ANALYSIS OF INNOVATIVE HVAC SYSTEM TECHNOLOGIES AND THEIR APPLICATION FOR OFFICE BUILDINGS IN HOT AND HUMID CLIMATES

A Thesis

by

OLEKSANDR TANSKYI

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

MASTER OF SCIENCE

December 2010

Major Subject: Mechanical Engineering

Analysis of Innovative HVAC System Technologies and Their

Application for Office Buildings in Hot and Humid Climates

ANALYSIS OF INNOVATIVE HVAC SYSTEM TECHNOLOGIES AND THEIR APPLICATION FOR OFFICE BUILDINGS IN HOT AND HUMID CLIMATES

A Thesis

by

OLEKSANDR TANSKYI

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

MASTER OF SCIENCE

Approved by:

Co-Chairs of Committee,	David E. Claridge
	Michael B. Pate
Committee Members,	Charles H. Culp
	Juan-Carlos Baltazar-Cervantes
Head of Department,	Dennis L. O'Neal

December 2010

Major Subject: Mechanical Engineering

ABSTRACT

Analysis of Innovative HVAC System Technologies and Their Application for Office Buildings in Hot and Humid Climates. (December 2010)

Oleksandr Tanskyi, B.S., National Technical University of Ukraine;

M.S., National Technical University of Ukraine

Co-Chairs of Advisory Committee: Dr. David E. Claridge
Dr. Michael B. Pate

The commercial buildings sector in the United States used 18% (17.93 Quads) of the U.S. primary energy in 2006. Office buildings are the largest single energy consumption category in the commercial buildings sector of the United States with annual energy consumption around 1.1 Quads. Traditional approaches used in commercial building designs are not adequate to save energy in both depth and scale. One of the most effective ways to reduce energy consumption is to improve energy performance of HVAC systems.

High-performance HVAC systems and components, as well as application of renewable energy sources, were surveyed for buildings in hot and humid climates. An analysis of performance and energy saving potential estimation for selected HVAC systems in hot and humid climates was developed based on energy consumption simulation models in DOE-2.1E.

A calibrated energy consumption model of an existing office building located in the hot and humid climate conditions of Texas was developed. Based on this model, the energy saving potential of the building was estimated.

In addition, energy consumption simulation models were developed for a new office building, including simulation of energy saving measures that could be achieved with further improvements of HVAC system above the energy conservation codes requirements. The theoretical minimum energy consumption level for the same office

building was estimated for the purpose of evaluating the whole building energy efficiency level. The theoretical minimum energy consumption model of the office building was designed to provide the same level of comfort and services to the building occupants as provided in the actual building simulation model.

Finally, the energy efficiency of the building that satisfies valid energy conservation codes and the building with an improved HVAC system was estimated based on theoretically minimum energy consumption level.

The analysis provided herein can be used for new building practitioners and existing building owners to evaluate energy reduction potential and the performance of innovative technologies such as dedicated outdoor air system, displacement ventilation, improved cooling system efficiency, air source heat pumps and natural gas heat pumps.

DEDICATION

To my Mother who has supported and encouraged me throughout my life and provided an exceptional model for me to follow as a researcher that continues to inspire me.

ACKNOWLEDGEMENTS

I would like to express my sincere gratitude to my graduate advisor, Dr. David Claridge, for his great guidance, mentorship, stimulating discussions, support and encouragement throughout all the years that I have been fortunate to know him.

I also would like to acknowledge the help of all other members of my committee: Dr. Juan-Carlos Baltazar-Cervantes, for encouragement and advice throughout all my research work; Dr. Charles Culp, for suggestions that even extend beyond the scope of this work; and Dr. Michael Pate, for broadening my knowledge about a variety of building systems. I am fortunate to have these professionals as my committee.

Many thanks go to Dr. Jeff Haberl, for help and support with DOE-2.1E simulations. I am grateful for the discussions and support given by Mr. Jijun Zhou. I greatly appreciate Mr. Michael Martine and Mr. Iraj Solouki for their help to install the data acquisition equipment and gather experimental data. I also thank Ms. Rose Sauser for editing this manuscript.

Finally, the appreciation is extended to all my friends and colleagues in the Energy Systems Laboratory for their continuous support, assistance in answering my questions and well wishes.

NOMENCLATURE

ChW – Chilled water

db – Dry-bulb

DDC – Direct digital control

DOAS – Dedicated outdoor air systems

DV – Displacement ventilation

EER – Energy efficiency ratio

ESL – Energy System Laboratory

GHX – Ground heat exchanger

HHW – Heating hot water

hp – Horsepower

HVAC – Heating ventilation air conditioning

IECC – International Energy Conservation Code

MBE – Mean bias error

MTL – Material testing laboratory

OA – Outside air

PV – Photovoltaic

Quad – Quadrillion Btu (10^{15} Btu)

RMSE – Root mean square error

RT– Refrigeration ton

SC – Shading coefficient

SEER – Seasonal energy efficiency ratio

SHGC – Solar heat gain coefficient

UFAD – Underfloor air distribution system

VAV – Variable air volume

VFD – Variable frequency drive

wb – wet-bulb

WERC – Wisenbaker Engineering Research Center

TABLE OF CONTENTS

Page

Page

Page

LIST OF FIGURES

Page

Page

Page

LIST OF TABLES

Page

Page

CHAPTER I

INTRODUCTION

1.1. Background

The buildings energy sector in the United States consumed 39 % (38.77 Quads) of primary energy use in 2006 (U.S. DOE 2009). Electricity represents the majority of the primary energy consumption (74%) in the buildings sector and is projected to increase 33% by 2030 (U.S. DOE 2009). The commercial buildings sector used 18 % (17.93 Quads) of the U.S. primary energy in 2006 with electricity accounting for 79 % (14.09 Quads). According to the forecast for 2030, electricity consumption will increase by 44 %. Office buildings have more floor area (12.2 billion ft^2) than any other commercial building type in the U.S. and have the highest total site energy consumption (1.1 Quads) of any building type (Shapiro 2009). In the United States, heating ventilation air conditioning (HVAC) systems use the following percentages of the total energy consumed in buildings: 12.7 % for cooling, 19.8 % for space heating and 2.8 % for ventilation of primary total energy consumed in buildings. Another significant end-use energy consumer in buildings is the lighting system which uses 17.7 % of primary total energy consumed in buildings (U.S. DOE 2009). In the Southern region of the U.S., commercial buildings consume 2.2 Quads. From this amount of energy 1.5 Quads is electricity (EIA 2009).

According to the Rocky Mountain Institute traditional approaches used in commercial buildings design are not adequate to save energy in both depth (savings per square foot) and scale (square footage improved each year) (RMI 2010). There are several ways to decrease energy consumption and CO_2 emission caused by generation of energy that is consumed in the building sector: more efficient HVAC system design, more efficient building insulation and fenestration systems, increase performance of

---

This thesis follows the style of *ASRHAE Transactions*.

existing buildings by retro commissioning (Liu et al. 2002) and retrofits, and efficient utilization of available renewable energy sources.

One goal of this thesis is to estimate the energy consumption reduction potential for existing building based on the example of Wisenbaker Engineering Research Center (WERC) located on the Texas A&M University campus in College Station. The other goal is to determine the potential HVAC system design strategy and maximum efficiency for a new building located in a hot and humid climate.

1.2. Purpose and objectives

The purpose of this study is to review the existing work related to building HVAC systems and investigate the opportunity for energy-efficiency improvements in existing and new commercial buildings in the hot and humid climate of Texas by using available building technologies and simulation software. Energy consumption trend data obtained during a building experiment along with simulation will also be used in this analysis and the results will be compared with the theoretical limit for efficient building energy performance.

The objectives of this research include:

- Investigating different approaches to HVAC system design and control, as well as application of renewable energy sources and case studies of highly energy efficient and zero-net buildings.
- Determining HVAC system designs that could be efficiently used in hot and humid climates based on previous research.
- Investigating possible applications of renewable energy systems in hot and humid climates.
- Creating a simulation model of an existing office building and evaluating the energy saving potential.
- Creating a simulation of a new office building and evaluating selected energy consumption improvement measures efficiency based on the measured energy consumption profile.

- Evaluating the theoretically possible minimum level of energy consumption by an office building providing the present level of comfort, indoor air quality, and services to occupants.

1.3. Significance of the work

The climate of College Station, Texas is hot and humid. Many other regions of the world have similar climates including other regions of North America, South America, the Middle East, and Asia. Therefore, the calculations are applicable to other areas. The intent of this work is to determine efficient HVAC design approaches and control strategies in hot and humid climate conditions.

1.4. Limitations of the work

Because of the time constraints and limitations in the capabilities of the software used, the following issues are not taken into consideration:

1. Simulation and analysis is limited to available computing software opportunities.

2. A hybrid geothermal heat pump system was not analyzed due to lack of knowledge about local ground properties.

3. An absorption chiller application was not considered in the work due to simulation software limits.

4. A displacement ventilation system was analyzed only in only general details since all available software programs assume isothermal surfaces and zero temperature gradients inside of the zone.

CHAPTER II

LITERATURE REVIEW

The literature review findings, discussed in this section, focus on efficient HVAC system design approaches in hot and humid climates of Southwest U.S. The review reflects the publication of real building case studies, modulation of individual systems or analytical building simulations. This chapter provides an overview of topics connected with advanced building technologies. The in-depth review of alternative HVAC systems have shown that most research has been done in two major areas: improving energy distribution systems and the utilization of low-grade natural energy resources. This literature review covers the following areas: dedicated outdoor air systems (DOAS), chilled ceiling and displacement ventilation (DV) systems, underfloor air distribution system (UFAD) with passive chilled beams, solar thermal and electrical generation, ground source heat pumps, cogeneration and absorption chilling application, and thermal energy storage systems.

High HVAC system energy efficiencies in hot and humid climates could be achieved by advanced building technologies: innovative HVAC system design, improved system control and utilization of renewable energy sources.

The main challenge in HVAC system design for hot and humid climates is achieving reliable humidity control with a high energy efficiency level. In most conventional HVAC systems, the humidity level is not directly controlled and is regulated separately from the sensible cooling load. During off-design conditions, when the sensible cooling load is small and the dehumidification load is high, humidity control can cause occupant discomfort, poor indoor air quality and excess energy use. Considering that buildings operate with average occupancies well below the design values most of the time (typical office space has a utilization factor of 0.85 (Brandemehl and Katejanekarn 2004)), this problem is very important. Conventional air conditioning systems cool indoor air (sensible cooling) and simultaneously dehumidify air (latent cooling) with latent capacity in the range of 20-30% of the total cooling capacity (Dieckmann et al. 2009).

The main sources of humidity problems in buildings are high levels of internal moisture generation and communications with humid outdoor conditions through infiltration and ventilation. Internal moisture generation in a building includes moisture generated by occupants and activities that involve water.

According to ASHRAE (2009) moisture generation by human occupants depends on the activity level. For commercial buildings, the latent load per person is specified between 105-250 Btu/hr.

Outdoor air can influence humidity by entering a building through cracks in the building envelope due to pressure differences across the envelope. To limit or prevent infiltration, commercial buildings typically operate at a positive pressure relative to outside air. Another way that outdoor air influences the humidity level is through the outside air (OA) ventilation. The value of OA ventilation required by ASHRAE Standard 62.1-2004 (ASHRAE 2004) is based on indoor air quality requirements and depends on occupancy level, floor area and area application. This is the main factor that influences latent load in office buildings. Increasing the OA ratio in hot and humid climates increases the building dehumidification load faster than sensible cooling requirements.

These factors became even more significant since energy efficiency improvements over time reduce the building sensible cooling load. Another important factor is improvements in air-conditioning equipment efficiency. Some of these improvements have been achieved by increasing the cooling coil relative size. This increases evaporator temperature and system efficiency, but at the same time decreases the dehumidification capacity (Brandemehl and Katejanekarn 2004).

Buildings in different climates could have different types of ventilation systems: mechanical ventilation, natural ventilation or mixed systems. Combinations of natural and mechanical ventilation in a hybrid system for some climate zones (Lyon, France or Copenhagen, Denmark) can provide 20% savings compared with a mechanical ventilation system (El Mankibi 2009). This strategy provides a good indoor environment at the lowest possible cost by utilizing simple techniques of controlling single parameters in one hierarchical fuzzy controller to eliminate contradictory strategies when the climate allows temporary natural conditioning (El Mankibi 2009).

In a hot and humid climate, ventilation and air-conditioning technologies for energy-efficient office buildings should provide another strategy. To achieve good dehumidifying performance of cooling coils and a high effectiveness of air distribution strategies, the HVAC system should be divided into two systems: a DOAS that provides the required amount of OA, and an air conditioning system that provides zone temperature control.

2.1. Dedicated outdoor air ventilation system

The dedicated outdoor air system with parallel sensible cooling began its development from the decoupled system concept, which can be summarized as a separation of ventilation and air-conditioning functions, or separation of sensible and latent load functions. The first system removes the latent load from the OA intake and supplies it to spaces using a 100% OA ventilation system (i.e. DOAS). The second system, removes the space sensible loads by using a parallel mechanical cooling system, such as fan coil units, conventional Variable Air Volume (VAV) systems, and/or ceiling radiant cooling panels operating independently from the ventilation system (Jeong and Mumma 2006).

For example, in the paper by Sekhar (2007), the DOAS includes a single coil twin-fan (SCTF) system employing a compartment cooling coil, desiccant dehumidification system and heat pipes for OA treatment. The air conditioning system uses an independent secondary ventilation and thermal cooling system with zone-based demand control, utilizing radiant chilled ceilings for a displacement air distribution system.

The dedicated outdoor air system conditions OA to the required off-coil conditions by a dedicated dehumidifying coil. The return chilled water (ChW) is used in a secondary coil of floor-based air-handling units to cover the building sensible cooling load of the recirculated air (Sekhar 2007). The dedicated outdoor air system application reduces off-coil dew-point temperature and substantially increases the dehumidification level. A heat recovery feature, like an air-to-air heat exchanger or wrap-around coil, gives the possibility of pre-cooling the incoming OA and additionally increases system efficiency. Even though advanced air conditioning systems demonstrate substantial variety, research in most of these areas “is in its infancy, and very limited practical applications exist for a holistic comparison and evaluation” (Sekhar 2007). This problem often makes energy cost benefit analysis of the various systems difficult.

High performance, low-energy building designs in cold climates that intend to achieve a reduction in energy use greater than 50% compared with Standard 90.1-2007 (ASHRAE 2007) require reductions in the heating and cooling loads, plus minimization of internal heat generation and maximization of natural lighting (Farzam and Todesco 2010). In a hot and humid climate this design may also include a DOAS, a DV system, energy recovery systems, and personal ventilation system.

The dedicated outdoor air system is a 100% outdoor air constant-volume system designed to deliver the volumetric flow rate of ventilation to each conditioned space (Sekhar 2007). A dedicated outdoor air system consists of a preheat coil, an enthalpy wheel, a deep cooling coil, a sensible heat exchanger, and the prime movers.

Dedicated outdoor air system, as defined by ASHRAE (2009), uses a separate unit to condition (heat, cool, humidify, dehumidify) all of the outdoor air brought into a building for ventilation, and then delivers it directly to occupied space or to local HVAC units serving those spaces (Mumma 2009).

According to the definition described in Mumma and Jeong (2005), key elements of DOAS are:

- One hundred percent outdoor air in quantities defined by ASHRAE Standard 62.1 supplied directly to each space of the building via its own system.
- No recirculated air used.
- For most situations, the DOAS is constant volume.

- Supply air dew-point temperature selected sufficiently low to enable it to handle the entire space latent load, thereby decoupling the space sensible and latent loads.

In most cases the DOAS relies on natural stratification of displacement air conditioning to transport OA with a temperature 63-68ºF (17-20ºC) and a low discharge velocity of 50-70 fpm (0.25-0.35 m/s) through the space from low sidewall or floor based outlets (Loudermilk 2005), or OA can be injected into the supply air stream through a series of nozzles (Loudermilk 2009). In order to maintain the required space humidity level of 50-60% RH, according to Loudermilk (2009), the overhead supply OA must be cooled to saturation temperature at 50-54ºF (10-12ºC). Another publication considers 55ºF to be sufficient (Simmonds et al. 2006).

The higher discharge temperature of a DV system requires cooled OA at saturation temperature to be reheated prior to introduction to the space. In mixing systems this is achieved by bypassing a portion of recirculated return air around the cooling coil and mixing it with the saturated air mixture prior to delivery to the space (Loudermilk 2005). To prevent supply air from bypassing the space served and provide stratification, the discharge air temperature should be lower than the room temperature. This condition requires use of a separate heating system (Loudermilk 2005) or significant internal thermal load. An induction nozzle within the displacement terminal can provide a solution for these issues in DOAS. To avoid condensation problems, cooled OA can be injected through a series of nozzles near the sensible cooling coil with supply air discharge temperatures of 62-68ºF (17-20ºC) (Loudermilk 2005).

The sensible cooling system should have a ChW temperature at least 1.5ºF (1K) higher than the space dew point to avoid condensation (Loudermilk 2005). Simmonds et al. (2006a) suggests controlling the panel surface temperature to 3ºF above the OA dew point if operable windows are used. To remove any condensation that might occur during start-up, the conditioning system can use condensation trays (Loudermilk 2005).

To avoid infiltration and provide sufficient building pressurization, sensor pressure control (usually 0.03 in H_2O) should be avoided. Instead, OA supply control should be based on an area value (Mumma 2010). The recommended value of OA supply for an office building is 0.07 $\mathrm{cfm/ft^2}$ or 0.28 $\mathrm{L/s \cdot m^2}$ and in general is closely related to the ASHRAE Standard 62.1-2004 OA floor area component (Mumma 2010).

2.2. Personal ventilation system

A personal ventilation system is designed to provide a local supply of air according to individual OA requirements. According to Sekhar (2007), ventilation in the breathing zone can be improved (compared with perfect mixing ventilation) by employing desk-mounted or floor-mounted air supply outlets. The personal ventilation concept is at the cutting edge of technological development and has tremendous potential in enhancing the acceptability of ventilation by supplying clean fresh air without mixing it with recirculated air.

2.3. Displacement air conditioning

Displacement ventilation systems could provide better thermal comfort and indoor air quality than conventional systems with considerably higher ventilation and energy efficiency, but depend to a large extent on relatively tall occupied enclosures (Sekhar 2007). According to Ghaddar et al. (2008), a chilled ceiling system in hot and humid climate requires ceiling height of 10 ft to ensure acceptable indoor comfort and air quality in the occupied zone. “In humid warm climates, designing the chilled ceiling DV system for a high stratification height might be quite costly” (Ghaddar et al. 2008). Displacement ventilation, used with or without a chilled ceiling, is characterized by distinct vertical temperature gradients and radiant asymmetry (Rees and Haves 2001). This prevents simulation of a DV system in the majority of simulation programs since they assume isothermal surfaces and well-mixed air.

A hydronic radiant cooling system combined with a constant volume ventilating system allows the air supply system to be downsized since it supplies only outdoor air for each person and provides dehumidification. A displacement ventilation system could utilize different hydronic technologies to provide cooling or heating for conditioned space by radiant chilled ceilings (Bahman et al. 2009), floor cooling systems (Simmonds et al. 2006b), chilled beams (Walker 2007), wall convective panels (Kilkis 2006), or concrete core cooling (Olesen et al. 2006).

The main advantage of hydronic radiant cooling is the ability to operate with a ChW supply temperature of 55.4ºF (Ameen and Mahmud 2005) (50-61ºF according to Dieckmann et al. (2004)) in contrast to the typical range of 42.8-44.6ºF for a conventional system (Ameen and Mahmud 2005) or 40-45ºF according to Dieckmann et al. (2004)). This change will increase chiller evaporator temperature and improve refrigeration cycle efficiency (Dieckmann et al. 2004). In addition, air fan power for a ventilation-only air supply is a fraction of that of the conventional system, while pump work for ChW circulation is about the same and the temperature elevation could utilize low grade energy sources such as solar or waste heat energy for cooling using absorption heat pumps. The hydronic radiant system that is used in DV also requires less space than a conventional air-conditioning system.

Building systems with radiant ceiling cooling/chilled ceiling (Weidner et al. 2009) systems, also known as “chilled beam” systems, incorporate pipes in the ceilings through which ChW flows. The pipes lie close to the ceiling surfaces or in panels and cool the room air via natural convection and radiation heat transfer (Dieckmann et al. 2004). Radiant cooled ceiling systems use cooling panels mounted within the ceiling to provide sensible cooling of the occupied space. The panels usually have a face panel compatible with the installed ceiling, which is combined with ChW tubing that circulates ChW through the back surface of the face. An exposed face of each radiant panel is maintained at approximately 60-65ºF (15.6-18.3ºC) that should be above the space dew-point temperature to avoid condensation (Weidner et al. 2009).

There are three principal types of overhang hydronic radiant cooling systems: a concrete core ceiling, lightweight metal ceiling radiant cooling panels, and capillary tube cooling grids embedded in plaster ceiling (Xia and Mumma 2006). There are two main practices in cooling panel construction - one is a dropped ceiling, or T grid type, and another is the hanging element type (Ameen and Mahmud 2005).

Displacement ventilation could be effective for cooling loads below $40\ \mathrm{W/m^2}$ (Rees and Haves 2001) and limited by a cold air draft in the occupied zone. The cooling load could be increased up to $120\ \mathrm{W/m^2}$ (Yuan et al. 1998) if the ventilation airflow rate is increased.

A combination of chilled ceiling and DV systems would be effective in cases if the cooling load is larger than $40\ \mathrm{W/m^2}$ (Rees and Haves 2001). Chilled ceilings with temperature 60.8-66.2ºF (16-19ºC) can provide cooling up to 100 $\mathrm{W/m^2}$ of floor area by a combined process of radiation and convection (Ameen and Mahmud 2005). Balancing of the two components is very important since increasing the chilled ceiling proportion will enhance the thermal comfort by decreasing the vertical temperature gradient and reduce indoor air quality at the same time, because of lower OA supply. An increase of the DV portion in heat removal will have the opposite effect (Ghaddar et al. 2008).

A publication dedicated to analytical analysis of ceiling radiant cooling panels performance was published by Mumma describing the governing heat transfer equations for a ceiling radiant cooling panel (Conroy and Mumma 2001; Xia and Mumma 2006). Design charts for chilled ceiling and DV systems based on a large number of simulations using a simplified transport plume multi-layer model of the chilled ceiling/displacement ventilation conditioned space were proposed by Ghaddar et al. (2008).

The main goal of typical chilled ceiling control strategies is prevention of condensation. It could be achieved by restricting the ceiling temperature. Chilled ceiling temperatures can be varied by changing the ChW flow rate or water temperature to keep it a few degrees above the measured space dew-point or the panel operation mode in a condition where humidity control has been achieved. One of the examples of this system application was described by Mossolly et al. (2008). The proposed system combines a chilled ceiling and a cooling coil for OA latent load removal with three three-way mixing valves and a constant volume pump for the chiller. The mixing of cooling coil discharge with chiller supply water eliminates need for chilled ceiling supply water heating. An alternative to this could be splitting cooling coil and ceiling panel loops.

To create the best indoor air quality, thermal comfort and energy efficiency utilizing chilled ceiling and DOAS should use either effective control strategies of ceiling temperature or DV air supply condition or both of them at the same time. This problem was studied by Mossolly et al. (2008) based on findings resulting from ASHRAE Research Project RP-1438 with a case study in Beirut (Lebanon).

For modeling of chilled ceiling panels, Mossolly divided the space into four horizontal and equal lumped air layers with their adjacent wall sections. The chilled ceiling temperature was assumed uniform, a thermal constant of less than three minutes and a constant overall heat transfer coefficient in the operational range 50-71.6ºF (10-22ºC). Models from Conroy and Mumma (2001) were used for the chilled ceiling panels. The cooling and dehumidifying coils were modeled after the lumped formulation approach used by Braun et al. (1989). The chiller performance model used COP determined from the Visual-DOE 4.0 Program Library (Visual DOE 4.0 software. 2005). The cooling flow for the chilled ceiling was constrained to be 1.5ºC higher than the space dew point and above 18ºC. Modeling results were close to the experimental data with less than 5% error.

Research results by Mossolly et al. (2008) show that simultaneous optimal control of both parameters provides 15% energy savings compared with a temperature control strategy and 10% savings compared with DV control.

Another paper by Bahman et al. (2009) based on findings from the same ASHRAE Research project RP-1438 describes the economic comparison of cooled ceiling/displacement ventilation and mixed air ventilation system in an office space located in the hot and dry climate of Kuwait. The case study used a 16.4x32.8x9.8 ft (5x10x3 m) office room. The fraction of the load removed by the chilled ceiling in July varied from 49 % to 55 %. The experiment results show that in Kuwaiti weather conditions, both systems have comparable levels of thermal comfort with lower radiant temperature of ceiling in case of application with a chilled ceiling DV system. The efficiency of a chilled ceiling DV system based on an experiment with 100 % OA was lower in comparison with a 30 % OA mixed ventilation system, and was 50 % higher than a 100 % OA mixed ventilation system. The main disadvantage of an installed chilled ceiling DV system is the high price of the energy recovery system (heat wheel). It was shown that the initial cost of the chilled ceiling DV system is higher, but the payback period based on transient operation is less than three years when used to replace a 100% fresh air conventional system. However, the chilled ceiling DV is not more efficient than a 30% fresh air conventional HVAC, but it provides a better indoor air quality.

The alternative to a chilled ceiling is a fan-driven convector used for both heating and cooling. A paper from Aalorg University in Denmark (Larsen et al. 2007) describes test results of this system in the cooling mode. The experimental unit consists of supply and return tubes along the room with fan driven convection through the ribs. During the experiment, a fan-driven convector was positioned in an upper corner of the test room (width - 13.8x11.8 ft (4.2x3.6 m) at a height of 2.5 m (8.2 ft)) along the ceiling. The experimental unit was able to cool up to 350W while maintaining comfortable conditions in the test room without air draft. The fresh air supply to the room in this experiment must be provided by another system. This study does not provide energy efficiency or acoustic noise generation comparison with mixing ventilation.

