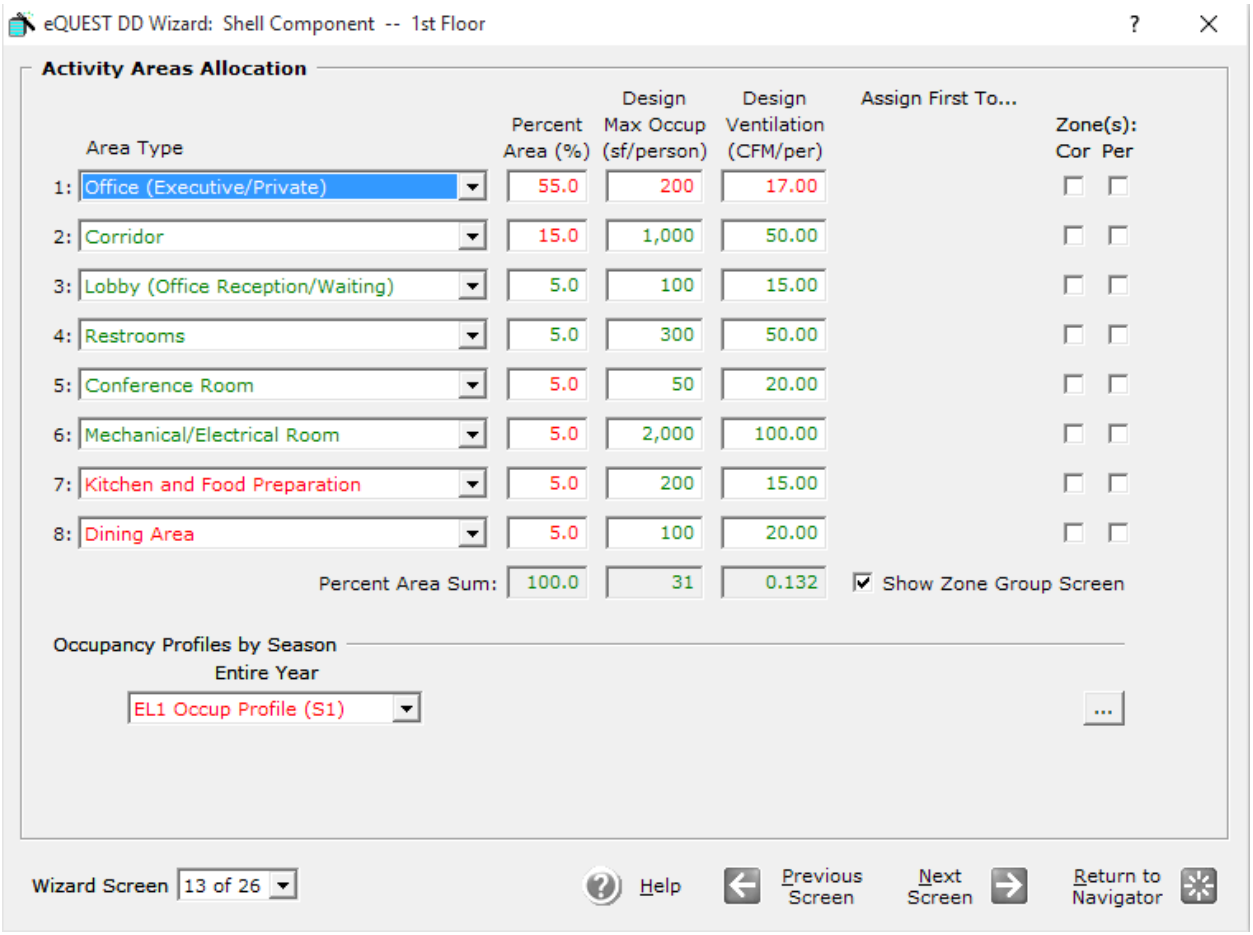
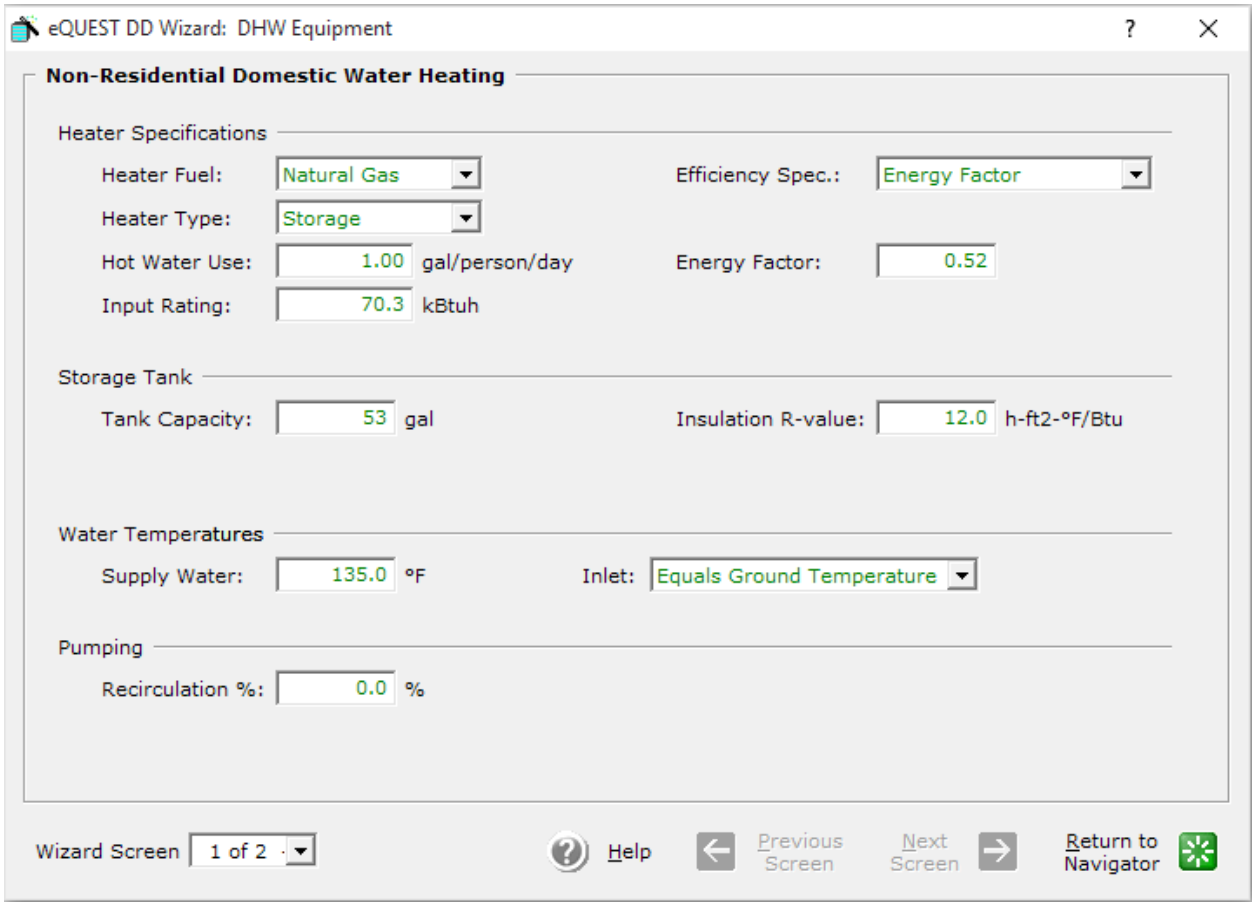
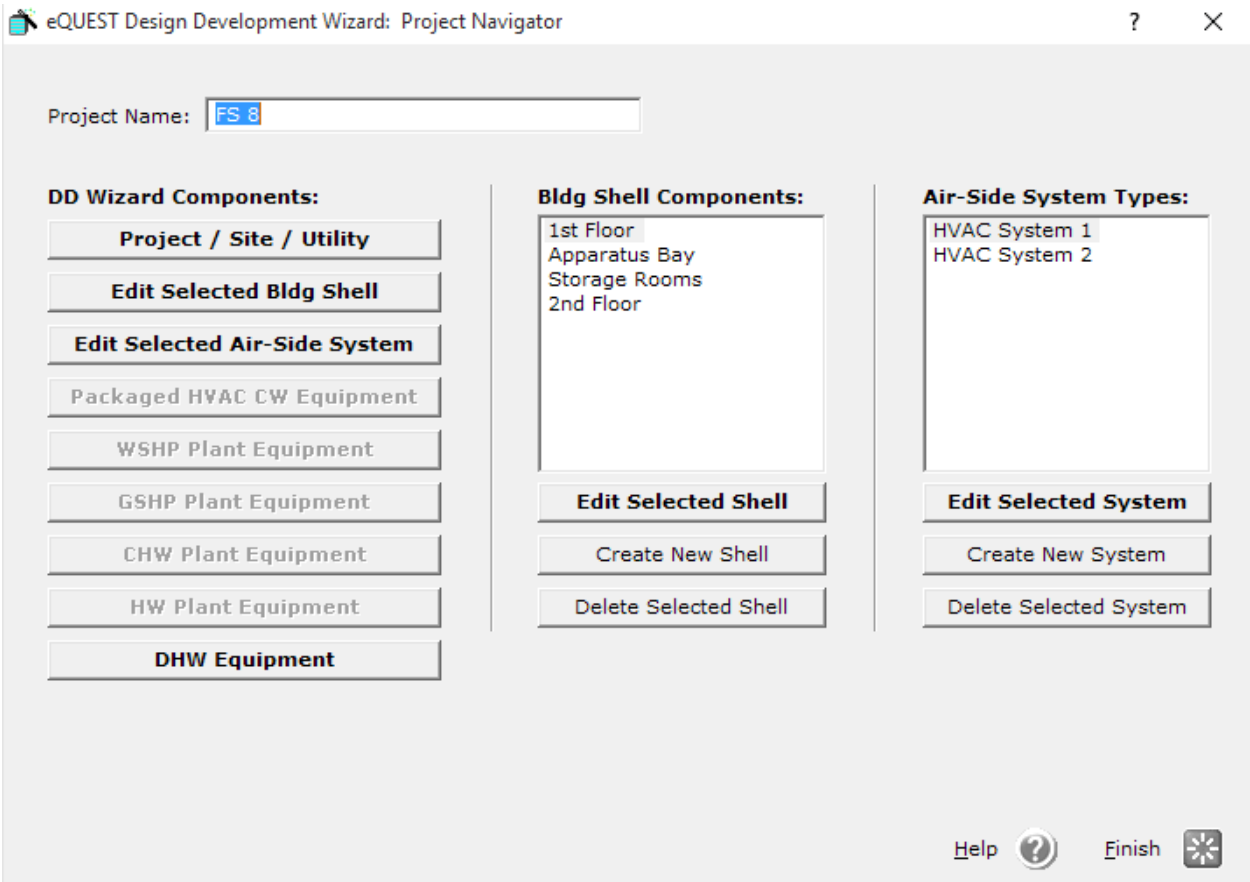
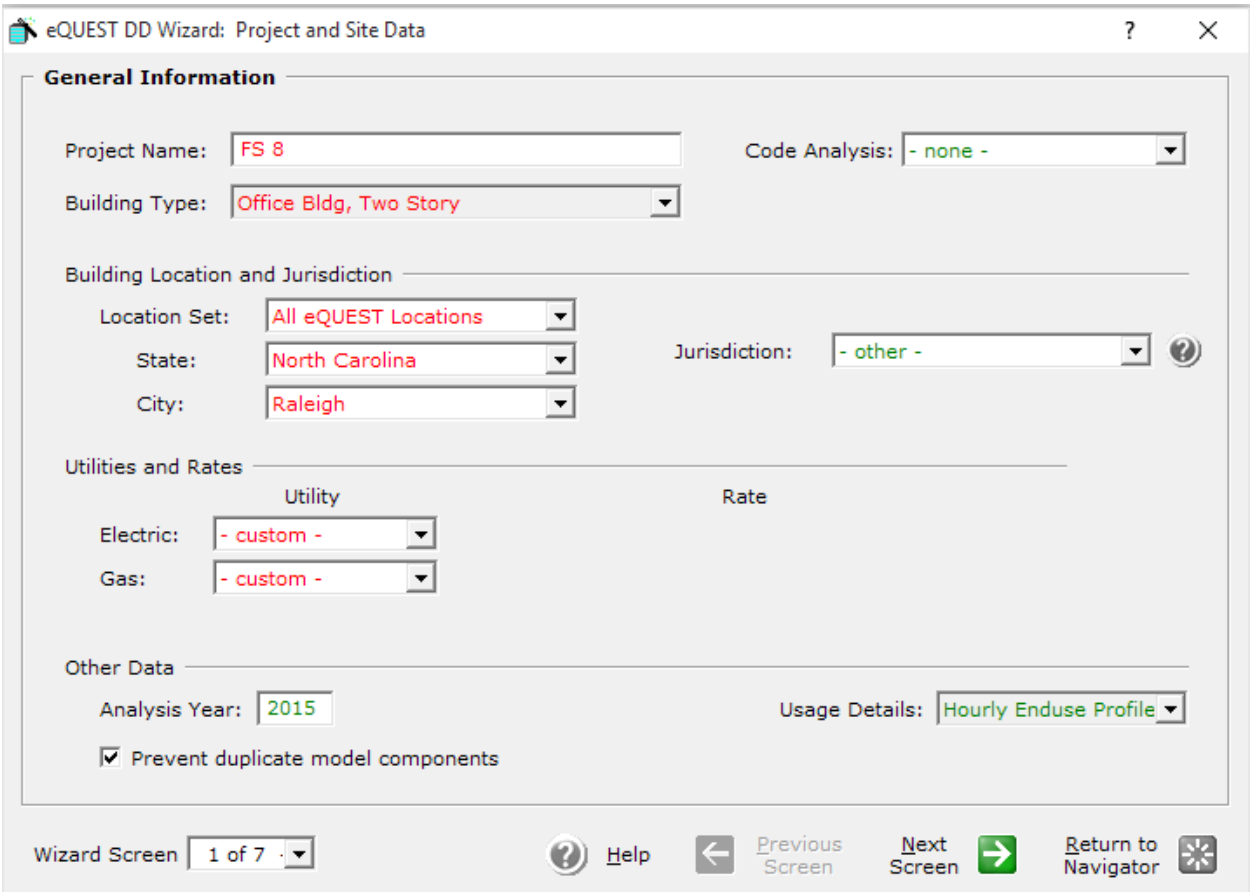
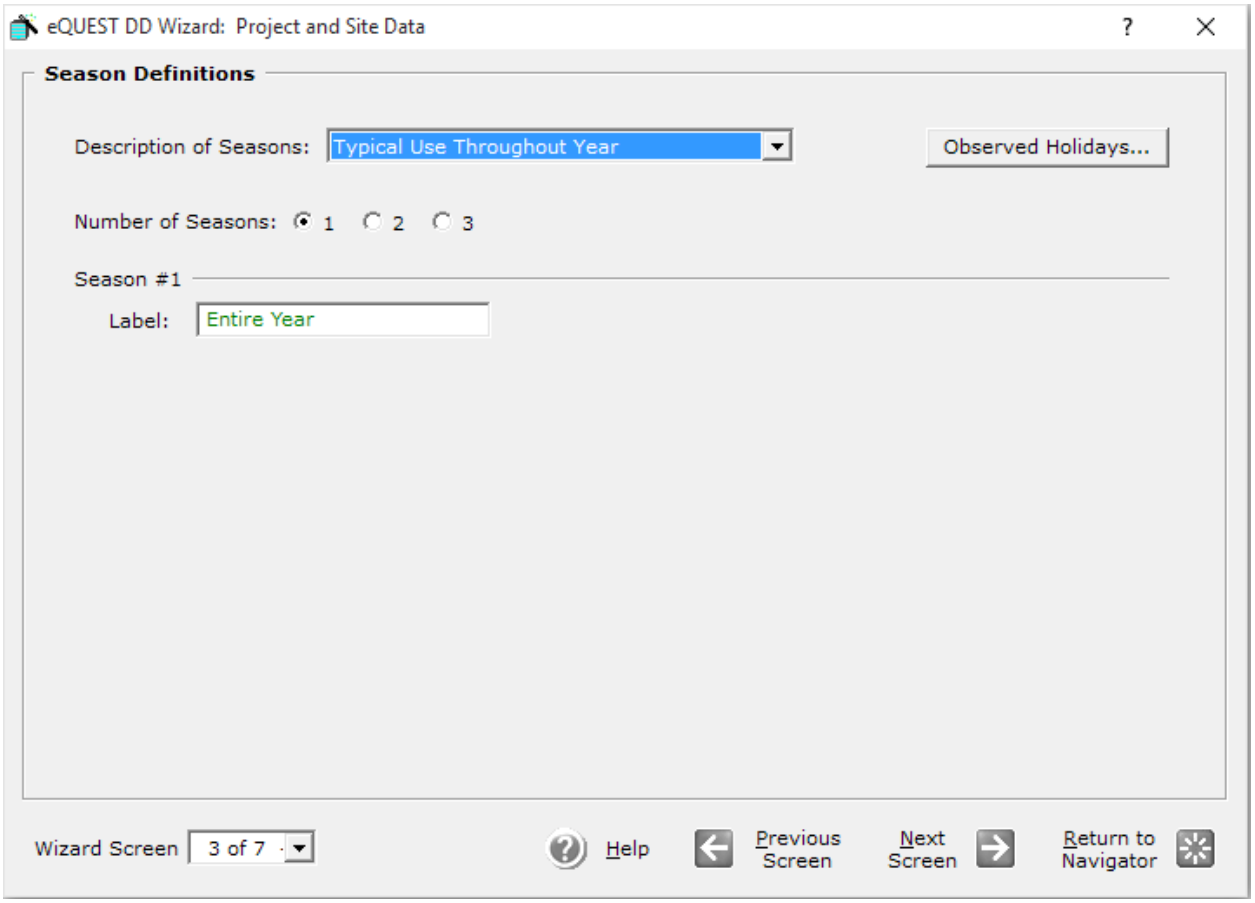
Return to Navigator


eQUEST DD Wizard: Shell Component -- 1st Floor
Activity Areas Allocation
Area Type
Percent Area (%)
Design Max Occup (sf/person)
Design Ventilation (CFM/per)
Assign First To...
Zone(s): Cor Per
1: Office (Executive/Private) 55.0 200 17.00
2: Corridor 15.0 1,000 50.00