The floor radiant system could potentially remove up to $14.5\,\mathrm{Btu/(hr \cdot ft^2)}$ ($45\,\mathrm{W/m^2}$) (Simmonds et al. 2006b) when an operative design temperature of 78.8ºF (26ºC) is used. This result was obtained with a floor temperature of 66ºF (19ºC) which is in the acceptable range between 65ºF (18ºC) and 84ºF (29ºC) according to ASHRAE Standard 55-2004 - Thermal Environmental Conditions for Human Occupancy. Floor radiant system absorption potential for shortwave radiation could be even more, up to $80\,\mathrm{W/m^2}$ (Simmonds et al. 2006b). Since the temperature distribution on the floor is not equal, the ChW temperature should be controlled by a humidity sensor and be above the dew point temperature of the zone air. For example, for air with dry-bulb temperature of 75ºF (24ºC) and relative humidity of 50% (dew point temperature 55.1ºF), the minimum entering water temperature should be 55.4ºF (13ºC) (Simmonds et al. 2006b). The temperature drop of the water may be selected around 5K (9ºF). The system should provide constant water volume with variable entering temperatures to avoid problems with condensation and comfort.

Passive chilled beams use ChW piping circulation system through coil-like structures at or above the space's suspended ceiling or in space without a suspended ceiling. This unit operates similar to finned-tube convertors, except that the chilled beams operate based on the greater density of cooled air within the space, as opposed to the finned-tube convectors' reliance on the lesser density of the heated air relative to the air within the space (Weidner et al. 2009). The condensation temperature limit applies to a chilled beam as well as to any hydronic cooling system.

Chilled beams operate according to the following principle: the OA is cooled and dehumidified by the central air handling system and ducted to the distribution plenum within the beam from which it is injected through a series of nozzles. The outside air is mixed with recirculated air before it is discharged to the space. The volume of induced room air is typically two to five times that of the primary air, depending on the induction nozzle (Loudermilk 2009). The ratio of the induced airflow rate to the primary airflow rate is referred to as the beam's induction ratio. The sensible cooling coil within the beam is maintained above the room air dew point and supplies 50-70% of the required space sensible heat removal. Chilled beams should not be used in low ceiling height applications where the distance between the ceiling and the top of the occupied zone is less than 3 ft (0.9m) (Loudermilk 2009).

The article by Weidner et al. (2009) describes an efficiency comparison of two alternative systems for office space cooling: a system using a combination of radiant cooled ceiling and underfloor air distribution versus a system using a combination of chilled beams and underfloor air-distribution.

The main factors against radiant cooled ceiling in this design were initial cost and aesthetic considerations, since to provide a cooling of 30 $\mathrm{Btu/h \cdot ft^2}$ ($95\ \mathrm{W/m^2}$) would require covering the entire ceiling with chilled panels.

Chilled beams were able to provide up to 180 Btu/h per lineal foot (173 W per lineal meter) of cooling and had a lower cost relative to radiant cooled ceilings. Chilled water used in chilled beams was return water from dehumidification air handling coils.

The economic evaluation shows that chilled beam application could provide an operating cost payback in less than two years compared with the more than 50 years of the chilled ceiling (Weidner et al. 2009).

The underfloor air distribution system (UFAD), as well as chilled beams, supplies cold air from floor-mounted diffusers, but in comparison, the UFAD supplies air at a high velocity to achieve mixing in the occupied zone and stratification in the upper zone. This can cause potential discomfort near the supply outlet and would require either a clearance zone or reducing the supply air speed with an increasing number of outlets (Sekhar 2007). The supply air dry-bulb temperature in the UFAD system is higher than in a conventional ducted system is 62°F (16.7°C) with UFAD versus 55°F (12.8°C) in conventional systems (Weidner et al. 2009).

The combination of UFAD and chilled beams described by Weidner et al. (2009) required reheat for supply air that was provided by bypassing a portion of the return airflow around the cooling coil and mixing it with the cooling coil leaving the airstream prior to entering the draw-through supply fan. The cooling coil reduces the entering air temperature from 87°F db/77°F wb (30.6°C db/25.0°C wb) to an exit temperature condition of 50°F db/49°F wb (10°C db/9.4°C wb). The air is cooled and after the coil mixes with 81°F db/63°F wb (27.2°C db/17.2°C wb) return air (estimated 79°F db (26.1°C db) return plenum temperature plus 2°F db ((1.1°C db) fan heat), at a ratio of approximately 66% coil air to 34% bypassed return air. This yield a supply air temperature of 61°F db/54°F wb (16.1°C db/12.2°C wb), after the supply fan heat and supply duct pickup (Weidner et al. 2009). The system (Weidner et al. 2009) incorporates carbon dioxide monitors within the occupied spaces to control OA supply that minimizes energy consumption while ensuring that ventilation air requirements are satisfied. The automatic dry- and wet-bulb temperature controls were used for two-way control of chilled beam secondary loop to maintain entering water temperature at not less than 3ºF

(1.7ºC) above space dew-point temperature. The installed system energy consumption was 42% less than a traditional overhead VAV system (1.89 kWh/ft^2 vs. 3.22 kWh/ft^2).

The application of underfloor air DV system with chilled ceiling, perimeter trench heaters and perimeter chilled beams in a London (United Kingdom) office building is described in a publication by Walker et al. (2007). The system was designed to supply 100% OA by a constant-volume air-handling unit during occupied hours. Chilled ceiling panels provide additional cooling to cover internal loads and chilled beams provide additional cooling near perimeter windows due to solar gains. The trench heaters are used at the perimeter to overcome conduction heat losses through windows and OA infiltration during the winter.

2.4. Heat recovery and dehumidification system

Energy consumption requirements for OA treatment could be substantially reduced by using air-to-air heat or energy exchangers. The dehumidification load accounts for an estimated one-third of the primary energy consumption for air conditioning in the Southern region of the U.S. The Southern region accounts for more than 60 % of the total U.S. cooling energy consumption (Dieckmann et al. 2009). Traditional methods of air dehumidification require water vapor condensation on the evaporator coil; this can potentially create a problem from fungi growth possibilities in the drain. Application of enthalpy wheels or other energy recovery systems can reduce cooling requirements associated with air dehumidification by directly transferring water between outside and exhaust air.

2.4.1. Air-to-air heat recovery and dehumidification system

Air-to-air heat or energy exchangers recover sensible heat and/or moisture from an airstream at a high temperature or humidity to an airstream at a low temperature or humidity (ASHRAE 2008).

The main factors that determine energy/heat recovery efficiency are outside airflow, climate conditions, length of operation, recovery system parameters and cost.

Energy can be recovered either in its sensible (heat recovery) or latent form, or a combination of both (enthalpy/energy recovery) from multiple sources (ASHRAE 2008).

Air-to-air heat exchangers for a new building have payback durations of one to five years in warm humid climates (Besant and Simonson 2000), and for enthalpy wheels, the payback could be even smaller (less than 3 years (Sekhar 2007)). Typical heat and enthalpy exchanger efficiencies range from 55 % to almost 80 % (Dieckmann et al. 2003). To operate effectively energy and enthalpy exchangers must meet two key requirements:

1. The exhaust flow rate should be equal to a significant fraction of the outside airflow rate (>75%);

2. The temperature and humidity (for an enthalpy recovery system) of the exhaust air must be close to that of the conditioned space (Dieckmann et al. 2003).

Air-to-air heat exchangers should be bypassed or have significantly decreased rotational speed (for enthalpy/energy wheels) in the case of economizer operations. The main types of energy recovery systems are: plate-fin (cross-flow) arrangements, regenerative rotary wheels (heat or enthalpy), heat pipes, heat pump, runaround loops and thermosiphons.

Plate-fin arrangements transfer only sensible heat between the makeup and exhaust airstreams. Plate-fin heat recovery ventilation utilizes exhaust air from the building interior that passes through one side of the exchanger, counterflow to the incoming makeup air, which passes through the other side of the exchanger. This provides pre-cooling the incoming OA during the cooling mode. There are no commercialized plate-fin arrangements that would be able to transfer moisture (Dieckmann et al. 2003).

Heat pipes are passive devices that recover energy from cold exhaust air to pre-cool the incoming outdoor air. The main purpose of a heat pipe is the transportation of a large amount of energy over its length with a small temperature drop. Energy is transferred through liquid evaporation in one section of a heat pipe and condensation in another with vapor movement in one direction and liquid in the other. Heat pipes are mostly used in sensible heat recovery devices, but could be used as well for pre-cooling in a dehumidification system. Heat pipes can provide heat transfer rate up to 1000 times greater than through copper (ASHRAE 2008). The main heat pipe types are conventional heat pipes, heat pipe panels, vapor-dynamic thermosiphons, sorption heat pipes, loop heat pipes, and micro heat pipes.

Heat and enthalpy wheels are slowly rotating discs made of thin metal, plastic, paper or ceramic surfaces, such as honeycomb or a random woven screen mesh, which create large surface area. They have a revolving cylinder filled with an air-permeable medium which has a large internal surface area. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (ASHRAE 2008).

Enthalpy wheels have the same construction as heat wheels, but with the addition of desiccant material adhered to the matrix material of the wheel surface. The desiccant material provides total enthalpy transfer that includes heat and mass (moisture) transfer, increase recovery system operation time and gives significant benefits in hot and humid climates or in systems with high OA requirements. A good example of a desiccant material that could be used is silica gel or dry lithium chloride mixed with zeolites granulated and packed into a bed (Sekhar 2007). According to Sekhar (2007) the most important components for performance of an enthalpy wheel are: inlet air temperature, moisture content, velocity and face of the desiccant bed. The desiccant material in an OA dehumidification system should be periodically reactivated with dry exhaust or dedicated hot air. Application of an enthalpy wheel is favored generally if the humidity ratio is greater than 0.012 kg/kg (Besant and Simonson 2000).

Runaround loops (coil energy recovery) place extended surface, finned-tube water coils in the supply and exhaust airstreams of a building. The coils are connected in a closed configuration by counterflow piping through which an intermediate heat transfer fluid (typically water or antifreeze solution) is pumped (ASHRAE 2008).

Air dehumidification could also be provided with low-grade solar energy dehumidification systems that separate the control of sensible and latent cooling load. The dehumidification system consists of an enthalpy wheel with reactivation by solar energy heated air. Application of a solar dehumidification system in combination with a chilled ceiling in hot and humid climates is described in Ameen and Mahmud (2005). The combination of desiccant dehumidification and hydronic radiant cooling is a relatively new concept and for now not economically viable as standalone system (Ameen and Mahmud 2005). This combination allows the cooling system to have its temperature higher than the dew-point air temperature resulting in the opportunity to downsize the refrigeration system and in addition to provide cooling directly and more evenly to the occupants without causing drafts. A desiccant dehumidification system that supplies dehumidified and precooled air could cover additional cooling requirements that have not been met by the radiant cooling system and avoids potential condensation problems in hot and humid climates. The main disadvantage of desiccant systems is the temperature rise of the desiccated air that should be recovered by heat pumps (Ameen and Mahmud 2005).

An alternative option is utilization of a heat pump for heating air to reactivate the enthalpy wheel. Application of a cooling system with a heat pump used for enthalpy wheel reactivation in hot and humid climates is described in Tsay et al. (2006). Conventional systems that utilize the common approach in enthalpy wheel reactivation with exhaust air have a coefficient of performance below one (Tsay et al. 2006). The proposed system uses exhaust heat from the condenser of a heat pump as the regeneration heat source of the desiccant dehumidifier while simultaneously absorbing heat from OA intake. An experimental system in Chiba City, Japan had a floor area of $180\ \text{ft}^2$ $(16.7\ \text{m}^2)$ and a volume of $1356\ \text{ft}^3$ $(38.4\ \text{m}^3)$ with good insulation toward the ground. The system simulated thermal load was 1200 W and a moisture load of 2.96 lb/h (1340 g/h). During the experiment, the enthalpy wheel reactivation temperature was 140-158ºF (60-70ºC) and could be lowered to 124ºF (51.3ºC) providing good indoor air quality conditions. During COP estimation, energy consumption of the fans and pumps were ignored. For the system with desiccant and sensible heat exchanger rotors, the COP values for the of regeneration heating power were 0.67-0.8. For the system with just a desiccant heat exchanger rotor, the COP values for the regeneration heating power was

0.43 and COP based on heating and cooling energy input was 0.6. In the case when heat supplied from the condenser of the heat pump completely substituted for the regeneration heater, the COP reaches 2.32 in cases when COP of the heat pump is 3.1-3.3.

A heat pump could also be used for ventilation heat recovery. During the cooling season, the heat pump recovers energy to cool and possibly dehumidify OA entering the system. Rejected heat goes to the exhaust airstream from the building. Generally, the exhaust air is cooler than the return ventilation air during the cooling season and warmer than the return ventilation air during the heating season. One of the main requirements to justify heat pump applications is low motor and compressor energy consumption. Evaluations of the heat pump ventilation energy recovery application for small commercial buildings are based on manufacturers and laboratory data for retrofit applications provided in Mercer and Braun (2005). The application of the heat pump was compared to a base case that does not employ energy recovery, but utilizes fixed minimum ventilation rates with an enthalpy economizer. According to Mercer and Braun (2005), heat pumps provide relatively small savings in hot or cold climates and are less efficient than enthalpy exchangers and demand controlled ventilation.

2.4.2. Dedicated heat recovery systems

An air conditioning unit in the cooling mode removes heat from an interior area and rejects it to the ambient. The energy flow rejected from the condenser of a small scroll or screw chiller can have a temperature up to 140°F (Durkin and Rishel 2003) that creates the opportunity for energy recovery and utilization of rejected energy for air preheating, heating, reheating systems or domestic water heating systems (Liesen 2004). These systems are called “dedicated” heat recovery because 100% of the heat generated by the dedicated heat recovery chiller can be used for water heating applications (Durkin and Rishel 2003). The heat theoretically available for recovery consists of the heat removed from the space and the heat of compression of the refrigerant in the compression cycle.

If a reciprocating chiller operates at full capacity and removes 20 RT of energy, it rejects about 24 RT of heat and 12.8 RT could be recovered by transferring it to the water. In other words, 70% of the heat removed from the space could be recovered (Jarnagin 2010).

Dedicated heat recovery systems can be used when hot water and cooling are used simultaneously, for example in buildings that require supply air reheat or consume large amounts of domestic water throughout the year.

Dedicated heat recovery can be utilized in three system types: heat recovery chillers, heat pumps or independent heat recovery units for smaller systems. Heat pumps used for heat recovery can produce 140-150ºF (60-65ºC) hot water using condenser heat with a coefficient of performance between 3.0 and 5.0 (McQuay International 2002).

The economic efficiency of a dedicated heat recovery system depends on a simultaneous need for cooling and hot water below 130ºF (54°C). The maximum efficiency could be obtained when dedicated heat recovery system is sized by the side of the system that has the lowest demand (simultaneous heating or cooling requirements). In addition dedicated heat recovery can slightly increase the capacity of the air conditioning unit. When there is no heating requirement, the dedicated heat recovery system could be bypassed. In addition, dedicated heat recovery slightly increases the capacity of the air conditioning unit.

2.5. Internal loads minimization potential

The main sources of sensible heat load in office buildings are the lighting system, office plug loads and the load from human occupants. ASHRAE Research project RP-822 results show that the level of general office equipment energy consumption varies over a small range and can be generalized, while results for equipment in laboratories and hospitals show the load varies widely and should not be generalized (Wilkins and Hosni 2000).

Using the sum of the total nameplate power values of office equipment does not provide a good estimate of equipment sensible load (Wilkins and Hosni 2000). Heat load from personal computers, monitors, laser printers, and copiers significantly depends on the model, whether it is Energy Star® certified and whether they use energy saving settings in addition to their maximum power consumption. The office plug load diversity factor also depends on specific characteristics, such as job schedules and the amount of employee travel.

Research completed 10 years ago shows that actual heat gain per unit area from equipment in office space ranged from 0.44 $\mathrm{W/ft^2}$ to 1.05 $\mathrm{W/ft^2}$ with an average power density 0.81 $\mathrm{W/ft^2}$ (Wilkins and Hosni 2000). This value is comparable to the 1997 estimates of 0.4-1.1 $\mathrm{W/ft^2}$ with a mean value 0.83 $\mathrm{W/ft^2}$ for office space heat gain per unit area by Komor (1997).

The lighting system in a commercial building typically consumes about 33% of the energy used (Liebel and Brodrick 2005). There are three ways to increase the energy efficiency of lighting systems: lighting system efficiency, lighting design, and lighting controls. The main factors that influence lighting system efficiency (efficacy) are: lamps, ballasts and luminaries. Efficient lighting system design should also include other important factors including lamp replacement and system maintenance costs.

Using T8 lamps with electronic ballast instead of T12s provides a minimum of 25% energy savings and in addition provides better light output retention over the life of the system (Liebel and Brodrick 2005). An additional source of potential savings is reduction of lighting levels from 55-115 foot-candles (592-1238 lux) to 30-70 foot-candles (323-753 lux) (Shapiro 2009) as recommended for office areas by IES Lighting Handbook (2000). Motion control systems should also be installed, at least in rooms not very often used, together with daylight sensors to reduce energy consumption (Plesser et al. 2008).

An efficient T8 lamp (32W/lamp) with electronic-extra efficient instant start ballast can provide 101.8 Lumens/W that is 57% more efficient than a T12-F34CW lamp (34W/lamp) with magnetic - rapid start ballasts providing 64.8 Lumens/W. Application of new lighting technologies can reduce lighting power density to around 0.90 $\mathrm{W/ft^2}$ (10 $\mathrm{W/m^2}$) or less in standard office applications (Liebel and Brodrick 2005) compared with 1.0 W/ft^2 by ASHRAE Standard 90.1-2007. The reduction of lighting power density will also provide simultaneous reduction of sensible cooling load. Lighting power density could be reduced even further to 0.75 W/ft^2 ($8W/m^2$) with the best fluorescent technologies (Farzam and Todesco 2010; Shapiro 2009).

2.6. Solar energy systems

One of the most widely available renewable energy sources that potentially could be used in buildings is solar energy. Solar energy is becoming a more economically available source as the costs of energy continue to increase and the cost efficiency of solar technologies improves. Low-density solar energy could be used for heating, cooling or for electricity generation (photovoltaic systems) (ASHRAE 2008).

Commercial and industrial solar-thermal energy systems can be divided in two groups according to the heat transfer medium used in the collector loop (air or liquid). Air systems are mostly used for forced air space heating and could be used for dehumidification processes (regeneration of enthalpy wheels (Ameen and Mahmud 2005)). This system usually does not have an energy storage capability. Air-solar systems circulate conditioning air through ducts to and from an air-heating collector. High effectiveness is achieved by using a direct air heating process without a heat exchanger. This system does not require protection against freezing, overheating, or corrosion (ASHRAE 2008). The main disadvantage of this type of system is higher fan energy consumption compared with pumps and thermal energy storage system utilization.

Liquid systems have a much bigger variety of applications including domestic water heating, hydronic space heating, energizing heat-driven air conditioning or as a source for heat pumps (ASHRAE 2008). Because of the larger variety of applications, they are much more common than air systems. Liquid systems circulate a liquid, often a water-based fluid, through a solar collector.

The liquid should be protected against freezing that could occur at air temperatures well above 32°F (ASHRAE 2008) and in addition they require over-temperature protection to ensure that the system operates within safe limits and to prevent collector fluid from corroding the absorber or heat exchanger. Liquid-solar systems are of two types: direct liquid systems (with water circulating through the collector) and indirect systems (with the collector loop separated from the water system by a heat exchanger). In commercial applications, indirect liquid systems with nonfreezing fluid freeze protection are used most commonly.

Solar collectors operate based on air heating, liquid heating, or liquid-vapor phase change to transfer heat. The flat-plate collector is the most common type of low-temperature (below 200°F (93ºC)) commercial solar collector (ASHRAE 2008). A flat-plate collector contains an absorber plate covered with a black coating and one or more transparent covers. The covers are transparent to incoming solar radiation and relatively opaque to outgoing (long-wave) radiation, but their principal purpose is to reduce convection heat loss (ASHRAE 2008). The collector box is insulated to prevent conduction heat loss from the back and edge of the absorber plate. This type of collector can supply hot water or air at temperatures up to 200°F, although efficiency diminishes rapidly above 160°F (ASHRAE 2008). The main advantages of this system are: simple construction, low relative cost, no moving parts, and durability. Evacuated tube collectors utilize an absorber mechanism (heat pipe or copper fin tube on a copper sheet) encased in a glass vacuum tube (ASHRAE 2008). Both types of collectors collect both direct and diffuse radiation. Evacuated tube collectors can operate with the higher operating temperatures required in heat-driven air-conditioning equipment.

Solar energy applications for cooling systems are not popular mainly because of the large initial investment required in solar collectors and the extremely high prices for absorption chiller of less than 15 kW capacity (Castro et al. 2008). The application of an air-cooled, hot-water-driven absorption chiller that operates on solar energy is described in Castro et al. (2008). The main improvement of the proposed system is air-cooling of the absorber and condenser. A pre-industrial prototype was created with a capacity of 7 kW, electricity consumption of 0.62 kW and COP of 0.7.

Photovoltaic (PV) systems utilize a large surface of semiconductor materials with p-n junctions that convert light from the sun directly into electricity. These cells are connected in series and parallel strings and packaged into modules to produce a specific voltage and current when illuminated (ASHRAE 2008). Photovoltaic systems can be designed as a stand-alone systems (located at sites where utility power is unavailable) or utility-interconnected systems.

A PV cell consists of the active PV material, metal grids, antireflection coatings, and supporting material. The complete cell is optimized to maximize both the amount of light entering the cell and the power out of the cell (ASHRAE 2008). A group of PV cells connected together are called a PV module and a group of modules connected together are called a solar panel.

The efficiency of a PV cell is the maximum electrical power output divided by the incident light power. It is commonly reported for a PV cell temperature of 77°F (25°C) and incident light at an irradiance of $92.94\ \mathrm{W/ft^2}$ with a spectrum of 1.5 atmospheres equivalent spectral absorption. The most commonly produced PV modules are made of crystalline silicon, either single crystal or polycrystalline (ASHRAE 2008).

The energy efficiency of a PV cell varies greatly between different commercially available technologies. Available multicrystalline silicone PV cells typically have efficiencies between 16-20% and cheaper thin film technologies may deliver efficiencies of around 10% (Phillips et al. 2009). The efficiency of three-junction concentrator PV-panels can reach 36% (Phillips et al. 2009). To maximize electric generation, an optimum orientation and tilt or a tracking system can be used. The position of PV panels can be optimized for summer-only, winter-only or annual energy production using low tilt, high tilt or latitude tilt angles, respectively.

A significant number of publications are dedicated to PV cell system applications in energy efficient buildings. In most publications, the design approach was to achieve net zero energy or some PV generation level for demonstration purposes regardless of cost (Phillips et al. 2009; Xu et al. 2007; Pless et al. 2006; Steinbock et al. 2007). In California, a 4 kW PV system cost around $32,000 that was reduced by a tax credit for an end user in the residential sector to $23,000 in 2009 (USA Today 2009). According to publications that were reviewed PV system payback time without government support is exceeding system lifetime period.

2.7. Heat pump systems

A heat pump is a machine or device that extracts heat from a source and transfers it to a sink at a higher temperature. According to this definition, all pieces of refrigeration equipment, including air conditioners and chillers with refrigeration cycles, are heat pumps. In engineering, however, the term heat pump is generally reserved for equipment that heats for beneficial purposes, rather than that which removes heat for cooling only (ASHRAE 2008). Dual-mode heat pumps could provide both heating and cooling for the building. Heat pumps could use one or more reciprocating or screw compressors or staged centrifugal compressors. Compression can be single-stage or multistage. Frequently, heating and cooling are supplied simultaneously to separate zones. Ground, well water, surface water, gray water, solar energy, the air, and internal building heat can be used as a heat source. Choice of heat source depends on geographic location, climate, initial cost availability, and type of building structure.

Air is most commonly used as a heat source or sink. Heat pumps can utilize ambient air or exhaust air from the building ventilation system. The main limiting factor for ambient air utilization in hot and humid climates is the local outdoor air temperature. As the outdoor temperature increases, the cooling capacity of an air-source heat pump decreases. Exhaust air from the building ventilation system can be used if a large amount of air is exhausted.

Water can be used as another source. Heat pumps can use ground water, surface water (lakes, rivers), city tap water, cooling towers, closed loop systems or waste water.

Heat pumps could utilize solar collectors in combination with an absorption chiller for building cooling (Castro et al. 2008). Solar energy could be used as the primary heat source or in combination with other sources.

Solar-source heat pumps used for heating are of two main types: direct and indirect. Direct system heat pumps have their refrigerant evaporator tubes in a solar collector. This system also can operate in cooling mode using heat exchanger in solar collector as a condenser and rejecting energy to outdoor air. An indirect system use water or air to transfer heat from solar collector to evaporator.

One of the most common energy sources is natural gas. Zaltash et al. (2008) described performance of a 10 RT natural gas engine-driven heat pump rooftop unit. The unit COP in the heating mode exceeds 1.6 at the 47ºF (8.3ºC) rating condition and in cooling mode exceeds 1.2 at the 95ºF (35ºC) rating condition. This type of system is popular in Japan where they have typically sold 40,000 systems per year (Yahagi et al. 2006) and has a 22.3% share of the entire air-conditioning market nationwide excluding residential use (Takahashi 2006). In Japan, absorption-type cooling and heating systems are widely used in large buildings, and gas engine heat pump-type systems are popular in medium and small buildings.

The experiment described in Zaltash et al. (2008) included a cooling mode test at 120ºF (48.9ºC) OA temperature that showed a gas cooling COP of 1.22 and a capacity of 118,322 Btu/h (9.4RT or 34.7kW). In heating mode: "the gas COP at the 47ºF (8.3ºC) rating conditions exceeded the goal of 1.6 COP (Zaltash et al. 2008). The comparison show a 7% efficiency decrease compared to electrical chillers (COP 3.52/ EER 12.0) and 0.32 electrical energy generation efficiency calculated from national average electricity losses based on higher heating value (Zaltash et al. 2008).

The main advantage of a gas engine-driven heat pump system is its ability to reduce electrical system peak demand. For example, a 10 RT electric rooftop unit with an EER of 12.0 and power consumption of 9.9 kW could be replaced with a gas driven heat pump (electrical energy consumption of 1.5 kW at 95ºF). This change saves 85% of the electric rooftop unit electrical energy demand.

Ground storage can be used as a heat sink or source through buried heat exchangers for a heat pump system. The efficiency of ground heat exchangers (GHX) is determined by soil parameters (O'Neal et al. 1994). The primary factor that influences heat transfer efficiency is thermal diffusivity. This parameter is hard to determine without local soil data. Thermal diffusivity is defined as the ratio of thermal conductivity to the product of density and specific heat. Conductivity greatly depends upon the soil's moisture content. Ground heat pumps can be groundwater, direct expansion (the ground-to-refrigerant heat exchanger is buried underground), or ground-coupled (closed loop heat exchanger). Ground-coupled heat pumps may have a refrigerant-to-water heat exchanger or could be direct-expansion. Groundwater heat pumps can either circulate source water directly to the heat pump or use an intermediate fluid in a closed loop, similar to the ground-coupled heat pump (ASHRAE 2008).

A ground heat exchanger could be positioned either horizontally or vertically. A horizontal system consists of single or multiple serpentine heat exchanger pipes buried 3 to 6 ft apart in a horizontal plane at a depth 3 to 6 ft below grade that determined by excavation costs and temperature (ASHRAE 2008). A vertical system uses a concentric tube or U-tube heat exchanger.

The significant benefit of ground source heat pumps is that there is much less variability of ground temperatures at about 3 ft depth than ambient air temperatures. Ground temperature is higher than air temperature during the winter months and lower than air temperature during the summer. As a result of the lower ground temperature of condenser during the cooling mode, the energy efficiency and cooling capacity are increased.

For commercial buildings in warm climates, the required cooling geothermal heat sink capacity is much greater than the required heat source capacity in the heating mode. For ground heat pump systems using closed-loop vertical GHX, this load imbalance can result in a ground temperature increase over time causing system performance deterioration (U.S. DOE 2001). Peak entering fluid temperature above 110ºF (43.3ºC) should be avoided to prevent degradation of heat pump performance (Yavuzturk, and Spitler 2000).

One possible solution for this problem is increasing the distance between adjacent heat exchanger boreholes or increasing the size of the GHX, which will result in higher system costs.

Another possible solution for this problem is the application of a hybrid geothermal heat pump system that combines a GHX with an additional auxiliary heat rejecter (cooling towers (Yavuzturk, and Spitler 2000), cooling ponds (Ramamoorthy et al. 2001), fluid coolers, or surface heat rejecters or some other option). In Yavuzturk and Spitler's (2000) case study, a cooling tower is used to handle the excess heat rejection loads during building cooling.

Open-circuit cooling towers and closed-circuit fluid are commonly used for supplemental heat rejection in ground-source heat pump systems (Yavuzturk and Spitler 2000). Open-circuit cooling towers are typically used in conjunction with isolation plate exchangers in order to avoid mixing of the loop heat transfer fluid and the cooling water.

In the weather conditions of Houston, Texas a hybrid geothermal heat pump system appears to be beneficial on both the initial cost and the annual operation cost (Yavuzturk and Spitler 2000). The size of the heat pump should be sufficient to cover the building heating requirements and vary depending from climate.

Closed loop heat pumps are the most common since they do not have special requirements for soil parameters. The closed loop of a GHX system may serve one or many heat pumps, depending on the application with additional heat rejection to wet or dry cooling tower (U.S. DOE 2001).

More stable geothermal temperatures allow compressors in the individual heat pump units of a ground-source heat pump system to operate much more efficiently than those in air source units. Utilization of a common water loop provides heating and cooling in different parts of a building simultaneously without energy transfer to the ground. In addition, a common water loop could be used for heat pump domestic water heating that also reduce thermal load of ground storage.

Application of hybrid geothermal heat pump systems combined with existing water tower can result in system cost reductions of more than 50% even when the building is not overly cooling load dominated (U.S. DOE 2001).

Yavuzturk and Spitler (2000) consider the application of control schedules for ground-source heat pumps for cooling-dominated commercial buildings with cooling towers as supplemental heat rejecters to reduce the initial cost of the system and to improve system performance. Simulations were developed in TRNSYS for hybrid geothermal heat pump systems used for an office building with an area 14,205 $\mathrm{ft}^2\,(1320\,\mathrm{m}^2)$ in the hot and humid conditions of Houston, Texas. The simulated system configurations include water tower heat rejection before the ground loop heat exchanger. The calculations assumed GHX prices at \$6.00 per foot of the borehole and 20 years operation.

In this simulation, a base case heat pump with ground-loop heat exchanger was designed to provide all required cooling, while alternative cases considered additional applications of a cooling tower with different control schedules. The most efficient system, according to the simulation is the application of a cooling tower with 5300 cfm of air flow rate and activation of the cooling tower fan and the secondary circulation fluid loop pump whenever the difference between the heat pump exiting fluid temperature and the ambient air wet-bulb temperature is greater than 3.6ºF (2.0ºC). The cooling tower fan and the secondary fluid circulation loop pump are turned off when this difference is less than 2.7ºF (1.5ºC). This control strategy takes advantage of the storage capacity of the GHX by "storing cold" in the ground during the winter. The application of the water tower and the afore-mentioned control schedule in Houston, Texas weather conditions reduced the present value of total cost from \$ 86,062 to \$42,803 or 50 % and surface area of the borehole field from 3906 ft^2 (361.1 m^2) to 937 ft^2 (87.1 m^2) or 76 % savings. The authors of both case studies (U.S. DOE 2001; Yavuzturk and Spitler 2000) point out that none of the hybrid system designs used for control simulations have been optimized.

Two publications are dedicated to the design methodology of a ground coupled heat pump (O’Neal et al. 1994; Hackel et al. 2009). Article (O’Neal et al. 1994) describes a technique for the sizing of vertical U-tube ground coupled heat pump heat exchangers for Texas climates based on a transient simulation model of a ground coupled heat pump and measured weather and soil data. The most important factors for sizing hybrid ground coupled heat pump heat exchangers were found to be soil temperature and thermal conductivity. The other factors that should be considered are ground density, entering water temperatures, and indoor air temperature.

The result of proposed sizing method was compared to methods developed by the National Water Well Association and the International Ground Source Heat Pump Association. Simulation comparison was made with constant soil, building and heat pump characteristics. For Houston conditions, simulation by the National Water Well Association method required a heat exchanger length of 370 ft, the International Ground Source Heat Pump Association method required a length of 465 ft, the method proposed by O’Neal et al. (1994), required a length of just 290 ft. According to O’Neal et al. (1994) design parameters could be very different even in same location because thermal conductivity could significantly vary depending on the type of soil.

The simulation study (Hackel et al. 2009) based on findings from ASHRAE Research Project RP-1384 investigated the benefits of using hybrid ground-coupled heat pump system and aimed to determine optimal system design (size and control strategy) to minimize lifecycle cost for different climate and building types. Analysis was made on sub-hourly basis and a period of time equal to 20 years with a hybrid ground coupled heat pump system that meet building loads at all times and cooling tower upstream of, and in series with the GHX. Vertical GHX with one U-tube per bore were used in the simulation (the most common configuration in the U.S.). The ground heat exchanger was positioned upstream of a cooling tower with a bypass that should provide better efficiency in hot and humid climate (Hackel et al. 2009). A supplemental cooling system used closed-circuit cooling tower with water evaporation on the outside surface of tubes and a dry cooler. Simulation was made for different types of buildings including an office with 127,000 ft^2 (11,811 m^2) area in St. Louise, Seattle and Phoenix.

According to this study economical parameters do not significantly influence optimal design and GHX should be sized to a possible smallest size to meet low temperature limits. In most moderate and southern climates hybrid ground-coupled systems have a lower lifecycle cost than other options because of higher efficiency in evaporative cooling (Hackel et al. 2009). Hybrid systems are significantly more efficient than GHX only systems (\$7/ft^2 in Atlanta and \$3/ft^2 in St. Louis) and \$1-2 ft^2 more efficient than (\$11-22 ft^2) boiler/tower systems, assuming GHX cost around 6\$/ft.

Another publication by Landsberg et al. (2008) describes the operation of ground source heat pumps with cooling dominated loads (data centers) combined with evaporative coolers. The simulated building ventilation system used an economizer cycle during the winter. For conditions when outdoor temperature was below 55°F, the GHX living water temperature was above 45°F and the water temperature leaving the evaporative coolers is less than the temperature leaving the GHX, the paper proposed charging GHX. As an another option to decrease the ground load, its proposed to increase the distance between vertical wells to 20-25 ft from 15 ft and to provide heat rejection to the ground during only warm periods. An additional benefit of this system for data centers is possibility of less than an hour potential source of backup cooling provided by GHXs.

For office buildings below three floors, Sekine et al. (2007) describes a commercially viable geothermal heat pump application that uses the cast-in-place concrete pile foundation of a building as a heat exchanger to reduce the initial boring cost. Traditionally energy pile systems have used precast, pre-stressed concrete piles or still pipe piles with tubes inside the piles. A full scale experiment was made on site at the University of Tokyo in Chiba for an office building that operated five days a week. The result of the experiment that used cast-in-place concrete piles with tubes positioned outside the diameter of the pile show the heat rejection rate 186-201 W/m of the pile with an average COP 3.7 (while cooling). The initial system cost was reduced from

$3/W to $0.79/W compared with standard borehole systems. The economical efficiency change is estimated based on the boring cost of $100/m in Japan. For the U.S., this value is estimated to be about $30/m.

2.8. Cogeneration systems

Cogeneration, or combined heat and power generation, is the simultaneous production of electrical or mechanical energy (power) and useful thermal energy from a single energy source (ASHRAE 2008). By capturing and using the recovered heat energy from a discharged stream that would otherwise be rejected to the environment, cogeneration systems can operate at utilization efficiencies greater than those achieved when heat and power are produced in separate processes, thus contributing to sustainable building solutions. Cogeneration systems can recover thermal energy exhausted from fuel used in reciprocating engines, steam or combustion turbines, Stirling engines, or fuel cells. In the building sector, the main applications of cogeneration systems are in education and health-care buildings with significant potential in other applications (Wagner et al. 2009).

The case study of a cooling heat and power cogeneration system for a hotel in San Francisco, CA is described in Morrow and Savarino (2006) and Wagner et al. (2009). The proposed packaged system integrated four microturbines, a double-effect absorption chiller, two fuel gas boosters and the control hardware. The chiller was energized directly from hot exhaust of microturbines and could provide either ChW or heating hot water (HHW). The system provided 277 kW of net electrical power and 142 RT of ChW at a 59°F (15°C) ambient temperature. The system demonstrated a maximum efficiency over 80% and annual efficiency of 54% (reduced by non-optimal integration) compared with overall traditional system efficiency of about 30% (Petrov et al. 2006). Annual system efficiency was estimated from total energy delivered by the system and the fuel energy consumed to operate it. According to the measured data (Wagner et al. 2009), the overall system efficiency of cogeneration systems with thermally activated equipment (absorption chiller) based on the Brayton power cycle, increase with rising ambient temperature in opposite to the simple Brayton cycle. The cogeneration system in this case study was integrated in a ChW loop. The operation of the absorption chiller was in parallel with two existing 300 RT electric chillers with flow balancing by a motorized isolation valve. Flow through the absorption chiller was set up at 17 L/s with the bypass loop flow varying according to the building ChW demand.

To prevent electrical energy generated on site from exporting to the electrical grid, the buffer value between the generator power and normal load was used. The system was operating with load above 60 kW of net electrical power during 94% time of the year and at least 10 RT during 92% time of the year. The cogeneration system reduced energy cost in 2006 by $73,560/year with an installation cost of $1,040,000. Installation price for similar projects in the future can potentially drop by 25%. An economic analysis shows a 4.2 year payback without incentives (Wagner et al. 2009).

Theoretical exergy analysis of a solid oxide fuel cell based cogeneration system with a fuel utilization efficiency of 68 % and exergetic efficiency 62 % was described by Colpan et al. (2008). These efficiency values were obtained for a steady state operation, an ideal gas application and neglecting other losses. According to the publication, the most suitable fuel cell technologies for cogeneration systems are solid oxide fuel cell operating at 932-1832°F (500-1000°C) and molten carbon fuel cells with a 1112-1292°F (600-700°C) operation range. The main advantages of solid oxide fuel cell are a simpler concept because of only two phases existing with no electrolyte or electrocatalysts combined with possible utilization of carbon monoxide as a fuel. This research proposes electricity generation in a topping cycle by a gas turbine with a sequential utilization of thermal energy in solid oxide fuel cell and remaining heat energy utilized by a steam generation.

2.9. Thermal storage systems

A thermal energy storage system removes heat from or adds heat to a storage medium for use at another time (ASHRAE 2008). This separation allows thermal storage systems to generate heating or cooling during the most favorable conditions. Thermal energy storage for HVAC applications can involve various temperatures associated with heating or cooling. High-temperature storage is used for a solar energy system application or high-temperature heating and cool storage for air-conditioning applications. Energy from the storage can be charged, stored, and discharged at any time in seasonal or rapid cycles.

Thermal energy can be stored in a sensible storage (liquids, solids), or a latent storage (phase-change materials). Most common sensible storage medium are water, oil or aqueous glycol (ASHRAE 2008). For HVAC systems, the building concrete core can also be used as a sensible cooling storage system that utilizes materials thermal mass to reduce HVAC power requirements (Olesen et al. 2006). Another important application of thermal storage is solar energy system. Usually during design and selection of solar energy systems, thermal energy storage is one of the most neglected elements even though it has an enormous influence on overall system cost, performance, and reliability (ASHRAE 2008).

Latent storage utilizes phase-change material — a substance that undergoes a change of a state, normally from solid to liquid or liquid to solid, while absorbing or rejecting thermal energy at a constant temperature (ASHRAE 2008). The main advantage of a phase-change material is isothermal behavior during charging and discharging. In addition, the higher energy density of a phase-change material decreases the size and associated energy loss of a thermal storage.

Schweigler et al. (2007) describes a system concept of a large capacity solar thermal installation for a space heating and sorption cooling in radiative heating and cooling facilities with energy transfer at moderate heating and cooling temperatures. This system also includes low-temperature heat storage based on phase-change material, which eliminates requirements for a wet cooling tower. Wet cooling towers could potentially be substituted by a combination of a latent heat storage system, a water/lithium bromide solution absorption chiller and dry cooling system while maintain the same efficiency level. This is possible by shifting the heat rejection of the chiller to night periods with lower ambient temperatures. Building thermal storage systems could be used in hot continental climates with low humidity levels, where a big difference between day and night temperatures exists.

Proposed by Schweigler et al. (2007), a 10 kW solar driven single stage absorption chiller requires 15 kW driving heat input and thus releases 25kW of rejected heat that traditionally require a wet cooling tower with a water temperature 95ºF/80.6 F (35/27 C). This could be replaced in moderate ambient air temperature climates with dry air coolers based on glycol-brine coolant cycle. This cycle is cooling glycol-brine from 104ºF (40ºC) to 96.8ºF (36ºC) at 89.6ºF (32ºC) ambient temperature. Then, glycol-brine latent heat storage cools the brine from 89.6ºF (32ºC) by internal phase-change at a temperature around 84.2ºF (29ºC).

A dry air cooler system with a water/LiBr-based absorption chiller does not produce ChW at 53.5ºF/42.8ºF (12ºC/6ºC) that is required by traditional types of HVAC systems, but it could produce ChW at 64.4ºF/59ºF (18ºC/15ºC), which is applicable for radiative cooling applications like ceiling cooling. According to the paper by Schweigler et al. (2007), integration of flat plate collector for solar energy generation and latent heat storage into the coolant loop allows to produce ChW at 53.5ºF/42.8ºF (12ºC/6ºC) with 49% efficiency. The application of a dry air cooler in the system increases the solar field by 40% without latent heat storage and just 20% with latent heat storage.

Another publication by Cabeza et al. (2006) from University of Lleida describes an application of granular phase-change graphite compound of about 90% sodium acetate trihydrate and 10% graphite with a melting point of 135.4ºF (58ºC). The simulation of the designed system was made using TRNSYS TYPE-60 component and shows very good concordance. The thermal energy storage system utilized thermal stratification and phase-change modules with different melting temperatures (¼ of volume) submerged in a water storage tank.

2.10. Case study of high energy efficient building technologies

A majority of publications describes high energy efficient buildings located in temperate climates (Iowa, Minnesota, Ohio, Wisconsin, or Beijing in China) and only a few of them describe case study in hot and humid climates.

A case study of a low energy office building in Dallas, Texas by Reilly et al. (2007) describes a building with an annual energy consumption of 76 $\mathrm{kBtu/ft^2 yr}$ ($240\,\mathrm{kWh/m^2 yr}$) and a peak electricity demand $3.6\,\mathrm{W/ft^2}$ ($39\,\mathrm{W/m^2}$) that was achieved with relatively low construction budget of $\$140/\mathrm{ft}^2$ ($\$1500/\mathrm{m}^2$). The simple payback of building green measures was nine years. A high energy efficiency level was achieved in a combination of building energy efficient construction materials, energy consumption control systems (continuous dimming daylighting controls in all perimeter spaces, occupancy sensors) and enthalpy wheels between the exhaust air and incoming fresh air. The lighting power consumption was reduced due to the use of T8 fluorescent lamps and electronic dimmable ballasts, occupancy and daylighting dimming controls. The lighting control was accomplished without integration to VAV boxes control. The building has VAV HVAC system with electrical reheat. Enthalpy recovery wheels operated with an efficiency between 0.73 and 0.78 and static pressure drop 132 Pa (0.53 inH_20) on the exhaust and 174 Pa (0.7 inH_20) on the supply that helped to reduce chiller capacity from 1200 to 1000 RT.

A building HVAC system designed for a part of the office building with an area of 2500 m^2 located in the hot and humid climate of Singapore is described by Sekhar (2007). This system had been installed at the Department of Buildings of the National University of Singapore. The building HVAC single coil, twin-fan system utilizes two variable volume systems with one cooling and one dehumidifying coil. In this system, fresh air is conditioned in the fresh air section of the AHU and is distributed to the VAV boxes controlled by local CO_2 sensors installed in ducts near the return air grill. The return air is conditioned at the recirculation section of AHU and is supplied by a separate set of VAV boxes. Each of them is controlled by their own local thermostat according to non-occupancy related factors. Temperature and CO_2 sensors were installed in every individual office or conference room. This system has an energy saving potential up to 12% (Sekhar et al. 2004) when compared with conventional systems.

Another high-performance office building in Iowa is described in a case study by McDougall et al. (2006). The building was constructed with a modest budget of $\$166/\mathrm{ft}^2$ ($\$1248/\mathrm{m}^2$). The main applied concepts include daylighting with dimming and occupancy sensors, geothermal heating and cooling system and an energy recovery enthalpy wheel. The building model was based on DOE-2.1E simulation. The geothermal constant-speed, single-stage heating and cooling system in this project offered smaller lifecycle cost compared with VAV systems, rooftop units. The building geothermal pumps consumed 36% of the total building HVAC energy.

A nine-story energy-efficient demonstration office building near Yuyuantan Park in Beijing, China is described in a study by Xu et al. (2007). The main energy-efficient technologies that were chosen for the building are: high-efficient lighting, bi-level light switches as a simplified daylighting strategy, staged chillers with an improved efficiency (COP 4.4), and an economizer. Because of the weather conditions and internal loads, the fan-coils were dry during most of the operation time. The building HVAC system has a two-pipe instead of a four-pipe fan coil without simultaneous heating and cooling mode. This type of a system is popular in China. The building uses a DOAS with a roof-mounted air-to-air heat exchanger. The efficiency of the air-to-air heat exchanger was around 0.78. The roof also had installed PV panels with a total area of 1,170 ft^2 (109 m^2) and a 15 kW capacity, together with a flat-plate solar hot-water system 215 ft^2 (20 m^2). The building lighting system includes automated occupancy controls, dimming ballasts and light shelves. Lighting power density was less than 0.37 $\mathrm{W/ft}^2$ (4 $\mathrm{W/m}^2$). The building heating system was connected to the city district heating system.

A case study of the energy-efficient academic building in Oberlin, Ohio is described in a publication by Pless et al. (2006). Main design features of this building are: daylighting, enhanced thermal envelope, ground-source heat pump, energy recovery ventilation, dimming lighting system, and natural ventilation.

The building is equipped with 60 kW PV arrays connected to a utility grid. During the third year of occupancy, the building consumed 29.8 $\mathrm{kBtu/ft^2yr}$ (94.0 $\mathrm{kWh/m^2yr}$), which is 48% less energy than a conventional ANSI/ASHRAE Standard 90.1-2001 energy code-compliant building. The photovoltaic system generated 45% of the required energy. All systems of the building were designed to consume only electrical energy with a purpose of utilization just energy from PV array. Building low-e windows have triple- and double-panes, argon filled with thermally broken aluminum frames. The exterior walls had R-19insulation, the roof R-30, and the slab R-10. The building lighting system is equipped with motion control and dimming sensors. The installed lighting power density is 1.26 $\mathrm{W/ft^2}$ in classrooms, 0.88 $\mathrm{W/ft^2}$ in offices, and 0.45 $\mathrm{W/ft^2}$ in corridors.

The photovoltaic system consists of 690 PV modules mounted on the roof. Each of them has a nominal power of 85W. The PV system is wired in three subarrays, each consisting of 10 strings wired in parallel from north to south. Each subarray is connected to a 15kW grid-tied inverter. Each inverter is connected to the building electrical panel through three insulation transformers. The PV system installation cost was $386,000, and it will not be recovered during lifetime of the system.

The building geothermal system includes water-to-air room heat pumps connected to a closed loop, glycol/water circuit. Variable speed drives circulate the glycol/water mix through twenty-four 240 ft (73 m) deep wells. A single large water-source heat pump, coupled to an energy recovery ventilator, handles 100 % of the outdoor ventilation air. The outdoor air system control is based on occupancy sensor data in each room and open supply dampers when the space is occupied. Installed CO_2 sensors are not used to control ventilation. For building heating was used a hydronic loop water-to-water ground source heat pump with an electric boiler backup. According to Pless et al. (2006), the enthalpy wheel and related fans should be in operation only in the heating mode when the OA temperature is lower than 65 F (18 C), because energy consumed to operate this system is greater than the recovered energy at outdoor temperatures above 65 F.

A paper by Bradley and Utzinger (2007) describes the design of a carbon neutral building for the Aldo Leopold Foundation in central Wisconsin. The project utilizes carbon dioxide level controls with earth duct and energy recovery ventilators. The mechanical ventilation is designed to operate during 80 % of the occupied hours. The displacement ventilation system was designed to provide only OA flow (0.07-$0.21\,\text{cfm/ft}^2$) with no recirculation and supply air temperature 65.8ºF (18.8ºC) and a fan power 1.05 W/cfm. The building non-ventilation loads are met using radiant slabs through which either hot or cold water is pumped from a tank that supplies energy from a staged bank of three geothermal heat pumps. For summer, ChW from the tank is used for the radiant floors and the cooling/dehumidifying coil in the air handler. Partial dehumidification of the OA is realized by the earth duct. The reheat system is using energy from 35kW PV system inverters. The most expensive components of this system were the earth duct and radiant floor system.

A net zero energy building designed for the purpose of demonstrating state-of-the-art energy efficiency and renewable energy features is described by Steinbocket al. (2007). This building is a Science House at the Museum of Minnesota in Minneapolis, Minnesota. The building was equipped with occupancy sensors that switch off the light during periods of no activity, and daylight sensors that dim the electric lights when daylight is available to maintain constant light level. The electric lighting system energy consumption was designed to be $1\,\text{W/ft}^2$ ($10.8\ \text{W/m}^2$) which is significantly lower than Minnesota Energy Code based on ASHRAE 90.1-1989. All of the electrical energy, $22{,}385\ \text{Btu/ft}^2\text{yr}$ ($0.71\ \text{kWh/m}^2\text{yr}$), was supplied by a PV system, which generated $30{,}864\ \text{Btu/ft}^2\text{yr}$ ($0.97\,\text{kWh/m}^2\text{yr}$).

Photovoltaic panels with an efficiency of 6%-8% were installed on the roof and connected to four inverters. The installed PV array with an area of $1456\ \text{ft}^2$ ($135.3\,\text{m}^2$) cost \$62,685. With the current commercial rate in Minnesota, the payback is approximately 72 years, which is much longer than the system lifetime.

An experimental energy efficient HVAC system for a hot climate is described by Ameen and Mahmud (2005). Building cooling in this case study is provided by a dropdown chilled ceiling operated in combination with desiccant dehumidifier. Both systems are supplied by a 3.3 RT (11.72 kW, 39,988 Btu/hr) air cooled chiller. The temperature of the ceiling panels was controlled by a three-way bypass valve that also supplied bypass water to the precooler.