3: Lobby (Office Reception/Waiting) 5.0 100 15.00
4: Restrooms 5.0 300 50.00
5: Conference Room 5.0 50 20.00
6: Mechanical/Electrical Room 5.0 2,000 100.00
7: Kitchen and Food Preparation 5.0 200 15.00
8: Dining Area 5.0 100 20.00
Percent Area Sum: 100.0 31 0.132
Show Zone Group Screen
Occupancy Profiles by Season
Entire Year
EL1 Occup Profile (S1)
Wizard Screen 13 of 26
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: DHW Equipment
Non-Residential Domestic Water Heating
Heater Specifications
Heater Fuel:
Natural Gas
Efficiency Spec.:
Energy Factor
Heater Type:
Storage
Hot Water Use:
1.00
gal/person/day
Energy Factor:
0.52
Input Rating:
70.3
kBtuh
Storage Tank
Tank Capacity:
53
gal
Insulation R-value:
12.0
h-ft2-°F/Btu
Water Temperatures
Supply Water:
135.0
°F
Inlet:
Equals Ground Temperature
Pumping
Recirculation %:
0.0
%
Wizard Screen
1 of 2
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST Design Development Wizard: Project Navigator
Project Name:
FS 8
DD Wizard Components:
Project / Site / Utility
Edit Selected Bldg Shell
Edit Selected Air-Side System
Packaged HVAC CW Equipment
WSHP Plant Equipment
GSHP Plant Equipment
CHW Plant Equipment
HW Plant Equipment
DHW Equipment
Bldg Shell Components:
1st Floor
Apparatus Bay
Storage Rooms
2nd Floor
Edit Selected Shell
Create New Shell
Delete Selected Shell
Air-Side System Types:
HVAC System 1
HVAC System 2
Edit Selected System
Create New System
Delete Selected System
Help
Finish


eQUEST DD Wizard: Project and Site Data
General Information
Project Name:
FS 8
Code Analysis:
- none -
Building Type:
Office Bldg, Two Story
Building Location and Jurisdiction
Location Set:
All eQUEST Locations
State:
North Carolina
City:
Raleigh
Jurisdiction:
- other -
Utilities and Rates
Utility
Rate
Electric:
- custom -
Gas:
- custom -
Other Data
Analysis Year:
2015
Usage Details:
Hourly Enduse Profile
Prevent duplicate model components
Wizard Screen
1 of 7
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Project and Site Data
Season Definitions
Description of Seasons:
Typical Use Throughout Year
Observed Holidays...