The chilled ceiling system combined 12 flat panels made of aluminum plates of 0.029 in (1 mm) thickness covering 70 % of the total ceiling area. The copper cooling tubes used in the system have 0.468 in (12 mm) diameter and 5.85 in (50 mm) spacing between them. Outside air was supplied by a desiccant-based air conditioner. According to Ameen and Mahmud (2005), desiccant dehumidification is the most appropriate system in combination with chilled ceiling. The dehumidification system used a commercial silica gel desiccant wheel which operates by continuous physical adsorption with outside heated air reactivation. During the simulation process, the experimental room with sides 13.94 ft x 12.3 ft and height 9.84 ft (4.25 m x 3.75 m x 3 m) had inside air conditions, of 77ºF (25ºC) and 50 % RH, while the outdoor design conditions were 93.2ºF (34ºC) db and 82.4ºF (28ºC) wb. The ceiling temperature was maintained within a range 59-64.4ºF (15-18ºC) by thermostat controlling three-way bypass valve in the ChW line. Chilled water was also used to cool air in the heat exchanger downstream of the desiccant wheel after heat exchange with the exhaust air. Research data confirmed that even at a chilled ceiling panel temperature of 59ºF (15ºC) a condensation or sweating problem did not arise and a comfortable environment was maintained. Conclusions from the experiment show that the hybrid system that combines chilled ceiling and desiccant dehumidification is more energy efficient compared with a conventional system when the required OA ventilation rate is above 30 % (Ameen and Mahmud 2005).

2.11. Summary of literature review

This literature review focuses on innovative HVAC systems for commercial buildings in hot and humid climates. Publications that were reviewed reveal a large number of alternative design and control strategies for HVAC systems that could be used, and an almost endless number of their combinations. Nevertheless, the number of publications that compare different HVAC concepts is limited. Since the systems described are not commonly used, it is not always possible to provide a cost evaluation of the energy consumption reduction compared to the investment required. The findings of the literature review are summarized below:

A decoupled system concept splits HVAC systems into two components: a dedicated outdoor air system and an air conditioning system that provides sensible cooling and heating (DV system as an example). The decoupled system concept could provide better humidity control and reduced energy consumption by eliminating supply air reheat requirements and excessively low cooling coil temperatures. This provides a high level of humidity control that is usually highly influenced by OA humidity in hot and humid climates. Decoupled HVAC systems presently do not have wide applications and consequently an economic cost-benefit evaluation is difficult.

The application of personal ventilation systems is limited by the complicity of the system and flexibility of its application.

A displacement ventilation system can provide better air conditioning efficiency, but it requires a minimum stratification height limit of 4 ft and significant investment in construction. A radiant cooled ceiling has an operating cost payback in the range of 50 years and additional aesthetic disadvantages. The application of chilled beams in DV could potentially provide an operating cost payback in less than two years.

Air-to-air heat recovery systems can be efficient if the exhaust air flow rate is higher than 75 % of the supply OA rate. This requirement is usually not applicable to the office conditions where OA requirements are significantly lower. This makes the potential benefit of an air-to-air heat recovery component small in cases where the OA supply flow is controlled based on a demand level.

Dedicated heat recovery systems have low cost and could be used with a high efficiency for domestic water heating in office buildings. Maximum efficiency could be obtained when the dedicated heat recovery system is sized by the side of the system that has the lowest demand.

Average office building plug load in the United States stays constant around $0.83\ \text{W/ft}^2$ with improvement in energy efficiency balanced by an increase in performance of office equipment. Most efficient lighting systems can reduce lighting power density to $0.90\ \text{W/ft}^2$ $(10\,\text{W/m}^2)$ or less in standard office applications by using T8 luminescent lamps with electronic instant start ballast.

The most available renewable energy source – solar energy could be used with high efficiency for domestic water systems. Solar energy cooling systems have a major disadvantage with their large initial investment in the solar collector and the absorption chiller which prevents these systems from widespread applications. In the case where electrical power supply from the grid is available, photovoltaic systems are not economically feasible at the present state of technology without significant government support. All case studies that were reviewed and utilized photovoltaic system did not include economical efficiency as a factor of optimization during the system design.

Utilization of a heat pump that operates on natural gas can cool for less money than electrical chillers when natural gas is available for a relatively lower price than electricity. An additional advantage of this technology is a reduction of electrical system peak demand.

Ground heat pumps used in hot climates should utilize an additional component such as a cooling tower for heat rejection during the cooling mode operation, which balances the ground storage load. In the weather conditions of southern Texas, a hybrid geothermal heat pump system appears to be beneficial on the basis of both first cost and annual operating cost (Yavuzturk and Spitler. 2000). The economic efficiency of the system strongly depends on local ground properties and the price of GHX construction.

Use of cogeneration systems is limited to applications that require thermal energy. The traditional application utilizes a cogeneration system for heating needs, but they also could be used with absorption chillers for building cooling.

Thermal storage systems could be used in hot continental climates with low humidity levels where big difference exists between day and night temperatures.

According to case studies the most common strategies for energy efficient office buildings were: energy efficient construction materials, DV systems, DOAS, energy consumption control systems (continuous dimming daylighting and occupancy controls), air-to-air energy recovery by enthalpy wheels and economizer mode utilization, CO_2 sensors control for OA supply, geothermal heating and cooling system, PV systems.

CHAPTER III

WISENBAKER ENGINEERING RESEARCH CENTER SIMULATION

This chapter describes modeling and evaluating of energy saving opportunities in an existing office building. Wisenbaker Engineering Research Center, which is where the Energy Systems Laboratory' (ESL) office is currently located, was chosen for the energy efficiency analysis. The provided analysis is limited by the amount of data that was available for simulation, but the obtained results are valid for the detection of energy savings opportunities. For the calibration process of the simulated building, the characteristic calibration signature method was used.

3.1. Description of the whole-building energy simulation software

In the past half century, more than 380 different programs and tools were developed to evaluate energy efficiency, renewable energy, and sustainability in buildings (U.S. DOE 2010a). Some of these programs were dedicated to certain applications (load calculations, renewable energy, retrofit analysis, environmental performance or energy code compliance) while others are capable of whole-building energy analysis. An extensive building simulation history review provided by Malhotra (2009) reveals the history of simulation program development. At the present moment, the most widely used whole-building energy simulation programs are: DOE-2.1E, TRNSYS, and EnergyPlus.

Programs like BLAST, DOE-2.1E utilize an hourly fixed schematic principle based on steady state methods like Degree-Days and Bin Analysis. TRNSYS and EnergyPlus were developed later and are based on a modular structure with the possibility to use less than an hour simulation time steps (U.S. DOE 2010c).

The Building Loads Analysis and System Thermodynamics (BLAST) program was released in 1977. It was developed by U.S. Army Construction Engineering Research Laboratory's (CERL) based on an earlier computer simulation programs by National Bureau of Standards Load Determination (NBSLD). The NBSLD program combines algorithms for transient conduction in the building structure, solar heat gains, radiant transfer, and convection between building surfaces and the room air in order to predict interior temperatures and heating/cooling loads of a single zone building under dynamic conditions.

DOE-2 was released later by the Lawrence Berkeley Laboratory (LBL) in 1978 based on a building energy analysis program for U.S. Postal Service and was sponsored by the Energy Research and Development Administration (ERDA) (now known as the Department of Energy, (DOE)). DOE-2 uses one subprogram for the translation of inputs and four simulation subprograms (LOADS, SYSTEMS, PLANT and ECONOMICS) executed in sequence.

TRNSYS (TRaNsient SYstem Simulation program) was developed by the Solar Energy Laboratory at the University of Wisconsin, primarily as a program for simulating solar thermal systems (Klein 1973), which later incorporated general HVAC system simulations. TRNSYS subroutines representing the physical components are combined and solved simultaneously with the building envelope thermal balance and the air network at each time step. The TRNSYS library includes components for multi-zone building models, low energy buildings, HVAC systems, and renewable energy systems, including passive solar, active solar thermal and PV systems, wind energy, fuel cells and cogeneration, etc. Furthermore, the modular nature of TRNSYS facilitates the addition of new mathematical models to the program (Malhotra 2009).

EnergyPlus was developed on a modular principle with a combination of algorithms of BLAST and DOE-2.1E. Similar to BLAST and DOE-2.1E, EnergyPlus also uses the response factor method for transient heat transfer through multilayered walls. It also allows users to evaluate realistic system controls, moisture adsorption and desorption in the building elements, radiant heating and cooling systems, and interzone air flow, PV systems and fuel cells (Malhotra 2009).

Taking into account the thesis purpose DOE-2.1E was chosen for the building energy simulation. The main advantage of this program is an opportunity of detailed, hourly, whole-building energy analysis of multiple zones in buildings of complex design

(U.S. DOE 2010b). DOE-2.1E is widely recognized as the industry standard (U.S. DOE 2010b). This program has been extensively validated for accuracy and consistency and is usually the program against which other programs are compared (Haberl and Cho 2004). Another strong advantage of this program is the public domain and the opportunity to trace back to the source input commands that were used.

3.2. Weather Data: Typical Meteorological Year (TMY3)

The analysis of WERC energy efficiency (with DOE-2.1E) was performed with Typical Meteorological Year version 3 (TMY3) weather data for College Station Easterwood Airport, TX (U.S. DOE 2010d). A TMY3 weather file for current location is comprised of hourly measured data of the months with the most representative temperature and solar radiation characteristics, which are linked to form a complete, contiguous year. TMY3 hourly datasets were available in the format required for the analysis with DOE-2.1E.

3.3. Wisenbaker Engineering Research Center building description

One of the aims of this research is to document energy saving potential for a real office building. For this purposes WERC was chosen as a research object. Wisenbaker Engineering Research Center is located on the east side of the main campus of Texas A&M University, College Station, Texas (Figure 3.1).

The building was built in 1983. The net area of the building is 174,016 ft^2 with three floors and a basement. The building construction is of high mass with concrete floors and basement walls of poured concrete. Wisenbaker Engineering Research Center has three stories above the ground facing 45° north of east (Table 3.1).

Originally the basement was designed as an underground parking lot, but later in the 1990's it was redesigned for offices and laboratories serving different departments. Today, this multipurpose building houses the offices of the ESL as well as other different research organizations. In addition to office areas and conference rooms, it also has a number of different laboratories including a large material testing laboratory

(MTL), which is a separate structure constructed of sheet metal and served by the WERC HVAC system.

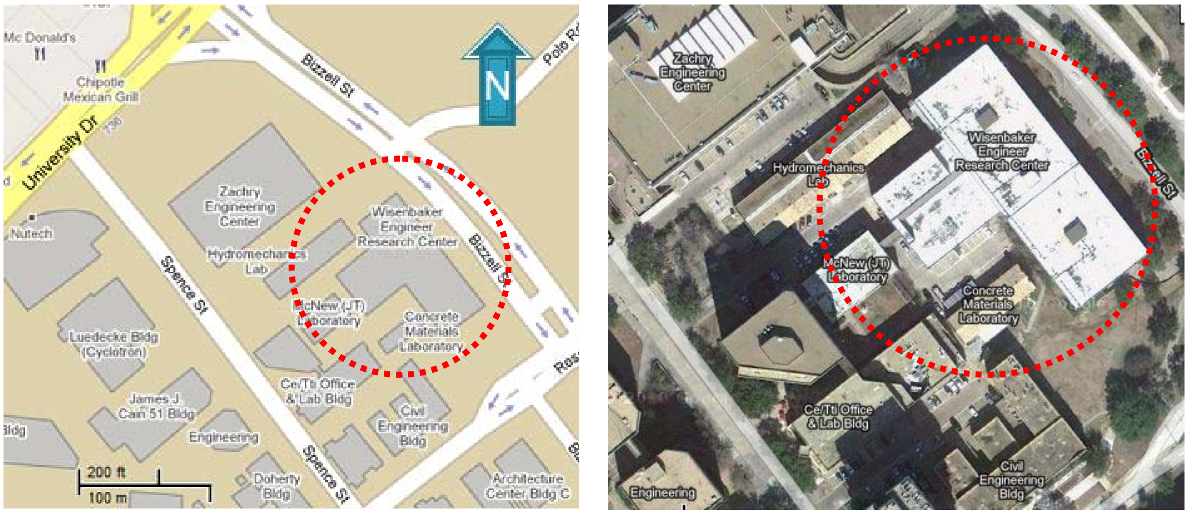


Figure 3.1. Wisenbaker Engineering Research Center geographical location (Google Maps 2010)

Table 3.1. Location of Wisenbaker Engineering Research Center

DOE-2 Keywords	DOE-values	Comments
Latitude	30.61	Precise value (30° 37’ 13.32”N)
Longitude	96.3	Precise value (96° 20’ 24.57”W)
Altitude	325	
Azimuth	-135	45° north of east
Time zone	6	Central Time Zone
Daylight savings	Yes	Daylight saving time
Holiday	Yes	The U.S. Holiday
Ground temperature	No	Auto calculated by DOE-2

The single duct HVAC system of the building consists of seven variable volume air handlers serving the first, second, and third floors, and four constant volume air handlers serving the basement and MTL. Three air handler units that supply the basement are completely OA systems. In addition, forty fan-coil units are also used for conditioning the basement. The HVAC for all floors, except the basement, connected to a Siemens® APOGEE energy management control system.

The architectural and mechanical drawings of the building were obtained from the facilities office of the Texas A&M campus. However the drawings for the basement have not been updated in that they do not include the physical changes related to both structures and systems, which have been incorporated into the buildings after the initial construction.

To obtain the missing information, detailed walkthroughs were conducted. During the walkthroughs zone operation schedules were obtained, and the control settings and performances of all AHU system where checked. The walk-through provided detailed information about the envelope and the architectural layout of the building. The information obtained showed that the AHU with the variable frequency drives (VFD) operates at almost a constant speed mode, since all terminal boxes had some problems with the flow control related to stuck valves and faulty pneumatic controls.

3.4. Data acquisition and design information

For building simulation, the hourly whole-building electricity, ChW and HHW consumption data from 1/1/2008 to 8/1/2009 was obtained. For this period the electrical energy base demand was around 300 kW (1.7 W/ft^2), with maximum average energy consumption increases to 450kW during the week day (Figure 3.2, Figure 3.3).

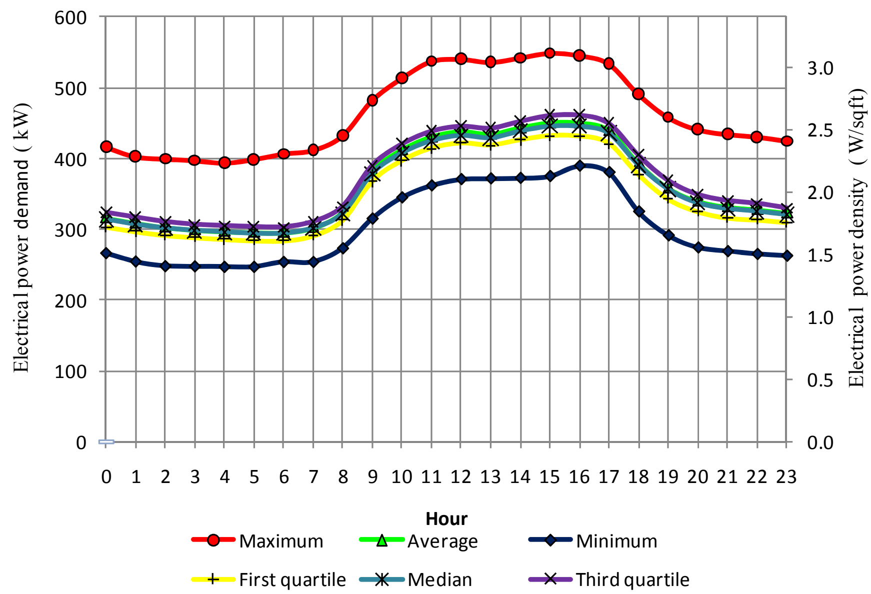


Figure 3.2. Wisenbaker Engineering Research Center whole building electrical energy consumption profile during weekdays (1/1/2008 to 8/1/2009)

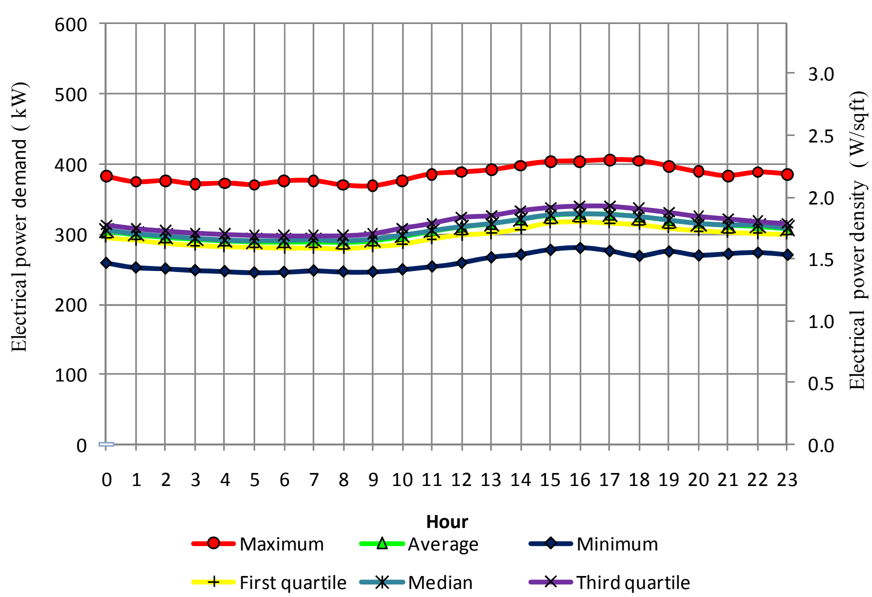


Figure 3.3. Wisenbaker Engineering Research Center whole building electrical energy consumption profile during weekends (1/1/2008 to 8/1/2009)

During the weekends and holidays, electrical energy consumption was relatively stable around 300 kW (Figure 3.4). The building's HVAC system consumes ChW and HHW that is supplied from the central utility plant. The chilled water energy consumption varies from 20 MMBtu/day to 100 MMBtu/day and HHW from 14 MMBtu/day to 65 MMBtu/day (Figure 3.5).

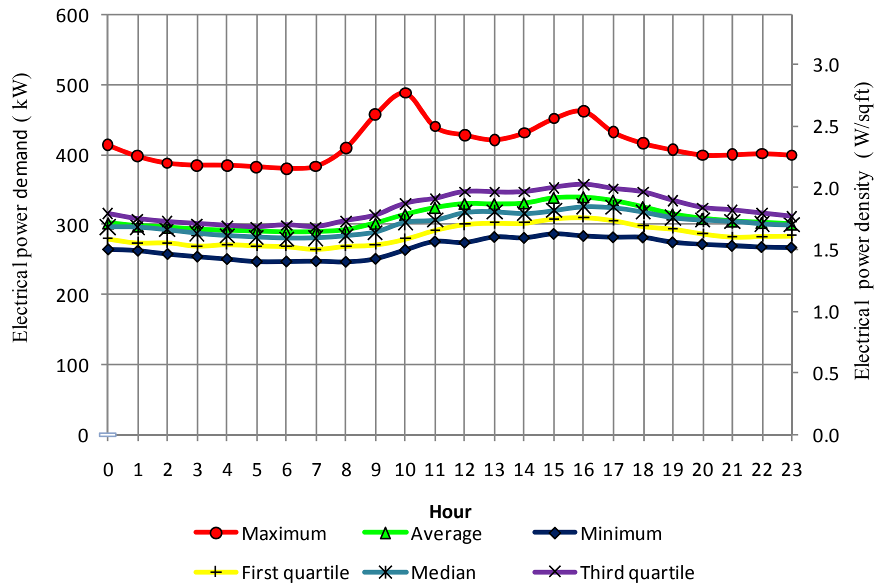


Figure 3.4. Wisenbaker Engineering Research Center whole building electrical energy consumption profile during holidays (1/1/2008 to 8/1/2009)

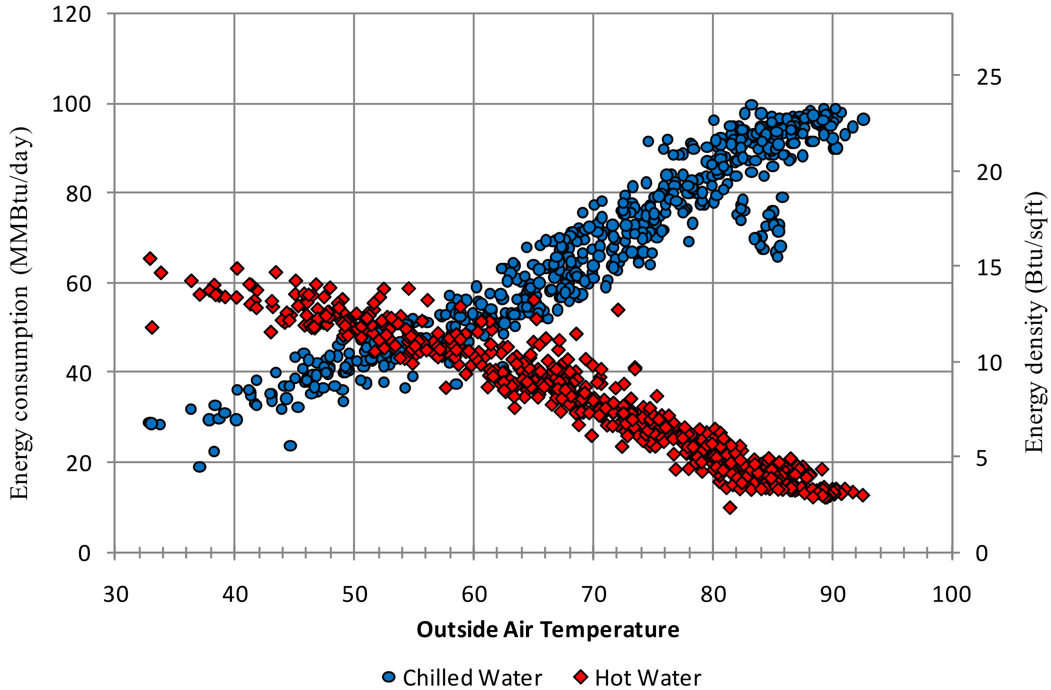


Figure 3.5. Wisenbaker Engineering Research Center chilled and heating hot water energy consumption (1/1/2008 - 8/1/2009)

To estimate the internal energy gain in the WERC office area, measurements of electrical consumption by lighting and equipment load in the area occupied by ESL offices in the basement, first floor and second floor were performed during February and March 2010 (Figure 3.6-3.8).

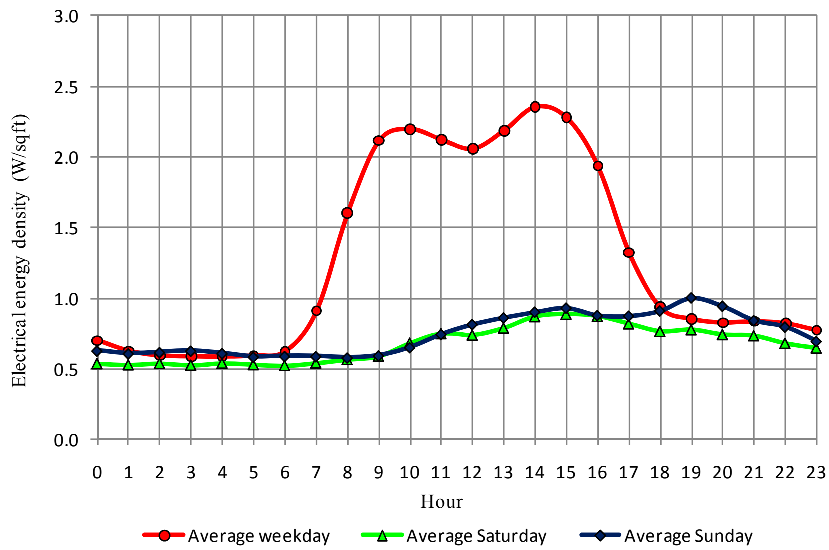


Figure 3.6. Average electricity consumption of lighting and office equipment in the ESL basement office area (2000 ft^2) during 2/18/2010 - 3/26/2010

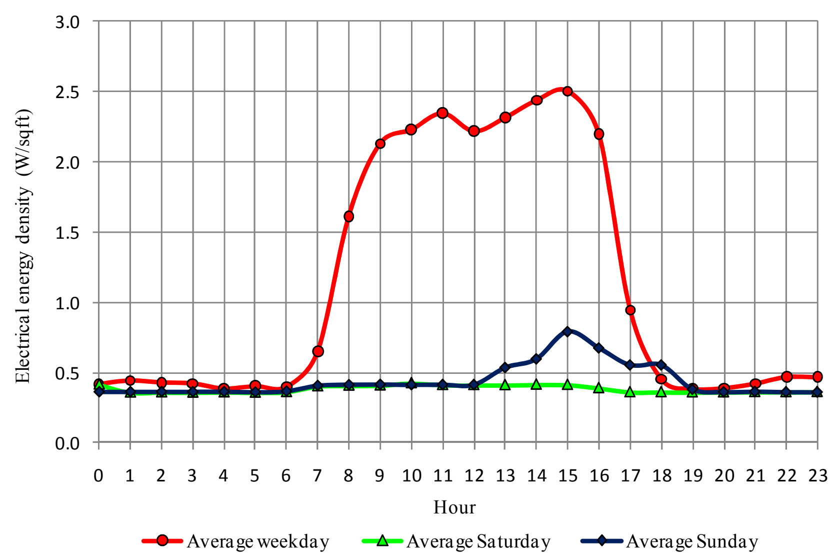


Figure 3.7. Average electricity consumption of lighting and office equipment in the ESL first floor office area (540 ft^2) during 2/18/2010-3/25/2010

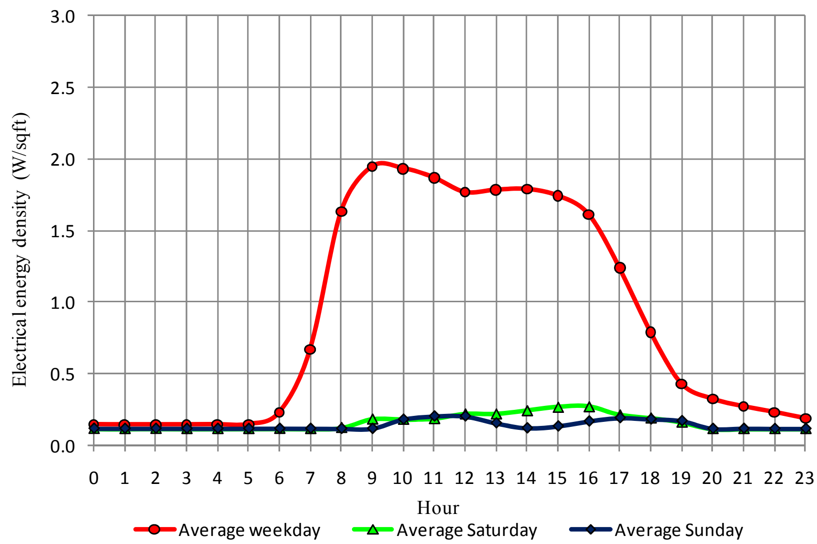


Figure 3.8. Average electricity consumption of lighting and office equipment in the second floor ESL office area (4400 ft^2) during 2/2/2010 - 3/26/2010

Office energy consumption plots in all three zones have common characteristics: the ratio of maximum-to-base load is below 25% for week days, office consumption during weekend is close to the unoccupied night time energy consumption.

The overall electricity consumption of ESL office space, with a total area 6,940 ft^2, has a significantly lower ratio of base load (13%) to an average daily maximum value compared with the whole building electricity consumption profile (70%) (Figure 3.9). This difference could be explained by the influence on the whole building electrical energy consumption profile by research laboratory equipment.

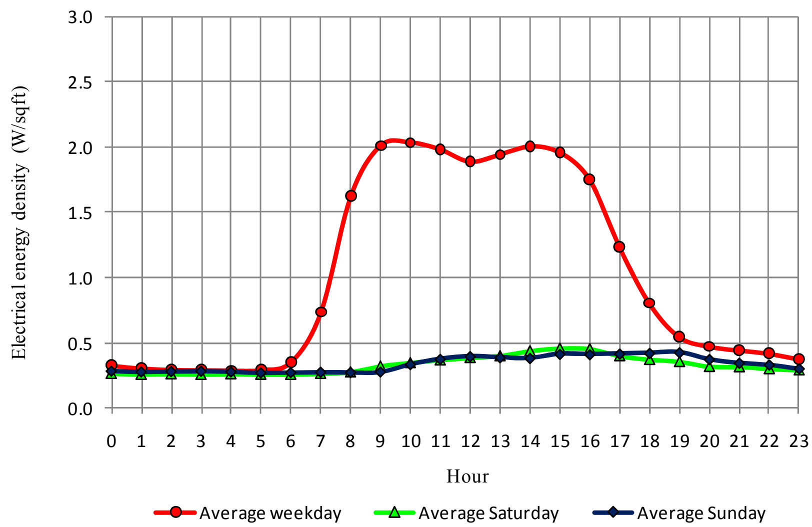


Figure 3.9. Wisenbaker Engineering Research Center lighting and office equipment average electricity consumption in the ESL office area (6,940 ft^2)

The second floor office area electrical energy consumption was measured separately for lighting and office equipment (Figure 3.10).

Measurement results agree with the previously published average value for office equipment consumption 0.83 W/ft^2 (Komor 1997), even though the previous publication is more than 13 years old. The value of the lighting power density, 1.2 W/ft^2, is higher than 1.0 W/ft^2, which is the lighting power density recommended by ASHRAE Standard 90.1-2007 (ASHRAE 2007).

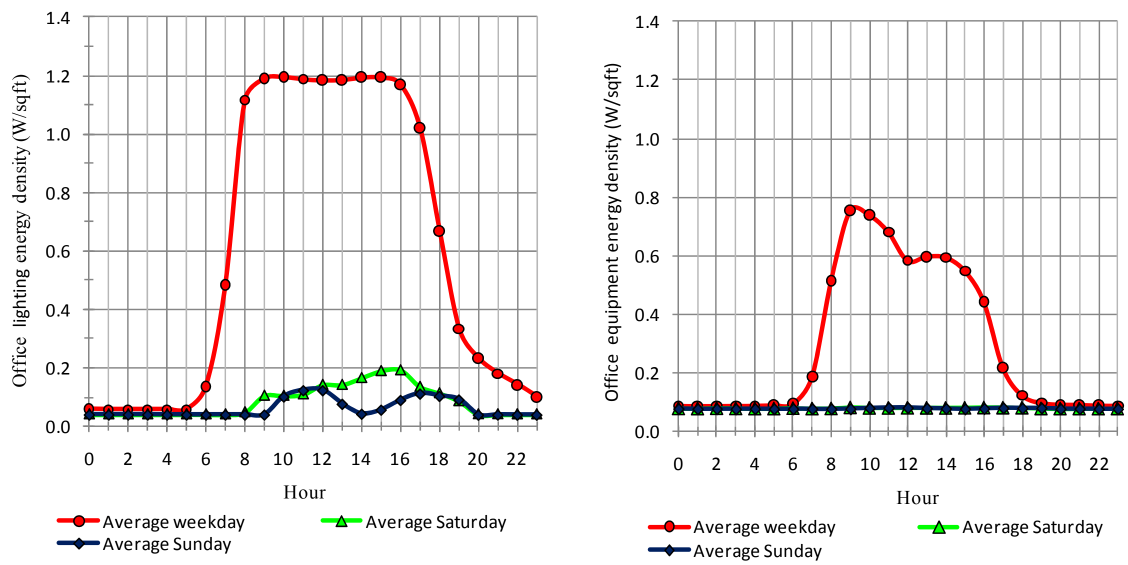


Figure 3.10. Average lighting and office equipment energy consumption profile in the ESL second floor office area (2/2/2010 - 3/26/2010)

A walkthrough of AHUs show that all AHU supply fans with VFD control were operating with constant speed since the VAV terminal boxes control were not functioning (Table 3.2, Table 3.3).

Table 3.2. Wisenbaker Engineering Research Center AHU supply fan parameters

AHU	VFD	System type	Supply fan control	Supply fan mode	Nominal Design H.P/U	Total air, cfm	OA, cfm	Name Plate H.P.	Recorded value		
									f, Hz	Power H.P.	Current, A
1	1	SDVAV	VFD	Hand	30/480V	27500	5500	25	40.3	7.98	15.5
2	2	SDVAV	VFD	Automatic	30/480V	27500	5500	25	51.7	16.7	23.77
3	3	SDVAV	VFD	Automatic	20/480V	18500	1500	20	43.8	7.32	13.9
4	4	SDVAV	VFD	Automatic	20/480V	18500	1500	20	51.1	11.8	17.74
5	5	SDVAV	VFD	Hand	25/480V	21500	1500	25	40.3	7.78	16.27
6	6	SDVAV	VFD	Hand	25/480V	21500	1500	25	40.1	7.43	15.65
7	7	SDVAV	VFD	Automatic	10/480V	12000	2000	15	60	6.3	9.17
8		SDVAV	Constant volume	Automatic	25/460V	20000	1000	20	60	25	

Table 3.2. (Continued)

AHU	VFD	System type	Supply fan control	Supply fan mode	Nominal Design H.P/U	Total air, cfm	OA, cfm	Name Plate H.P.	Recorded value		
									f, Hz	Power H.P.	Current, A
1B	NA	DOAS +FCU*	APOGEE ON/OFF, night reset	Constant	1/460V			1	60	1	
2B	NA	DOAS +FCU	APOGEE ON/OFF, night reset	Constant	3/460V			3	60	3	
3B	NA	DOAS +FCU	APOGEE ON/OFF, night reset	Constant	2/460V			2	60	2	
					191			181		96.31	

*FCU – Fan Coil Unit

Table 3.3. Wisenbaker Engineering Research Center AHU return fan parameters

Room #	VFD	AHU #	Return fan control	Operation Mode	Nominal return power, H.P.	Recorded value		
						f, Hz	H.P.	Current, A
124	8	1	VFD	Hand	15	40	6.14	12.37
149	9	2	VFD	Automatic	15	41.7	7.04	13.18
231	10	3	VFD	Hand	15	40.3	2.96	7.10
216	11	4	VFD	Automatic	10	31.6	0.9	4.53
317	12	5	VFD	Hand	10	40	2.16	6.30
330	13	6	VFD	Automatic	10	24.9	0.4	2.82
227	14	7	VFD	Automatic	5	47.3	1.42	3.19
227		8	None					
Total					80		21.02	

The total measured supply fans electrical energy consumption was 96.31 hp or 71.82 kW and for return fans 21.02 hp or 15.67 kW.

3.5. DOE-2.1E simulation

3.5.1. Wisenbaker Engineering Research Center building envelope and loads simulation

Initial building simulation in DOE-2.1E was developed based on the information obtained from construction drawings and measured energy consumption. Table 3.4 shows detailed information about the WERC construction materials simulated in this study. Each material was selected from the DOE-2.1 material library (LBNL 1993a) corresponding to the actual materials in as-built drawings, field observations and data described by Mushtaq (2003). The inside air film resistance was defined as 0.68 Hr·ft^2·°F/Btu for all vertical wall surfaces. For the roof with an upward heat flow, inside air fi1m resistance of 0.61 Hr·ft^2·°F/Btu was used (LASL 1980).

Table 3.4. Wisenbaker Engineering Research Center construction material properties

Items	U-NAME		Description	Thermal Properties				
	Construction	Material		Thickness	Conductivity	Density	Specific heat	Resistance
				ft	Btu-ft/Hr-ft^2-°F	Lb/ft^3	Btu/Lb-°F	Hr-ft^2-°F/Btu
Roof	ROOF-1	RG01	1/2 inch Roof Gravel or Slag	0.0417	0.8340	55.0	0.40	0.05
		IN72	1 inch Preformed Roof Insulation	0.0833	0.0300	16.0	0.20	2.78
		CB12	8 inch Heavy Weight Concrete Filled Concrete Block	0.6667	0.7575	140.0	0.20	0.88
		GP01	1/2 inch Gypsum or Plaster Board	0.0417	0.0926	50.0	0.20	0.45
Exterior wall main building	WALL-1	BK02	8 inch Common Brick	0.6667	0.4167	120.0	0.20	1.60
		IN01	Batt, R-7+ Mineral Wool/Fiber	0.1882	0.0250	0.60	0.20	7.53
		GP01	1/2 inch Gypsum or Plaster Board	0.0417	0.0926	50.0	0.20	0.45
Exterior wall MTL	LAB-1	AS01	Aluminum or Steel Siding	0.0050	26.000	480.0	0.10	0.00
		HB01	3/4 inch Medium Density Siding Hard Board	0.0625	0.0544	40.0	0.28	1.15
		IN21	7/8 inch, Preformed Mineral Board R-3	0.0729	0.0240	15.0	0.17	3.04
		HB01	3/4 inch Medium Density Siding Hard Board	0.0625	0.0544	40.0	0.28	1.15
		AS01	Aluminum or Steel Siding	0.0050	26.000	480.0	0.10	0
Underground constructions	WALL-U	GP05	1 inch Light Weight Aggregate Gypsum Plaster	0.0833	0.1330	45.0	0.2	0.63
		IN42	3/4 inch Expanded Polyurethane	0.0625	0.0133	1.5	0.38	4.67
		CB12	8 inch Heavy Weight Concrete Filled Concrete Block	0.6667	0.7575	140.0	0.2	0.88
Floor	FLOOR-1	AV01	Asbestos Vinyl Tile	-	-	-	0.3	0.05
		GP05	1 inch Light Weight Aggregate Gypsum Plaster	0.0833	0.1330	45.0	0.2	0.63
		CB12	8 inch Heavy Weight Concrete Filled Concrete Block	0.6667	0.7575	140.0	0.2	0.88
		GP05	1 inch Light Weight Aggregate Gypsum Plaster	0.0833	0.1330	45.0	0.2	0.63

Wisenbaker Engineering Research Center has three floors above the ground and a basement partially below the ground. The front façade of the building is depicted on Figure 3.11.

The building roof is assumed to have one inch of preformed roof insulation on an eight inch heavy weight concrete filled concrete slab. The exterior building walls are typically composed of eight inch common brick, R-7 mineral wool insulation and 1/2 inch gypsum board.

Material testing laboratory walls are assumed to be constructed from two layers of ¾ inch medium density siding hard board with 7/8 inch preformed mineral board R-3 insulation. The underground wall and floor construction are assumed to be 8 in heavy weight concrete filled concrete block with ¾ in expanded polyurethane insulation.

Figure 3.11. Wisenbaker Engineering Research Center front façade view

The building has windows with a single pane glazing and reflective coating. For simulation purposes, a window with glass type code 9 was chosen, since the manufacturer's glass properties were not available. The entrance doors are constructed from clear glass with metal frames. For door glass simulation, a glass type code 5 (single pane uncoated glass) (LASL 1980) was chosen. All windows were simulated with a 2 ft setback from exterior wall surfaces (Table 3.5).

Table 3.5. Wisenbaker Engineering Research Center exterior windows properties

Properties	Glass-type-code	Transmittance at normal incidence for overall solar spectrum, %	Reflectance at normal incidence for overall solar spectrum, %	Default conductance includes inside air film, but excludes outside air film, Btu-ft/Hr-ft^2-°F	Overall U-value (including outside air film) for 7.5 mph wind speed
Building window	9	50	30	1.47	1.02
Building door	5	61	6	1.47	1.02

The space zoning of WERC was determined based on the area served by the AHU without distinguishing interior and perimeter zones (since they were served by the same unit) and recent building drawings.

Figure 3.12 through Figure 3.15 show a plan view and space zoning used for the simulation of the basement, first floor, second and third floor, respectively.

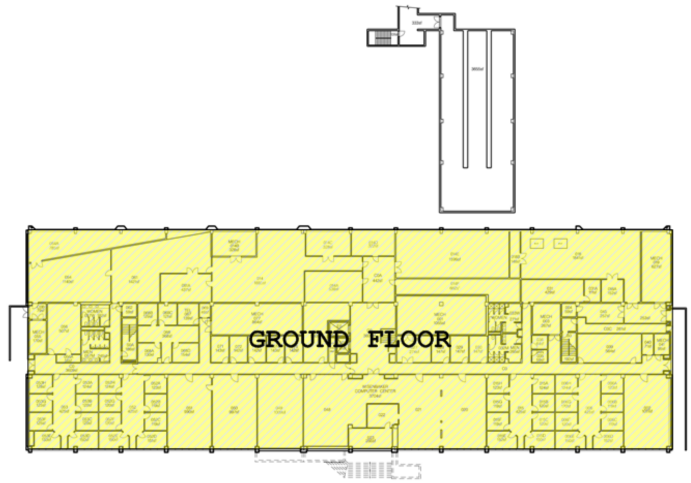


Figure 3.12. Wisenbaker Engineering Research Center basement zoning plan

Figure 3.13. Wisenbaker Engineering Research Center first floor zoning plan

Figure 3.14. Wisenbaker Engineering Research Center second floor zoning plan

Figure 3.15. Wisenbaker Engineering Research Center third floor zoning plan

Table 3.6 specifies WERC space conditions for the office area and the MTL. Building occupancy and people density were estimated based on observations. Maximum building office area occupancy was estimated at 200 ft^2/person. It is assumed that the maximum occupancy occurs between 8:00 am and 11:00 am in the morning and from 3:00 pm to 6:00 pm in the afternoon. Due to the presence of students in the building 24 hours a day, it is assumed that the minimum occupancy level is 5% all the time. For the weekends, it is assumed that occupancy is 10% from 9:00 am till 5:00 pm and 5% the rest of the time.

Lighting and equipment schedules were determined based on the WERC whole-building average energy consumption for weekdays, Saturday, Sunday and holidays individually. Maximum electrical consumption density by lighting and office equipment was estimated according to the ESL office area measured value. The occupancy schedule was developed based on observations and ESL typical lighting schedules. Lighting equipment density was assumed 1.2 $\mathrm{W/ft^2}$ with 20% of the energy rejected to the return air stream. Office equipment power density was assumed to be 0.8 $\mathrm{W/ft^2}$.

Table 3.6. Wisenbaker Engineering Research Center space conditions of the office and the Material testing laboratory area

DOE-2 Key word	Units	Office	Material testing laboratory	Description
TEMPERATURE	F	72 F	72 F	Midpoint of heating and cooling setpoint used for LOADS simulation
PEOPLE-SCHEDULE	-	OCCUPY-1	OCCUPY-1	Based on average ESL office area occupancy schedule
AREA/PERSON	ft^2/person	200	-	Based on average ESL area occupancy level
NUMBER-OF-PEOPLE	person	-	10	Based on the average area occupancy of MTL
PEOPLE-HEAT-GAIN	Btu/hr-person	-	800	Light bench work (ASHRAE 2009)
PEOPLE-HG-LAT	Btu/hr-person	200	-	Moderately active office work (ASHRAE 2009)
PEOPLE-HG-SENS	Btu/hr-person	250	-	Moderately active office work(ASHRAE 2009)
LIGHTING-SCHEDULE	-	LIGHTS-1	LIGHTS-1	Based on the WERC whole-building energy consumption
LIGHTING-TYPE	-	REC-FLUOR-RV	REC-FLUOR-NV	Recessed fluorescent RV -vented to return air
LIGHT-TO-SPACE		0.80	1.0	The fraction, if any, of the lighting energy that is added t o the space energy balance as a sensible heat gain. Based on observation
LIGHTING-W/SQFT	W/ft^2	1.2	1.2	The maximum overhead lighting energy use. Based on measured value
EQUIP-SCHEDULE	-	EQUIP-1	EQUIP-1	Based on the WERC whole-building energy consumption
EQUIPMENT-W/SQFT	W/ft^2	0.8	0.8	The maximum equipment energy use. Based on measured value
INF-METHOD	-	AIR-CHANGE	AIR-CHANGE	
AIR-CHANGES/HR	1/hr	0	2.0	
INF-SCHEDULE	-	INFIL-SCH1	INFIL-SCH2	
FLOOR-WEIGHT	Lb/ft^2	(Auto)	(Auto)	The composite weight of the floor, furnishings and interior walls of a space divided by the floor area of the space. Automatic calculated based on construction materials

No infiltration was assumed in the DOE-2 simulation for the office area since the HVAC system is always on and the building is assumed to be pressurized. For the material testing laboratory, infiltration was assumed in the amount of two air changes-per-hour since the loading dock door is open most of the time.

The building construction weight was estimated by a custom-weighting factor based on building construction material properties used in the simulation instead of DOE-2 default value of 70 lb/ft^2 for a medium construction. Using the DrawBDL program (Huang 2000), the front façade and north-east elevation of WERC is illustrated in Figure 3.16. Shading surfaces with zero transmittance were developed to account for the shading effect from the adjacent buildings.

The building was divided into two types of occupancy zones according to the operation conditions. The OFFICE zone conditions were used for all zones of the main building and the LAB conditions were used for the MTL located at the rear of the building.

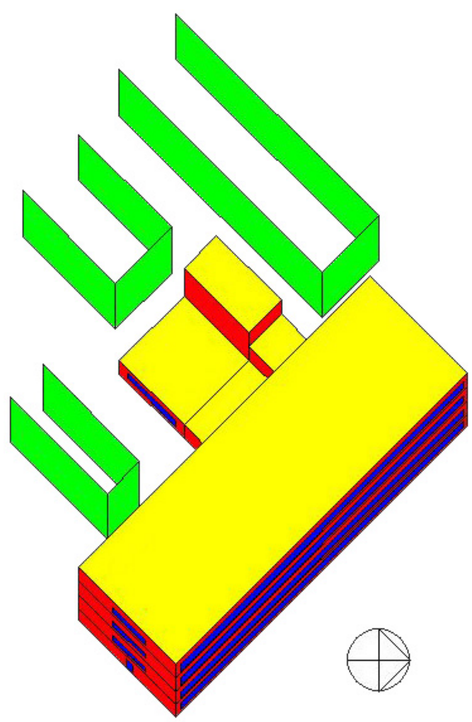


Figure 3.16. Wisenbaker Engineering Research Center building simulation model

3.5.2. Wisenbaker Engineering Research Center HVAC system simulation

As was mentioned in Chapter III Section 3.3, the first, second and third floor of the building are served by single-duct VAV systems operating in a constant volume mode. This part of HVAC system includes seven variable-volume air handlers that were simulated as SYST-1 in DOE-2.1E input file. The single-duct system has mixing boxes capable of reducing flow in response to a decrease in cooling demand for each control zone (Figure 3.17). The constant-volume air handler serves the MTL and is simulated as SYST-2. Three constant-volume air handlers units serve the basement of WERC and are simulated as SYST-3. Optional components of the systems are shown in dashed boxes. A heat recovery coil and humidifier described in Figure 3.17 are not available in the building and they are not enabled in the simulation.

Table 3.7. shows the initial input of the WERC HVAC system simulation.

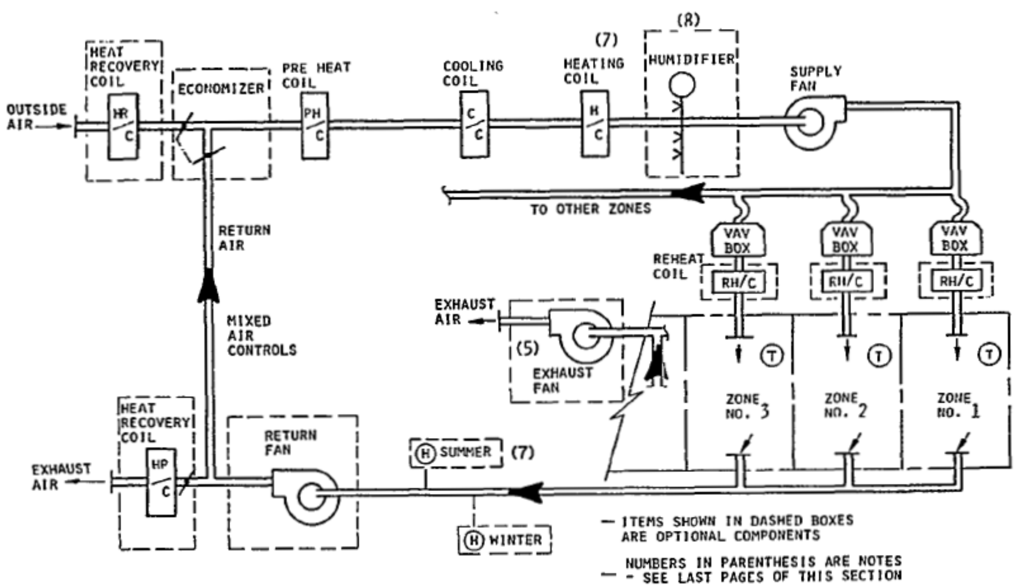


Figure 3.17. Single duct VAV systems with optional reheat (VAVS)
(LASL 1980)

Table 3.7. Initial parameters for the Wisenbaker Engineering Research Center HVAC system simulation model

Command	DOE-2.1 Keyword	Set-point for office zone	Material testing laboratory	Description
SPACE-CONDITIONS	-	OFFICE	LAB	
ZONE-AIR	CFM/SQFT	1 cfm/ft^2	1 cfm/ft^2	Minimum air flow rate to the zone per ft^2 of zone area
	OUTSIDE-AIR-CFM	Assigned for each zone. Total 17,402 cfm	320 cfm	The minimum flow rate of OA (0.1 cfm/ft^2)
ZONE-CONTROL	DESIGN-HEAT-T	68 ºF	68 ºF	The space temperature that the program uses to calculate the supply air flow rate required to meet peak heating loads for the zone
	DESIGN-COOL-T	78 ºF	78 ºF	The space temperature that the program uses to calculate the supply air flow rate required to meet peak cooling loads for the zone
	HEAT-TEMP-SCH	HEAT-SCHED	HEAT-SCHED	72ºF with no weekend reset
	COOL-TEMP-SCH	COOL-SCHED	COOL-SCHED	78ºF with no weekend reset
	THERMOSTAT-TYPE	REVERSE-ACTION	REVERSE-ACTION	VAV System
ZONE	SIZING-OPTION	ADJUST-LOADS	ADJUST-LOADS	The size of the supply air fan is determined by the block peak load
	ZONE-TYPE	CONDITIONED	CONDITIONED	The zone is heated and/or cooled, depending on the type of system selected.
SYSTEM-CONTROL	COOLING-SCHEDULE	COOLOFF	COOLOFF	Cooling is always available
	HEATING-SCHEDULE	HEATOFF	HEATOFF	Heating is always available
	MAX-SUPPLY-T	105 ºF	105 ºF	DOE-2 Default. The highest allowable temperature f or air supplied to the space
	MIN-SUPPLY-T	55 ºF	55 ºF	DOE-2 Default. The lowest allowable temperature f or air supplied to the space
SYSTEM-FANS	FAN-SCHEDULE	FAN-SCHED	FAN-SCHED	Fans are always on
	FAN-CONTROL	SPEED	SPEED	
	SUPPLY-STATIC	3.0 inH$_2$0	3.0 inH$_2$0	The total pressure produced by the system supply fan at design flow rate
	SUPPLY-EFF	0.7	0.7	the combined efficiency of the zone supply fan and motor at design condition
	MOTOR-PLACEMENT	IN-AIRFLOW	IN-AIRFLOW	
	RETURN-STATIC	2.4 inH$_2$0	None	The total pressure produced by the system return fan at design flow rate
	RETURN-EFF	0.7	None	the combined efficiency of the zone return fan and motor at design condition
SYSTEM-TERMINAL	REHEAT-DELTA-T	55 ºF	55 ºF	The maximum increase in temperature for supply air passing through all the zone reheat coils in the system.
	MIN-CFM-RATIO	1	1	System simulated in constant air flow operation according to AHU measured values.
SYSTEM	SYSTEM-TYPE	VAVS	VAVS	Variable Volume Fan System with reheat
	ECONO-LIMIT-T	55 ºF	55 ºF	Economizer mode is not available for the system
	ECONO-LOW-LIMIT	55 ºF	55 ºF	Economizer mode is not available for the system
	RETURN-AIR-PATH	DUCT	DUCT	Route that return air takes in getting back to the air handling unit

The cooling (76ºF) and heating (70ºF) setpoints were developed based on ASHRAE Standard 55-2004 comfort chart. Reverse-action type thermostats were used for modeling the VAV system. The thermostat allows the supply air flow rate to be controlled above the minimum flow rate as defined in the MIN-CFM-RATIO command. This ratio was set to one based on an observed AHUs constant supply air fan speed. The main draw-through fans in each AHU run to circulate the conditioned air in the building. The supply air temperature after the cooling coil was set to 55ºF, based on default DOE-2 values. Minimum outside air flow was assigned to each zone according to the zone area. The AHU fans are always on and the supply static pressure is 3.0 inH_20.