Number of Seasons:
1
2
3
Season #1
Label:
Entire Year
Wizard Screen
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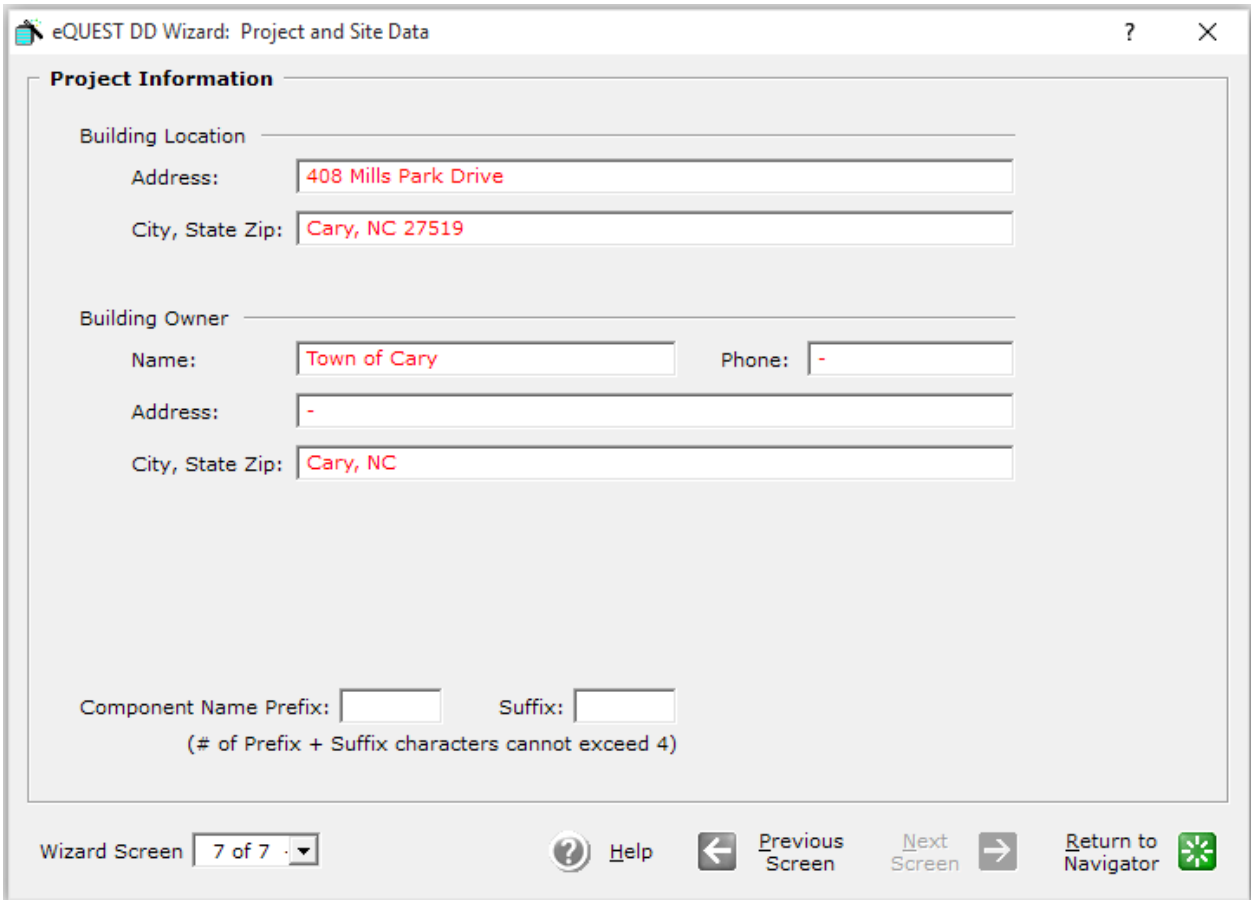
Help
Previous Screen
Next Screen
Return to Navigator


eQUEST DD Wizard: Project and Site Data
Project Information
Building Location
Address:
408 Mills Park Drive
City, State Zip:
Cary, NC 27519
Building Owner
Name:
Town of Cary
Phone:
-
Address:
-
City, State Zip:
Cary, NC
Component Name Prefix:
Suffix:
(# of Prefix + Suffix characters cannot exceed 4)
Wizard Screen
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Help
Previous Screen
Next Screen
Return to Navigator


VENMAR CES
WARNING
WARNING
DANGER
AVERTISSEMENT
AVERTISSEMENT
IMPORTANT