Wisenbaker Engineering Research Center building ChW and HHW are provided from the central utility plant. Since the central utility plant serves more than one building and energy bought from it has a fixed energy price, the efficiency of the central utility plant was not analyzed.

3.6. As-built building model calibration

The as-built building simulation energy model described in Chapter III, Section 3.4 - 3.5 was calibrated until it reached an acceptable level of conformity.

3.6.1. Statistical method used for building model calibration

The variability of a data set can be measured in different ways. The simplest way is the difference between the minimum and maximum level. A more efficient way is the application of a coefficient of variance and a mean bias error (MBE) or coefficient of variance of the root mean square error and MBE. To calibrate the simulation model to an acceptable level of energy consumption, a conformity analysis of residuals could be used. According to Liu et al. (2003), the simulation results based on design data and measured consumption could have differences of more than 50%. The main cause for this is errors in the input assumption for a particular building.

The building calibration process based on the calibrating signature method and calibration signatures for the single duct constant air volume system is described by Liu et al. (2003).

The calibration signature is a normalized plot of the difference between measured energy consumption values and the corresponding simulated values as a function of outdoor air temperature (Liu et al. 2003). For calibration, data on the daily average basis were used. Calibration signature values for heating or cooling energy consumptions are calculated by the formula:

$$\text{Calibration signature} = \frac{\text{- Residual}}{\text{Maximum measured energy}} \cdot 100\% \ , \tag{3.1}$$

where

$$\text{Residual} = \text{Simulated consumption - Measured consumption}$$

Calibration signatures can be defined for each simulation. By comparison of the shape between simulated calibration signatures and published calibration signatures, parameter that needs to be adjusted could be defined. Published characteristic calibration signatures are defined as follows:

$$\text{Characteristic signature} = \frac{\text{Change in energy consumption}}{\text{Maximum energy consumption}} \cdot 100\% \, . \tag{3.2}$$

Calibration signatures define the change in energy consumption (cooling or heating energy consumption value) with the change input minus the baseline value at the same temperature. The denominator value is defined over the entire range of OA temperatures contained in the weather file. The calibration process is based on changes in both heating and cooling signatures that typically show very different trends, and their combinations indicate those parameters that need to be changed. Characteristic signatures also depend on the humidity value of OA and a type of HVAC system (Liu et al. 2003).

The main parameters that can be used for the evaluation of calibration steps and the results of calibration processes are the Root Mean Square Error and Mean Bias Error.

The Root Mean Square Error (RMSE) is defined according to (3.3), and it reflects overall magnitude of the error.

$$RMSE = \sqrt{\frac{\sum_{i=1}^{n} \text{Residual}_i^2}{n-2}}, \tag{3.3}$$

where n - number of data points.

The Mean Bias Error (MBE) is defined according to (3.4), and it reflects bias of simulation results:

$$MBE = \frac{\sum_{i=1}^{n} \text{Residual}_i}{n}, \tag{3.4}$$

where n - number of data points.

Higher values of MBE can indicate potential errors in simulation input, meanwhile higher values of RMSE can indicate the influence of some not modeled factors. Since the HHW energy consumption is significantly less than the ChW consumption in hot climates, RMSE errors of HHW energy consumption have significantly less influence on total simulation error.

3.6.2. DOE-2.1E output file analysis

The DOE-2.1E simulation provides detailed information about the simulated building in the output file. The information reported in the output file categorized into

SYSTEMS-REPORT groups: SUMMARY, VERIFICATION and energy consumption HOURLY-REPORTs.

All four main sections of the DOE-2 simulation program (loads, system, plants, and economics) have separate reports. The summary reports provide the actual summary of building simulation results, and verification reports provide information about the building properties specified in the input file.

For calibration purposes the most convenient are hourly reports. This type of report provides hourly changes of those parameters defined in the input file. Alternatively, this report can provide both daily and monthly average and total values of energy consumption. The complete list of the variables that can be selected as output in the hourly reports is described in the DOE-2.1E Supplement (LBNL 1993b). The Hourly-report variable list provides type and number of variables for the current version of the program.

Hourly reports are generated for a time period defined in the input file that should not exceed 8760 data points or one year of simulation. AWK programming language executive file was used for data extraction from output file. The most important report parameters for calibrations are ChW, HHW usage and whole-building electricity consumption. Since building is supplied ChW and HHW from the plant, these variables are sufficient for calibration.

For current research, hourly reports extracted from the system portion of the output file were used, as well as summary and verification reports. Variables used in the calibration process obtained from hourly reports are summarized in Table 3.8

Table 3.8. DOE-2.1E system hourly reports variables

Variable-type	Variable list number	Variable in Fortran code	Description
GLOBAL	1	IYR	Year of simulation run
GLOBAL	2	IMO	Month of simulation run
GLOBAL	3	IDAY	Day of simulation run
GLOBAL	4	MR	Hour of simulation run
u-name of PLANT-ASSIGNMENT	1	QCPL	Total cooling load (Btu/hr)
u-name of PLANT-ASSIGNMENT	2	QHPL	Total heating load (Btu/hr)
u-name of PLANT-ASSIGNMENT	3	PKW	Total electrical load, kW

3.6.3. Initial simulation results

The initial simulation was used to estimate the basic energy consumption pattern of WERC building by using Typical Meteorological Year version 3 (TMY3) weather data for College Station Easterwood Airport, TX (U.S. DOE 2010d). The simulation was performed based on the building design data and the results of the energy consumption measurements and observations. Hourly simulation results of the HVAC system, equipment and lighting energy consumption were combined to obtain the total electrical load profile and then compared with the whole-building measured energy consumption.

The chilled water energy consumption plot vs. OA temperature shows the set of data points that could be considered outliers (Figure 3.18). All outliers data belongs to the same time interval 6/4/2008 - 7/1/2008 and could be explained by an error in the energy consumption measurement system since a decrease in ChW consumption occurred simultaneously with the whole-building electrical energy consumption increase (Figure 3.19). The outlier's data set was excluded from the RMSE and MBE estimation for all energy categories.

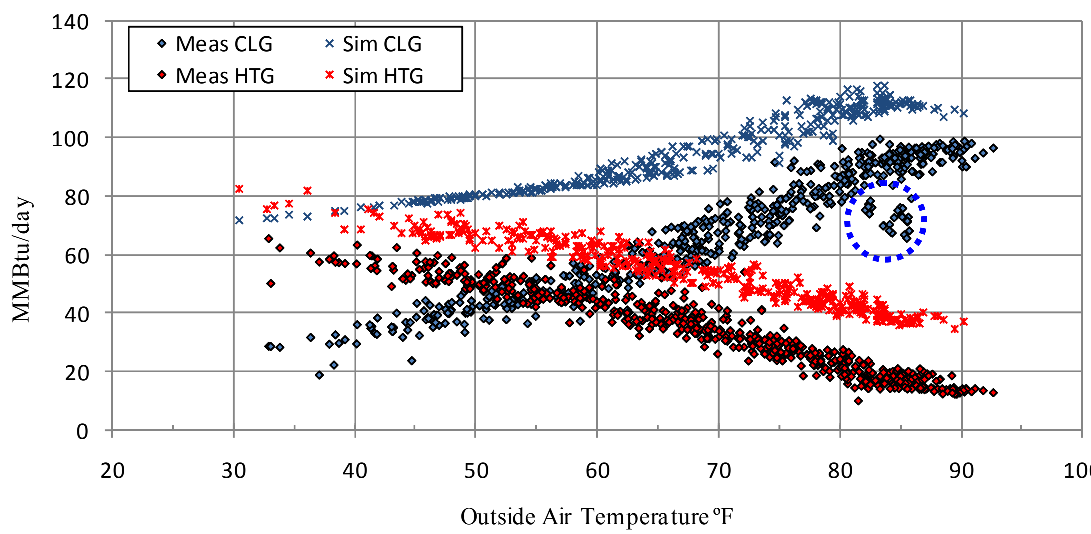


Figure 3.18. Initial building simulation results and measured chilled water (CLG) and heating hot water (HTG) energy consumption

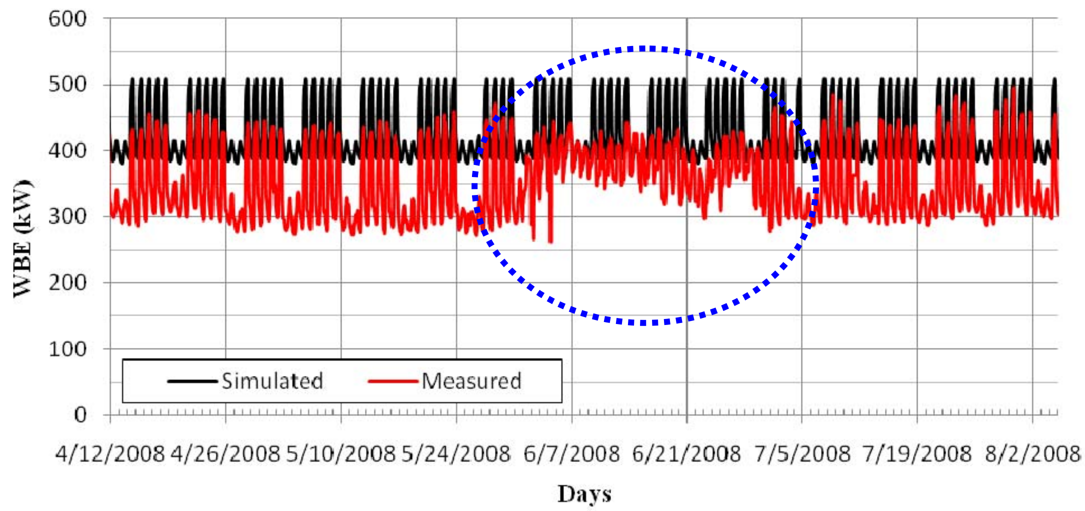


Figure 3.19. Initial building simulation results and measured whole-building electrical energy consumption (WBE)

Initial building simulation error is described in Table 3.9. Calibration signatures developed from the as-built base model plotted versus OA dry-bulb temperature are depicted in Figure 3.20.

Table 3.9. Initial Wisenbaker Engineering Research Center building energy consumption simulation error

Category	MBE,	RMSE, %
Chilled water	27.40 MMBtu/day	29.55
Heating hot water	20.33 MMBtu/day	21.48
Whole building electricity	1.79 MWh/day	1.898

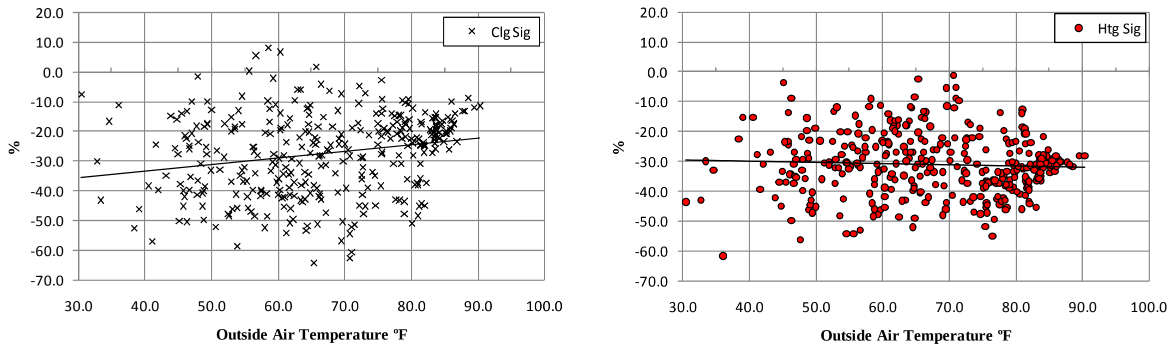


Figure 3.20. Initial calibration cooling and heating signatures for Wisenbaker Engineering Research Center simulation

3.6.4. Calibration step 1: Supply air flow rate adjustment

The calibration signatures in Figure 3.20 indicate that simulated cooling and heating energy consumptions should be decreased by up to 30 % over all dry-bulb temperature range. The simulated electricity use should also be increased in the overall temperature range. In this calibration step, the supply air flow rate was changed from $1\ \mathrm{cfm/ft^2}$ to $0.5626\,\mathrm{cfm/ft^2}$. Table 3.10 shows simulation errors, and Figure 3.21 shows calibration signatures after supply air flow rate adjustment.

Table 3.10. Calibration step 1. Wisenbaker Engineering Research Center simulation model error

Category	MBE	RMSE, %
Chilled water	-8.14 MMBtu/day	15.21
Heating hot water	-9.27 MMBtu/day	11.48
Whole building electricity	0.2031 MWh/day	0.645

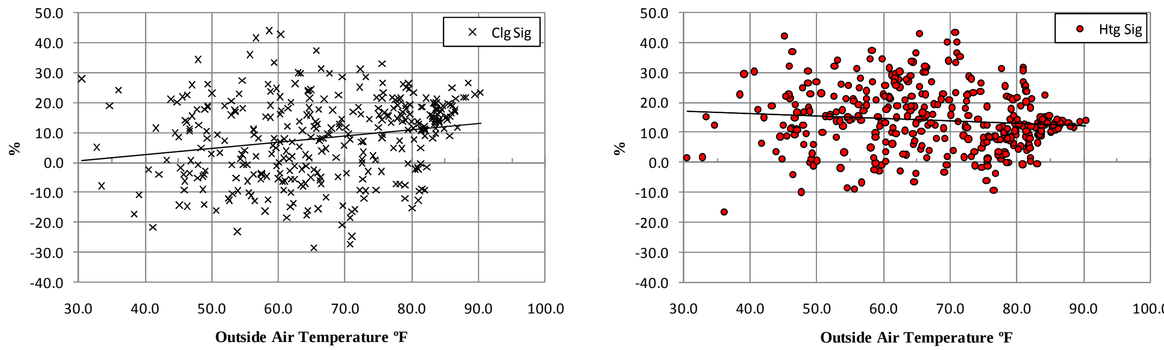


Figure 3.21. Calibration step 1. Cooling and heating signatures

This supply air flow rate adjustment decreased the MBE (ChW) from 27.40 MMBtu/day to -8.14 MMBtu/day and the MBE (HHW) from 20.33 MMBtu/day to -9.27 MMBtu/day. The whole building electricity RMSE decreased more than twice from 1.898% to 0.645%.

3.6.5. Calibration step 2: Outside air flow rate adjustment

In the next calibration step, the outside air flow rate was adjusted. This decision was made since calibration cooling signatures for ChW energy consumption increases with increase of OA temperature (Figure 3.21). The outside air flow rate was adjusted from 0.1 cfm/ft^2 (total 17,401 cfm) to 0.148 cfm/ft^2 (total 25,748 cfm). This flow rate adjustment decreased the MBE (ChW) from -8.14 MMBtu/day to -7.25 MMBtu/day (Table 3.11). Figure 3.22 shows calibration signatures after OA flow rate was adjusted.

Table 3.11. Calibration step 2. Wisenbaker Engineering Research Center simulation model error

Category	MBE	RMSE, %
Chilled water	-7.25 MMBtu/day	14.77
Heating hot water	-9.27 MMBtu/day	11.42
Whole building electricity	0.2031 MWh/day	0.645

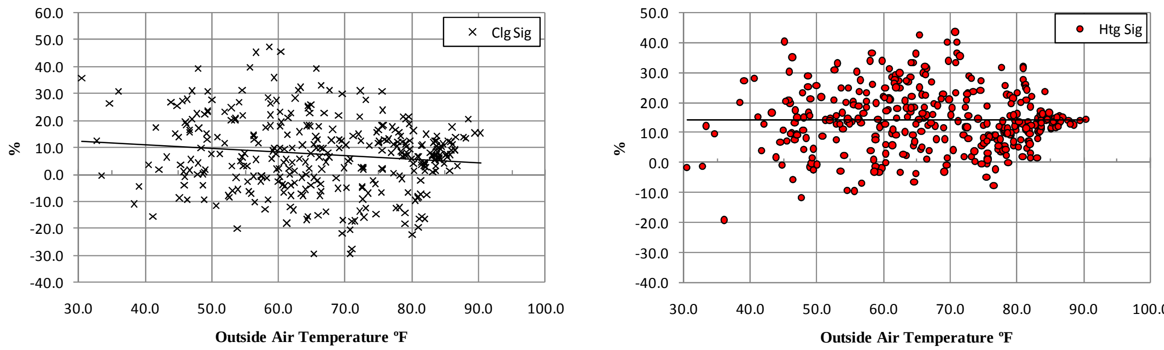


Figure 3.22. Calibration step 2. Cooling and heating signatures

3.6.6. Calibration step 3: Minimum supply air temperature adjustment

In the next calibration step the minimum supply air temperature was adjusted from 55ºF to 53ºF. This adjustment decreased the MBE (ChW) from -7.25 MMBtu/day to -4.72 MMBtu/day and the MBE (HHW) from -9.27MMBtu/day to -7.61 MMBtu/day (Table 3.12). Figure 3.23 shows calibration signatures after minimum supply air temperature was adjusted.

Table 3.12. Calibration step 3. Wisenbaker Engineering Research Center simulation model error

Category	MBE	RMSE, %
Chilled water	-4.72 MMBtu/day	13.94
Heating hot water	-7.61 MMBtu/day	10.11
Whole building electricity	0.1600 MWh/day	0.633

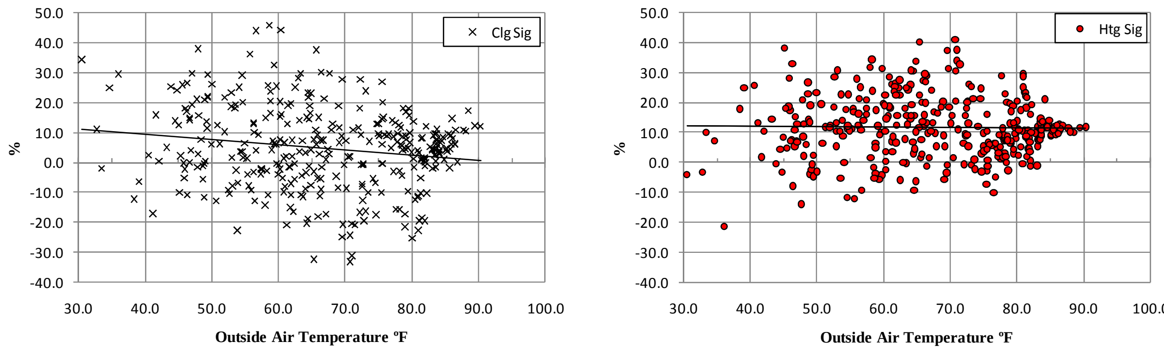


Figure 3.23. Calibration step 3. Cooling and heating signatures

3.6.7. Calibration step 4: Room air temperature set points adjustment

In this calibration step, the room air temperature range was changed from 72-78ºF to 75-76ºF. WERC zone air temperature setpoint does not have a seasonal or daily adjustment that was confirmed based on the observation and the data from portable temperature loggers. This adjustment decreased the MBE (ChW) from -4.72 to -0.27 and the MBE (HHW) from -7.61 MMBtu/day to 0.25MMBtu/day (Table 3.13). Figure 3.24 shows calibration signatures after minimum supply air temperature was adjusted.

Table 3.13. Calibration step 4. Wisenbaker Engineering Research Center simulation model error

Category	MBE	RMSE, %
Chilled water	-0.27 MMBtu/day	13.19
Heating hot water	0.25 MMBtu/day	6.62
Whole building electricity	0.16 MWh/day	0.633

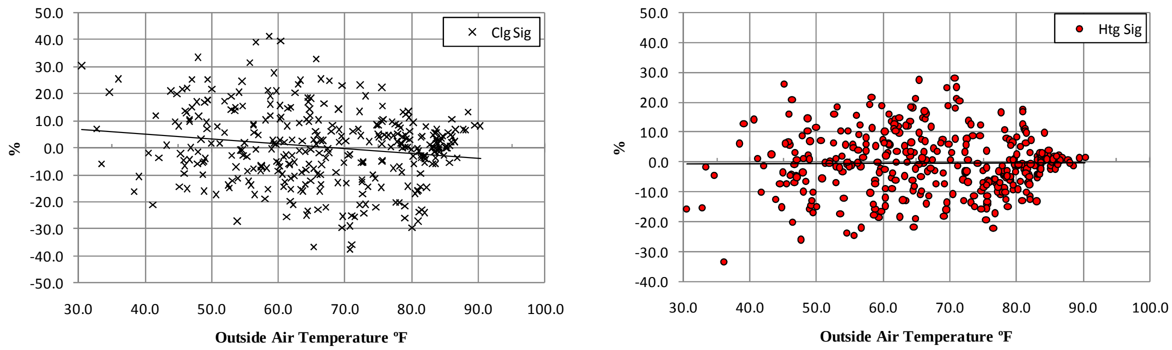


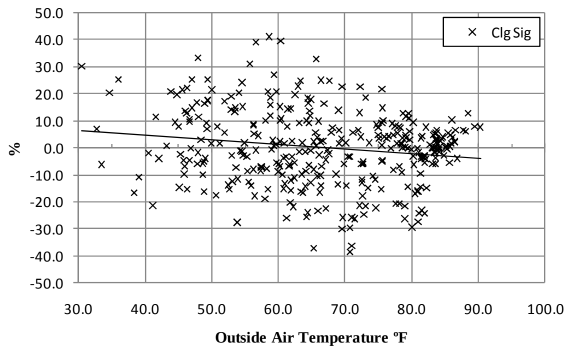
Figure 3.24. Calibration step 4. Cooling and heating signatures

3.6.8. Calibration step 5: Supply and return fans static pressure adjustment

In this calibration step, the supply fans static pressure was adjusted from 3 in$H_2$0 to 4.1 inH20 and the return fans static pressure was adjusted from 2.4 in$H_2$0 to 0.9 in$H_2$0. This adjustment decreased the MBE (ChW) from -0.27 to 0.011 and the MBE (HHW) from 0.25 MMBtu/day to 0.084 MMBtu/day (Table 3.14). Figure 3.25 shows calibration signatures after supply fans static pressure was adjusted.

Table 3.14. Calibration step 5. Wisenbaker Engineering Research Center simulation model error

Category	MBE	RMSE, %
Chilled water	0.011 MMBtu/day	13.212
Heating hot water	0.084 MMBtu/day	6.590
Whole building electricity	0.0462 MWh/day	0.614



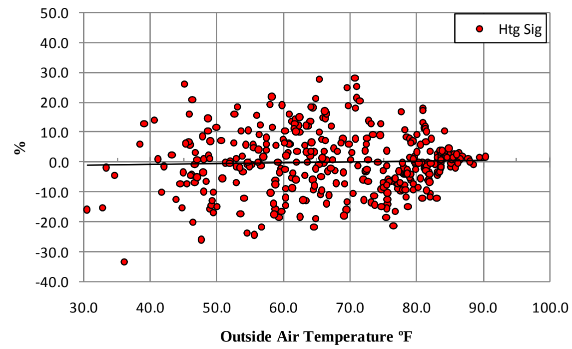


Figure 3.25. Calibration step 5. Cooling and heating signatures

3.6.9. Calibration summary

As described in Section 3.5, the WERC energy consumption model was calibrated to the measured energy consumption level during the whole 2008 year by changing input parameters of the baseline model and analyzing characteristic signature plots. During the calibration process, the MBE (ChW) was reduced from 27.40 MMBtu/day to 0.011 MMBtu/day and the MBE (HHW) from 20.33 MMBtu/day to 0.084 MMBtu/day (Figure 3.26). The HHW RMSE was reduced to 6.6% and the ChW RMSE to 13.2%.

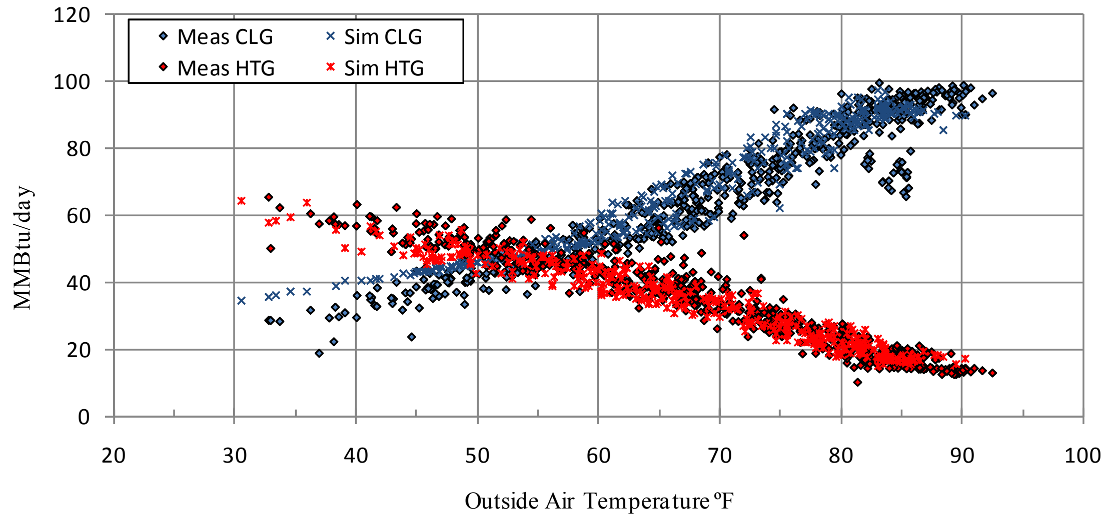


Figure 3.26. Wisenbaker Engineering Research Center calibrated model simulation results and measured chilled (CLG) and heating hot water (HTG) energy consumption

The whole building calibrated model electrical energy consumption has RMSE 0.614% and MBE (0.0462 MWh/day), which is less than 0.6 % from the building energy consumption level (Figure 3.27).

The DOE-2.1E input program code for the calibrated WERC energy consumption model is given in Appendix A.

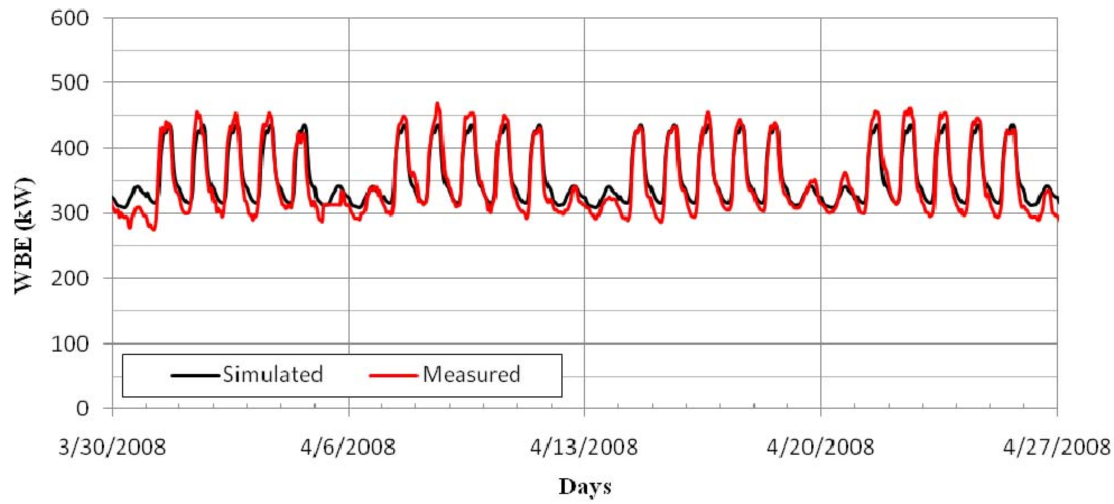


Figure 3.27. Wisenbaker Engineering Research Center calibrated model simulated and measured whole-building electricity consumption

Analysis of simulated and measured data trends for HHW and ChW shows some disagreement of energy consumption on short time intervals (Figure 3.28, Figure 3.29).

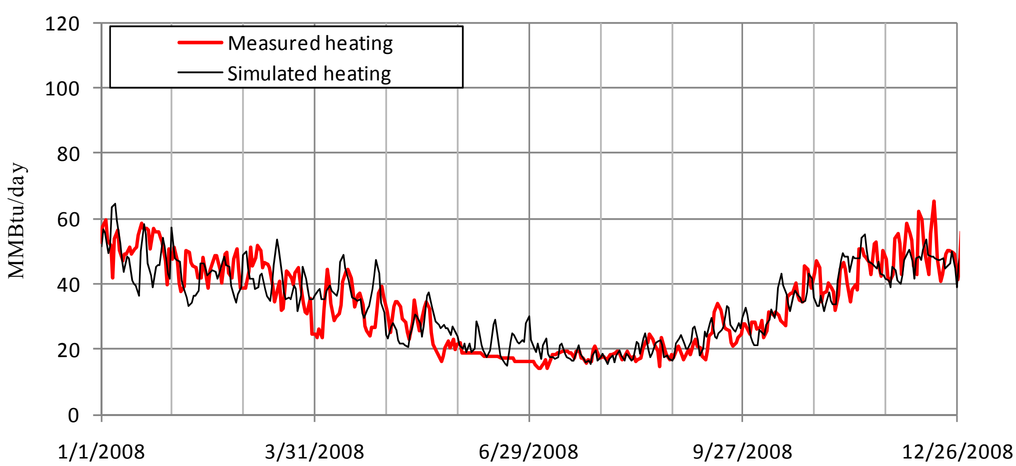


Figure 3.28. Wisenbaker Engineering Research Center calibrated model simulated and measured heating hot water energy consumption

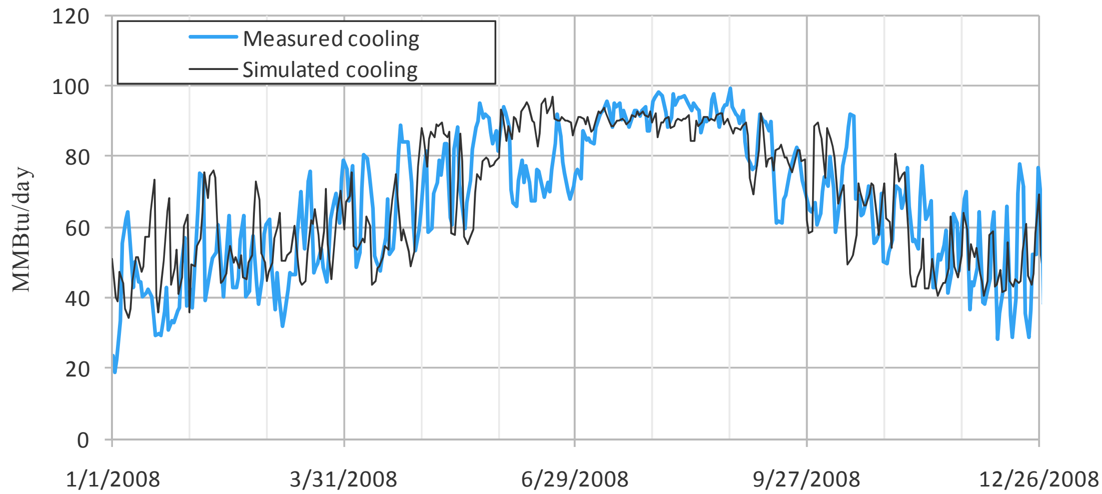


Figure 3.29. Wisenbaker Engineering Research Center calibrated model simulated and measured chilled water energy consumption

One of the factors that influence the calibrated model error is the difference between typical ambient parameters given in the TMY3 weather file and the actual measured ambient parameters. Figure 3.30 and Figure 3.31 shows that higher values of residuals and calibration signature dataset appear when differences between typical temperature and actual measured temperature increase.

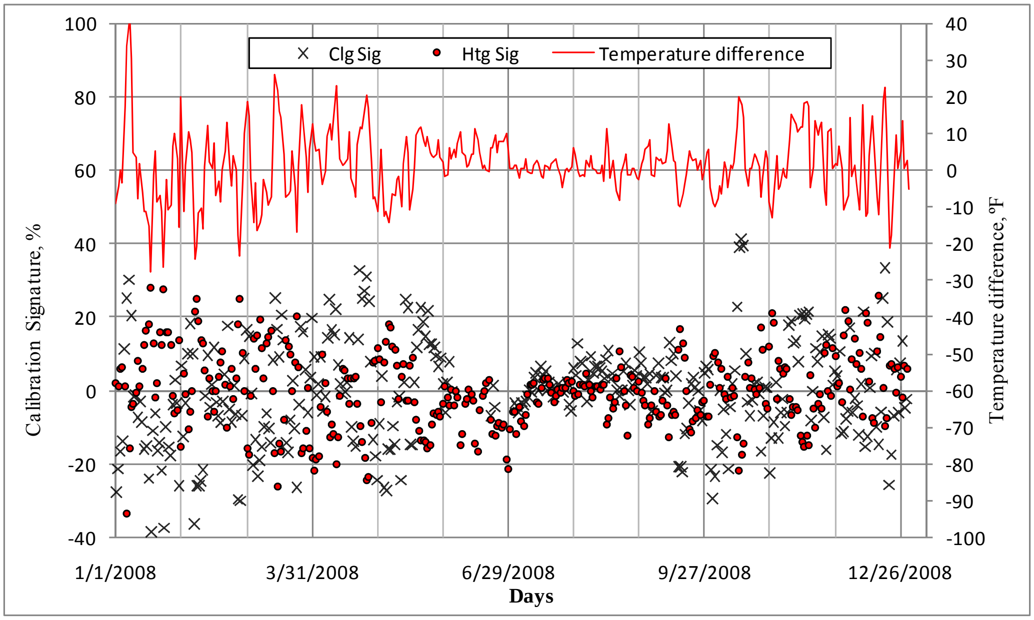


Figure 3.30. Wisenbaker Engineering Research Center calibrated model calibration signatures and difference between measured and TMY3 outside air temperature

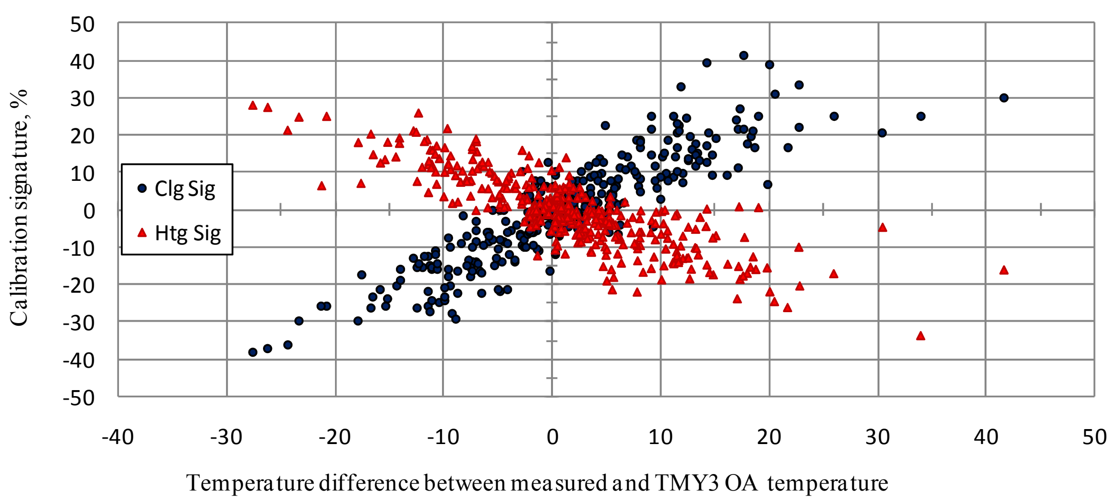


Figure 3.31. Wisenbaker Engineering Research Center calibration signatures value vs. outside air temperature difference between measured data and TMY3 weather file data

3.7. DOE-2.1E and Windows Air Model 4.0 calibrated model results comparison

Calibration results obtained from the DOE-2.1E program were compared with calibration results obtained from the Windows Air Model (Win AM 4.0). Win AM 4.0 is an independent whole building simulation program developed by the Texas Engineering Experiment Station ESL. The program is based on a simplified calculation algorithm and does not account for several parameters such as the building thermal capacity or solar radiation gain through the windows. As an input weather file, actual measured weather data for College Station, Texas were used during the simulation period.

An independent calibration process was conducted in Win AM 4.0, based on the same baseline model assumption that was used in DOE-2.1E simulation.

Comparisons of calibrated model parameters in both simulations show similar results (Table 3.15). The main difference between the two models is lower zone temperatures and higher supply and outside air flow rates in Win AM 4.0.

Table 3.15. Simulation parameters comparison between DOE-2.1E and Win AM 4.0 calibrated models

Parameter	Units	DOE-2.1E	Win AM 4.0
Chilled water pump	hp	19.7	25
Heating hot water pump	hp	4.5	5
System		Single duct constant air volume with reheat	Single duct constant air volume with reheat
Preheat		None	None
Conditioned floor area	ft^2	174,016	174,016
Interior zone percentage		None	70.0%
Exterior walls area/conduction	$ft^2/Btu/ft^2hrF$	36,158 / 0.105	36,158 / 0.105
Exterior windows	$ft^2/Btu/ft^2hrF$	14,242 / 1.142	14,242 / 1.142
Roof	$ft^2/Btu/ft^2hrF$	57,600 / 0.194	57,600 / 0.194
Max lighting usage	W/ft^2	1.2 (Based on ESL 2 floor consumption)	1.2 (Based on ESL 2 floor consumption)
Max Plug usage	W/ft^2	0.8 (Based on ESL 2 floor consumption)	0.8 (Based on ESL 2 floor consumption)
Average work space per person	ft^2/person	200	200
Average sensible heat gain per person in office zone	Btu/hr	250	250
Average latent heat gain per person in office zone	Btu/hr	200	200

Table 3.15. (Continued)

Parameter	Units	DOE-2.1E	Win AM 4.0
Temperature set points	°F	75	72
Volume flow rate	cfm/ft^2	0.5626	0.65
Outside air usage	cfm/ft^2	0.148	0.17
Economizer		No	No
Max fan electric usage	hp	97/19.4	97/19.4
Supply fan type		VFD	VFD
Cooling coil setpoint	°F	53.0-54.0	52.0-53.0
Heat gain schedule		Based on the WERC electricity trend	Based on the WERC electricity trend

The calibrated Win AM 4.0 model shows a better agreement of ChW energy consumption compared to DOE-2.1E (Figure 3.32).

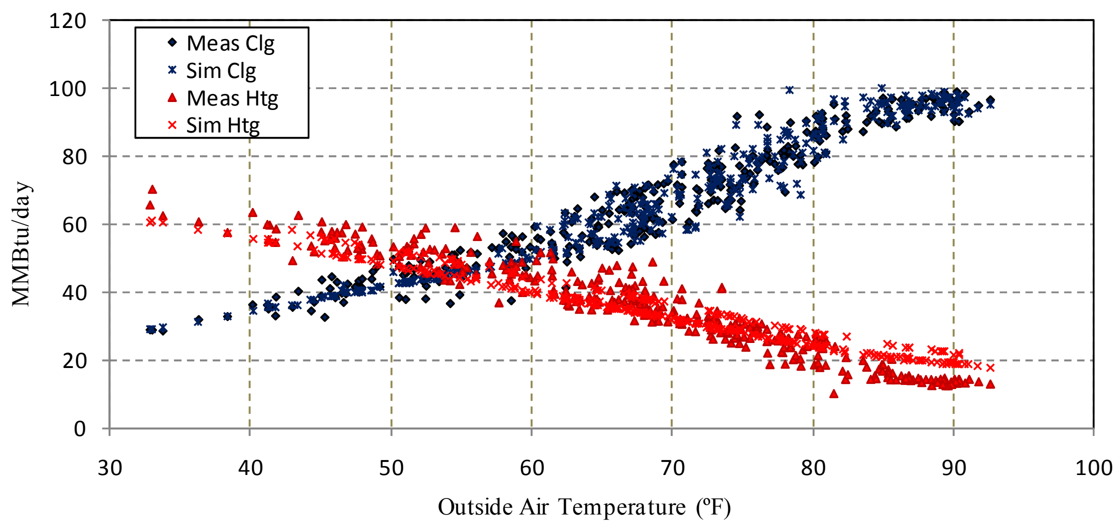


Figure 3.32. Wisenbaker Engineering Research Center calibrated model simulation results developed in Win AM 4.0

At the end of the calibration process, the MBE (ChW) was reduced to -0.13 MMBtu/day and the MBE (HHW) to 0.04 MMBtu/day, which is similar to DOE-2.1E results, but the RMSE in Win AM 4.0 was reduced to 3.77% for cooling and 5.14% for heating, which is better compared with DOE-2.1E model (Figure 3.33).

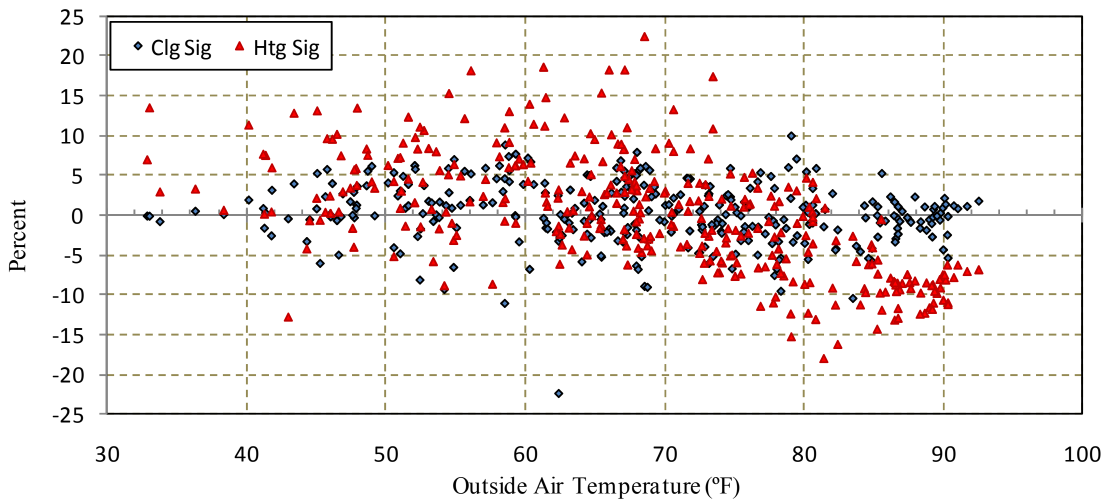


Figure 3.33. Daily calibration signatures for Wisenbaker Engineering Research Center calibrated model developed in Win AM 4.0

Dynamic trends at short time intervals for simulated HHW (Figure 3.34) and ChW energy consumption (Figure 3.35) shows much higher agreement with measured data in the Win AM 4.0 model compared with the DOE-2.1E simulation.

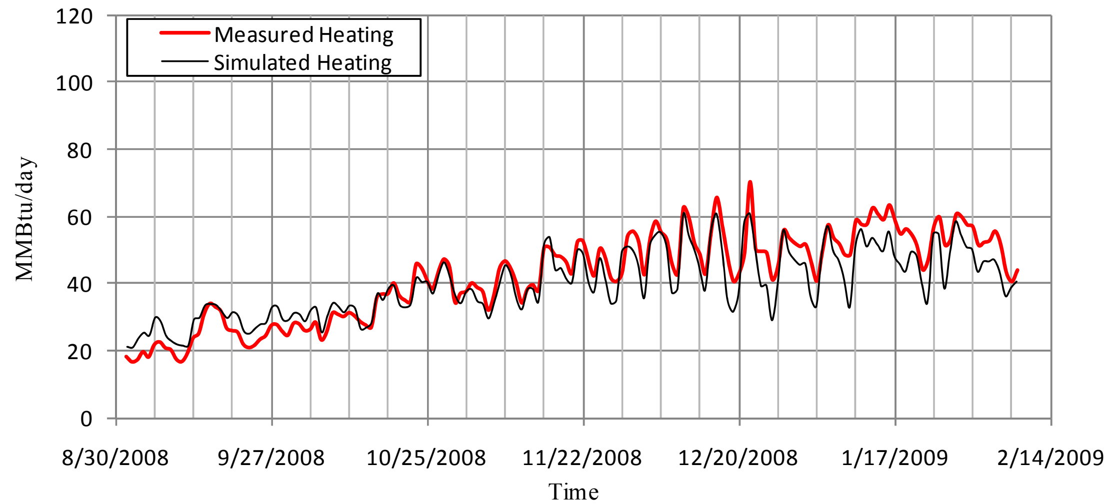


Figure 3.34. Heating energy consumption simulation results for Wisenbaker Engineering Research Center calibrated model developed in Win AM 4.0

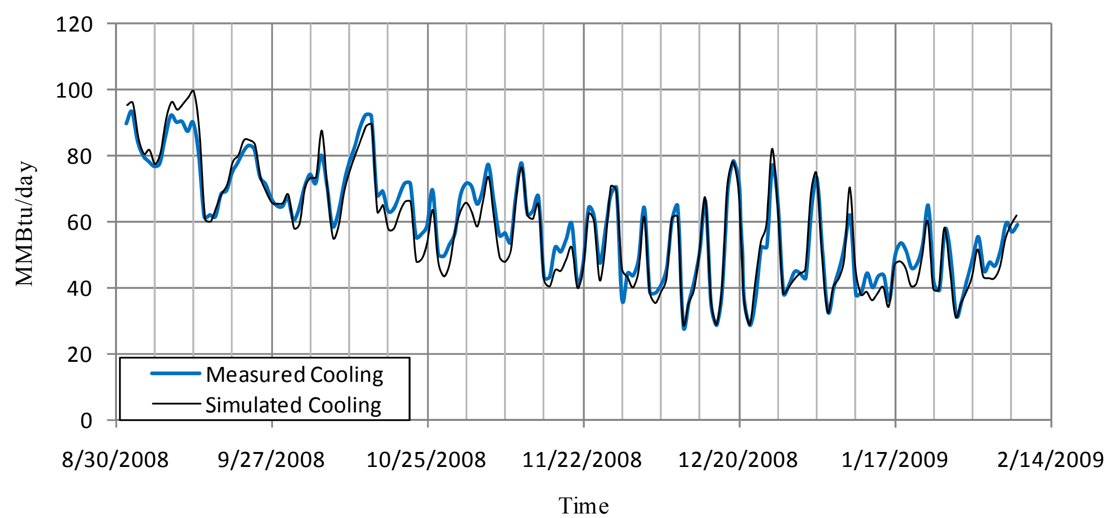


Figure 3.35. Cooling energy consumption simulation results for Wisenbaker Engineering Research Center calibrated model developed in Win AM 4.0

The main reason for better results obtained in the Win AM 4.0 simulation is usage of actual weather data instead of typical weather profile.

CHAPTER IV

ENERGY SAVING MEASURES FOR WISENBAKER ENGINEERING RESEARCH CENTER

4.1. Energy consumption analysis

Based on the existing WERC simulation model a series of high performance measures were developed according to recommendation from Liu et al. (2002).

The output file of the calibrated WERC energy consumption model simulation provided the AHU supply fans power consumption of 72.40 kW. This value differs from observation results (71.82 kW) less than 1%. The simulated power consumption of the return air fans is 14.50 KW, which differ from observation results (15.67 kW) by less than 8% (Table 4.1).

Table 4.1. Wisenbaker Engineering Research Center supply and return fans simulation model parameters

AHU	SUPPLY FAN			RETURN FAN			OUTSIDE AIR RATIO
	Flow	Power	DELTA-T	Flow	Power	DELTA-T	
	CFM	KW	°F	CFM	KW	°F	
1-7	74,615	50.79	2.1	74,615	11.148	0.5	0.263
8	9,306	6.33	2.1	-	-	-	0.051
1B,2B,3B	22,447	15.28	2.1	22447	3.354	0.5	0.263
Total	106,368	72.40		97,062	14.50		0.58

Based on the SS-N output file report, most of the time the humidity level is between 40-49% RH for the office area. During the year the maximum air humidity level does not exceed 60%. Based on the monthly energy end-use summary, the highest monthly energy consumption occurs in July (471,681 kWh) and the lowest in February (369,126 kWh) (Figure 4.1).

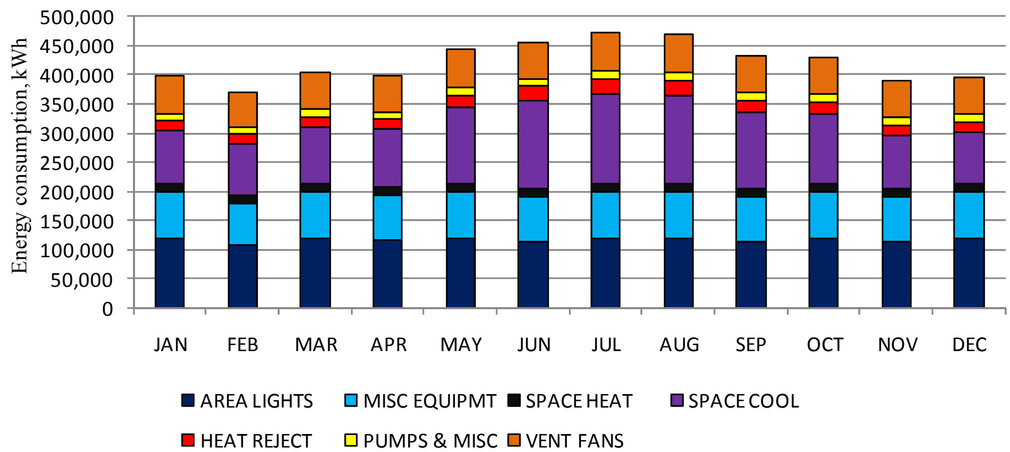


Figure 4.1. Wisenbaker Engineering Research Center end-use monthly electrical energy consumption

Total site energy consumption for WERC, according to the simulation reports of the calibrated model, is 200.4 $\text{kBtu/ft}^2\text{yr}$ (29.082 $\text{kWh/ft}^2\text{yr}$ of electrical energy and 1.011 $\text{term/ft}^2\text{yr}$ of natural gas). These values include the energy consumed in the Central Utility Plant to generate the ChW and HHW that is consumed in the building, based on assumed default energy efficiency level of the equipment. Main energy consumption categories are: space cooling, lighting and equipment load. Total annual building site energy consumption is 17,272 MMBtu of electrical energy (Figure 4.2) and 17,592.6 MMBtu of natural gas.

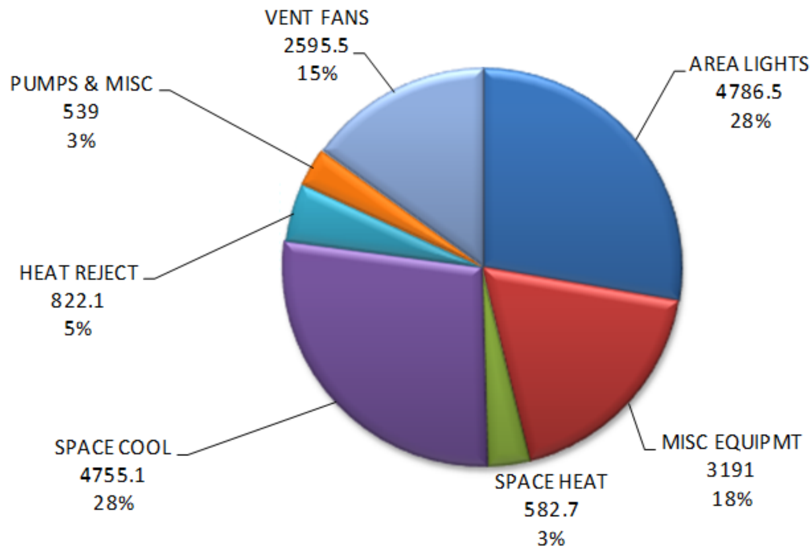


Figure 4.2. Wisenbaker Engineering Research Center annual site building electrical energy consumption summary in MMBtu

4.2. Estimation of energy savings potential in Wisenbaker Engineering Research Center

Based on the WERC calibrated energy consumption model, the following energy saving measures were proposed: adjustment of minimum supply air flow of VAV boxes, OA flow adjustment, AHU-8 (MTL) operation schedule improvement, lighting system improvement, economizer mode utilization, AHU 1-7 operation schedule improvement.

4.2.1. High performance measure 1: VAV boxes minimum supply air flow adjustment

Based on the observed energy consumption by the VFD supply fans, the HVAC system operates in the constant volume mode for all AHU. This is caused by malfunctions of VAV boxes connected to the HVAC system. In case if VAV boxes are repaired, the HVAC system simulated as SYST-1 can reduce the minimum air flow from 100% to at least 30%. The reduction of the minimum air flow by 70% will provide annual energy savings of 9,328.5 MMBtu (37.4%) for cooling energy and 6724.4 MMBtu (54.8%) for heating energy, that decreasing energy consumption rates to 131 $\text{kBtu/ft}^2\text{yr}$. Table 4.2 and Table 4.3 show simulation results including energy end-use for each improvement.

Table 4.2. Wisenbaker Engineering Research Center improvement measures comparison

#	System improved	DOE-2 command	Base Model	Improvement						
				1	2	3	4	5	6	1-6
1	SYST-1	MIN-CFM-RATIO	1.0	0.3	1.0	1.0	1.0	1.0	1.0	0.3
2	SYST-1,2,3	Total outside air flow (cfm/ft^2)	0.148	0.148	0.085	0.148	0.148	0.148	0.148	0.085
3	SYST-2	FAN-SCHEDULE	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED2	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED2
4	OFFICE zone	LIGHTING-W/SQFT	1.2	1.2	1.2	1.2	0.8	1.2	1.2	0.8
5	SYST-1	ECONO-LOW-LIMIT / ECONO-LIMIT-T	55/55	55/55	55/55	55/55	55/55	60/30	55/55	60/30
6	SYST-1	FAN-SCHEDULE	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED	FAN-SCHED2	FAN-SCHED2

Table 4.3. Wisenbaker Engineering Research Center improvement measures annual energy consumption summary

Category	Base Model	High performance measure energy consumption						
		1	2	3	4	5	6	Total 1-6
Area Light (MWh)	1,402	1,402	1,402	1,402	944	1,402	1,402	944
Office Equipment (MWh)	935	935	935	935	935	935	935	935
Ventilation fans (MWh)	760	338	760	743	760	760	607	301
Chilled water (MMBtu)	24,919	15,590	24,413	24,253	24,690	22,136	20,428	12,633
Heating hot water (MMBtu)	12,262	5,538	12,373	11,540	13,391	12,980	7,376	4,547
Total Electricity (MWh)	3,098	2,675	3,098	3,081	2,639	3,098	2,945	2,180
Total site energy kBtu/ft^2-yr	200	131	198	194	200	204	158	108

4.2.2. High performance measure 2: Outside air flow adjustment

According to the calibrated building simulation model the total outside supply air flow is 25,747 cfm or 0.148 cfm/ft^2. According to ASHRAE 62.1-2004, the required breathing zone outdoor airflow in the occupiable space (i.e. the design outdoor airflow required in the breathing zone) should be determined in accordance with equation:

$$V_{bz} = R_p \cdot P_z + R_a \cdot A_z \,, \tag{4.1}$$

where:

A_z - zone floor area: the net occupiable floor area of the zone ft^2;

P_z - zone population: the largest number of people expected to occupy the zone during usage;

R_p - outdoor airflow rate required per person as determined from (ASHRAE 2004);

R_a - outdoor airflow rate required per unit area as determined from (ASHRAE 2004).

The zone air distribution effectiveness (E_z) is determined according to (ASHRAE 2004).

Based on the required breathing zone outdoor airflow and the zone air distribution effectiveness outdoor airflow (V_{oz}) could be determined for the design zone. This is the outdoor airflow that must be provided to the zone by the supply air distribution system.

$$V_{oz} = V_{bz} / E_z \,. \tag{4.2}$$

According to the building simulation assumptions and the construction drawings, the building total area is 174,016 ft^2 and the occupancy is 865 persons. Based on ASHRAE 62.1-2004 (ASHRAE 2004), the outdoor airflow rate required per person for the office zone (R_p) is 5 cfm/person and the outdoor airflow rate required per unit area

(R_a) is 0.06 cfm/ft^2. Since Wisenbaker Engineering Research Center has a ceiling supply of cool air distribution configuration, the zone air distribution effectiveness is 1.0.

Based on this data, the required OA flow is:

$$V_{bz} = 5\frac{cfm}{person}\cdot 865\,person + 0.06\frac{cfm}{ft^2}\cdot 174{,}016\,ft^2 = 14{,}766cfm \tag{4.3}$$

The required outdoor airflow provided to WERC by the supply air distribution system is:

$$V_{oz} = \frac{V_{bz}}{E_z} = \frac{14{,}766cfm}{1.00} = 14{,}766cfm \tag{4.4}$$

Based on ASHRAE 62.1-2004, the outside air flow could be potentially reduced from 25,747 cfm (0.148 cfm/ft^2) to 14,766 cfm (0.085 cfm/ft^2) or 40 %. This measure will provide 506 MMBtu (2 %) of cooling energy savings (Table 4.3).

4.2.3. High performance measure 3: AHU-8 (Material Testing Laboratory) operation schedule improvement

The AHU-8 that serves MTL operates 24 hours 7 days a week. Since the laboratory is not occupied from 8 pm to 8 am the AHU-8 that operates in constant speed could be turned off during the night. This measure could be implemented by turning off the AHU from 8pm to 7am and using optimum start from 7 am to 8 am. Operation during weekends simulated with the same schedule since zone occupancy for weekends could not be accurately predicted.

Annual major energy saving potential from turning off AHU-8 is 722 MMBtu of HHW energy and 667 MMBtu of ChW energy.

4.2.4. High performance measure 4: Lighting system improvement

The most common type of light bulbs used in WERC is the fluorescent bulb F32T8 with a consumed power of 32 Watts. Estimation of lighting level in the building and possible reduction to the level recommended for the office area by IEC (Shapiro 2009) (30-70 footcandles / 323-753 lux) and change of all ballast to electronic instant start ballast can reduce energy consumption by lighting system from 1.2 W/ft^2 to 0.90 W/ft^2 (Liebel and Brodrick 2005). Installation of automatic occupancy control based on occupancy sensors can reduce lighting power density additionally by 10% to 0.80 W/ft^2 (ASHRAE 2007). Estimated major energy savings from this measure is 459 MWh of electrical energy.

4.2.5. High performance measure 5: Economizer mode

Based on the building HVAC system construction drawings and the AHU walk through data, the economizer mode was not implemented during design of the HVAC system. The variable air volume AHUs simulated by the SYST-1 are equipped with both supply and return air fans. The supply fans maximum nameplate total flow for AHU 1-7 (SYST-1) is 147,000 cfm while the simulated flow is 74,615 cfm and the energy consumption is 96.31 hp. The supply air fans during the normal operation are loaded at not more than 50%. The return air fans for the SYST-1 have a total nameplate power of 80 hp. Existing AHUs 1-7 (SYST-1) are designed to provide a total outside flow of 19,000 cfm according to construction schedules, but the ducts for the supply, return and exhaust air have approximately the same diameter. These parameters allow the economizer mode to be used in case, if the direct digital control (DDC) system is implemented.

The proposed enthalpy economizer operates between 30ºF and 60ºF, and the enthalpy economizer returns the OA damper to a minimum position, if the OA enthalpy is higher than the return air enthalpy. If the outside air enthalpy is lower than the return air enthalpy, the economizer control will maintain the supply air temperature by regulating the OA flow.

The possible annual energy savings that the enthalpy economizer mode can provide was estimated to be 2,782 MMBtu (11.2%) of ChW energy, which simultaneously increases the HHW energy consumption by 717 MMBtu of (Table 4.3).

4.2.6. High performance measure 6: AHU 1-7 operation schedule improvement

When implemented with the DDC system, the AHU 1-7 (SYST-1) could be set to off during the unoccupied period of time, instead of operating 24/7. The building has a low occupancy from 8 pm till 8 am, and for this period of time, the HVAC system could be set to off with forced cycling when the temperature of any zone increases above the cooling set point or decreases below the heating setpoint. Optimum start could be used from 7am to 8 am. The annual energy saving potential from turning off the AHU 1-7 during the unoccupied mode is estimated to be 4,490 MMBtu (18%) of ChW energy and 4,887 MMBtu (39.9%) of HHW energy. Some saving is also achieved in supply and return air fans energy consumption, 153 MWh (4.9%).

4.3. Summary of potential energy saving retrofits

Potential energy savings from the combined proposed improvements in WERC were simulated to be 12,285 MMBtu (49 %) of ChW, 7,715.7 MMBtu (62.9 %) of HHW and 917.9 MWh (29.6%) of total building electrical energy consumption during the year, based on simulated 2008 annual energy consumption data from DOE-2.1E reports (Table 4.4). Implementation of the proposed measures can reduce site energy consumption to 107.9 $\mathrm{kBtu/ft^2yr}$ (19.4 $\mathrm{kWh/ft^2yr}$ of electrical energy and 0.416 $\mathrm{term/ft^2yr}$ of natural gas).

Table 4.4. Wisenbaker Engineering Research Center high performance measures energy savings potential

Category	Base Model	Improved model energy savings potential						
		1	2	3	4	5	6	Total
Area Light (MWh)	1402.439	0	0	0	458.895	0	0	458.895
Office Equipment (MWh)	934.952	0	0	0	0	0	0	0
Ventilation fans (MWh)	760.48	422.726	0	17.083	0	0	153.261	458.999
Chilled water (MMBtu)	24918.5	9328.5	505.9	667	228.9	2782.4	4490.5	12285.3
Heating hot water (MMBtu)	12262.4	6724.4	-111	722.5	-1128.3	-717.5	4886.8	7715.7
Total Electricity (MWh)	3097.9	422.8	0	17.1	458.9	0	153.3	917.93
Total site energy kBtu/ft^2-yr	200.4	69.4	2.1	6	0.6	-3.1	42.7	92.5

Current utilities rates for buildings located on Texas A&M campus in College Station, TX (Texas A&M Utilities Rate 2010), valid from September 1, 2009 till August 31, 2010, provide the price of electrical energy at \$0.113/kWh, for ChW at \$14.582/MMBtu and \$18.147/MMBtu for HHW. Based on the utility rates, the annual energy savings potential is estimated to be \$423,000 (Figure 4.3). The main part of the energy savings potential is provided by measure 1 (minimum supply airflow adjustment of VAV boxes) and measure 6 (AHU 1-7 operation schedule improvement).

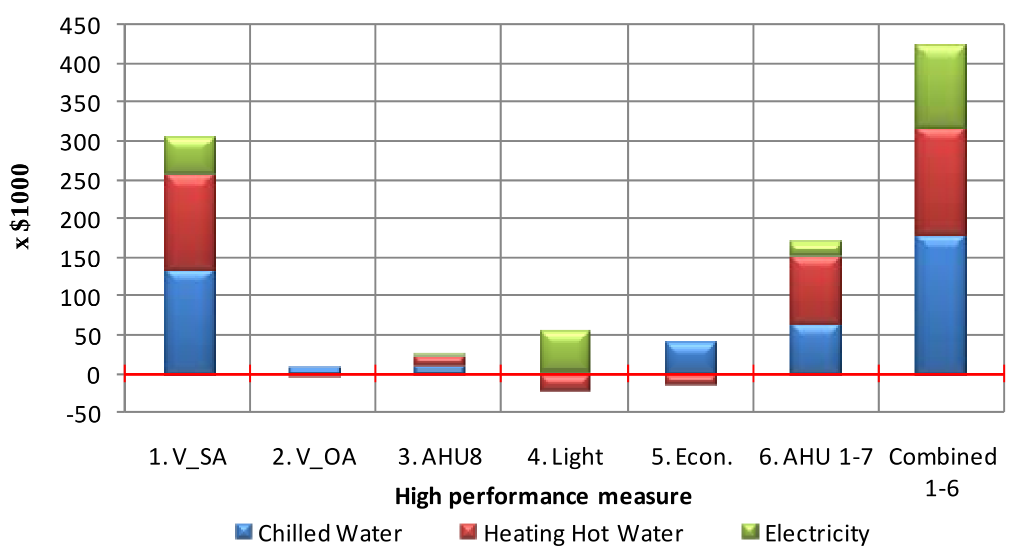


Figure 4.3. Economical estimation of proposed high performance measures for Wisenbaker Engineering Research Center

Implementation of these measures required the replacement of 177 VAV boxes and the installation of the DDC system for seven AHUs. For estimating the modification cost, RSMeans Mechanical and Electrical Cost Data were used (RSMeans Electrical cost data 2007; RSMeans Mechanical Cost Data 2007).

The installation price for the new VAV terminal boxes is estimated as $118,504 based on the RSMeans 233616.10 classification category (Table 4.5). The national average cost data were corrected by city cost index for Bryan, TX. Overhead and profit were estimated as a 10% above total cost. The present value of the VAV terminal boxes installation cost is estimated to be $122,000 by using GDP deflator (Williamson 2010).

Average U.S. bare cost of existing VAV terminal box demolition was $35, and the cost with overhead and profit was $53.50. With correction for labor city cost index, this price will be $33.7 in 2007 U.S. dollars per box. Estimated cost for demolition of all VAV boxes is $4,320, including overhead and profit. With correction for the GDP deflator index, this value will increase to $4,500.

Table 4.5. Estimated cost of variable volume mixing boxes installation (RSMeans Mechanical Cost Data 2007)

Type	Nominal flow, cfm	Reheat	Amount on the floor				Bare national average cost 2007, $		City cost index, %		Local category average cost 2007, $	
			First	Second	Third	Total	Material	Labor	Material	Labor	Total cost	Total O&P
A*	600	n	2	9	10	21	335.0	57.0	96.6	63.4	7,554	8,310
B*	800	n	6	17	29	52	345.0	69.5	96.6	63.4	19,621	21,583
C*	1200	n	1	6	2	9	360.0	69.5	96.6	63.4	3,526	3,879
D*	2800	n	1	5	5	11	400.0	89.5	96.6	63.4	4,874	5,362
E*	3000	n	1	0	0	1	410.0	105.0	96.6	63.4	462	508
AH**	500	y	0	4	4	8	640.0	62.5	96.6	63.4	5,262	5,789
BH**	800	y	5	1	0	6	660.0	78.5	96.6	63.4	4,123	4,536
CH**	1200	y	5	4	1	10	720.0	105.0	96.6	63.4	7,620	8,382
DH**	1800	y	3	0	0	3	785.0	157.0	96.6	63.4	2,573	2,830
EH**	3000	y	30	11	15	56	855.0	165.0	96.6	63.4	52,110	57,321
Total						177					107,731	118,504

* Variable volume mixing boxes includes electric or pneumatic motor, pressure independent.

** Electric with thermostat, pressure dependent, VAV cool and heating hot water coils, damper, actuator and thermostat.

The price for DDC implementation was estimated to be $33,314 based on a 2007 money value. The economical estimation is based on U.S. national averages provided by RSMeans with corrections for City Cost index (RSMeans Electrical cost data 2007). Assuming that existing sensors could be utilized in DDC system, the required cost is summarized in Table 4.6. Prices include software programming and checkout. The adjusted money value for 2010 year is $34,400 using the GDP deflator (Williamson 2010).

Table 4.6. Estimated cost of DDC system installation for 7 VAV AHU

#	Type	#	Bare national average cost 2007, $				City cost index, %			Local category average cost 2007, $	
			Mater.	Labor	Total	Total O&P	Mater.	Labor	Total	Total	Total O&P
1	Mechanical room DDC controller 16 point controller (incl. 120V/1 hp supply)	7			1,938	3,121	96.6	63.4	82.3	11,161	17,981
2	Basic maintains manager software	1			1,702	1,873	96.6	63.4	82.3	1,400	1,541
3	Time program (point)	7			6	7	96.6	63.4	82.3	34	37
4	Optimal Start/stop	7			36	40	96.6	63.4	82.3	207	227
5	Enthalpy program	7			36	40	96.6	63.4	82.3	207	227
6	Variable volume modulating motorized damper, incl. elect. mtr. 48"x48"	14	825	105	930	1,075	96.6	63.4	82.3	12,089	13,298
	Total									25,100	33,313

All proposed high performance retrofit costs, except lighting system improvements (Measure 4) are described in (Table 4.5, Table 4.6). The cost of lighting system improvements requires more detailed information that is not available for the current research and thus left for future investigation. Energy savings potential for all proposed measures, except lighting system improvements, is estimated to be 11,326 MMBtu for ChW, 8175 MMBtu for HHW and 436 MWh for total building electrical energy consumption during the year, based on the simulated 2008 annual energy consumption data from DOE-2.1E report. Annual building site energy consumption is 13,371.8 MMBtu for electrical energy and 6536 MMBtu for natural gas (Figure 4.4).

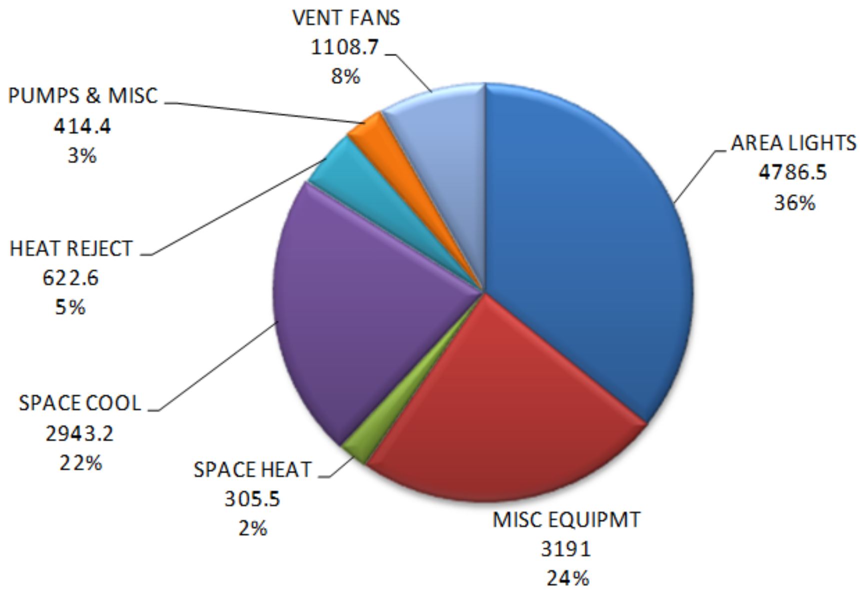


Figure 4.4. Wisenbaker Engineering Research Center improved model annual electrical energy consumption summary in MMBtu

Based on the current energy price, the equivalent cost for saved energy or net average annual revenue is \$362,700. Net initial investment is a sum of the estimated cost of the VAV mixing boxes demolition and the installation of new VAV boxes and DDC system installation that is equal to \$160,900.

Simple pay-back for measures 1-3, 5, 6 can be estimated according to formula:

$$\text{Payback} = \frac{\text{Net Initial Investment}}{\text{Net average Annual Revenue}} = \frac{\$160{,}900}{\$\,362{,}700/\text{ year}} = 0.45 \text{ year} \qquad (4.5)$$

Obtained result is based on a new construction price for construction projects with a budget higher than \$1,000,000 (RSMeans Mechanical Cost Data 2007). This assumption can cause increase of initial investment and simple pay-back period. Based on a simple pay-back period, proposed measures 1-3, 5, 6 with an annual energy saving potential of \$362,700 could be recommended.

CHAPTER V

VALLEY PARK OFFICE BUILDING SIMULATION

This chapter describes the simulation and analysis of possible improvements in HVAC systems for a new office building as compared with a conventional system. As a new construction project, Valley Park office building located on Harvey Mitchell Parkway in College Station, Texas, was chosen as a baseline. A standard reference design was developed based on construction drawings of this new building, measures of energy consumption by ESL office equipment and lighting system as well as current energy conservation codes. Improvements were proposed for HVAC systems without changes in the building envelop which is fixed on minimum code compliant level.

5.1. Energy standards and codes requirements

Energy standards and codes play a key role in setting the design goals and developing energy baselines for new construction projects. To define the new building baseline model, two standards were considered: a federal energy standard, ASHRAE Standard 90.1-2007 (ASHRAE 2007) and an international standard, International Energy Conservation Code (IECC 2009).

ASHRAE Standard 90.1 was first released in 1975 and revised in 1980, 1989, 1999, 2001, 2004, and 2007. This standard provides the minimum requirements for the design of energy efficient buildings, except for low-rise residential buildings. ASHRAE Standard 90.1 is updated every three years and provides mandatory requirements for building components, including: building envelopes, lighting systems, HVAC systems, and other equipment.

The International Energy Conservation Code (IECC) was first published in 1995 to replace the Model Energy Code (MEC) by the International Code Council. The IECC is also updated in three-year intervals. The last edition was published in 2009. The IECC 2009 contains prescriptive and performance-based methods for both residential and commercial buildings. Chapter 5 of IECC 2009 provide the minimum efficiency requirements for the commercial building envelope, mechanical systems, service water heating, and lighting systems. The general guidelines are also provided to determine total building performance.

According to both standards College Station, Brazos County, TX located in climate zone 2A of the building envelope climate criteria (hot and humid).

For climate zone 2A, the nonresidential building envelope requirements are the same for both standards and are summarized in Table 5.1 and Table 5.2. For buildings with a gross area above 25,000 ft^2, both standards require a temperature control dead band of at least 5°F. According to the standard, an air economizer is not required in a hot and humid climate. In case a fixed enthalpy economizer is used, it should be shut off when the outdoor air enthalpy exceeds 28 Btu/lb or for differential enthalpy control when the outdoor air enthalpy is above the return air enthalpy.

Table 5.1. Nonresidential building opaque envelope element minimum requirements for climate zone 2A by ASHRAE Standard 90.1-2007 and IECC 2009

Opaque Element	Assembly Maximum U value, Btu/Hr-ft^2-°F	Insulation Min. R-Value
Roofs insulation entirely above deck	U-0.048	R-20c.i.
Walls, above grade, mass	U-0.151	R-5.7 c.i.
Floors, mass	U-0.107	R-6.3 c.i.
Slab on grade, unheated	F-0.730	NR
Opaque doors, swinging	U-0.700	
Opaque doors, nonswinging	U-1.450	

c.i. – continuous insulation
NR- no requirements

Table 5.2. Nonresidential building minimum fenestration requirements for climate zone 2A by ASHRAE Standard 90.1-2007 and IECC 2009

Fenestration	Assembly Max. U, Btu/Hr-ft2-°F	Assembly Max. SHGC
Vertical glazing, metal framing (windows) (ASHRAE 2007)	U-0.75	0.25
Vertical fenestration (IECC 2009)	U-0.75	0.33 for ($0.25 \leq PF \leq 0.5$)

For an electronically operated unitary air conditioning system with air-cooled air conditioners with sizes between 240 and 760 kBtu/hr and electric resistance, the heating section minimum efficiency requirement is 10.0 EER. For water-cooled air conditioners with sizes above 240 kBtu/hr and electric resistance heating, 11.5 EER is required. According to the building area method, the interior lighting power allowance for the office area is 1.0 W/ft^2 in both standards.

According to the standard, the supply air-to-room air temperature difference should be 20ºF. The exhaust air energy recovery system should not be included in the baseline building design since minimum outdoor air supply is less than 70% of the design supply air capacity (ASHRAE 2007). The thermostat setpoint shall be 70ºF for heating and 75ºF for cooling with a night setback temperature setpoint of 55ºF for heating and 99ºF for cooling. For the envelope design comparison, buildings should be modeled with the same HVAC system.

According to ASHRAE (2007) DOE-2 is recommended as a simulation program. Another program, Windows Air Model (Win AM 4.0) is not applicable for building model simulations since it does not have the ability to explicitly model thermal mass effects (ASHRAE 2007).

The building design project codes compliance can be verified using the simplified computer program COMcheck. COMcheck 3.8.0 provides compliance verification with IECC 2000, 2001, 2003, 2004, 2006, 2009 and ASHRAE/IES Standard 90.1: 2001, 2004, 2007, including various state-developed energy codes (COMcheck. Version 3.8.0 2010). The simplicity of the program provides a big advantage in application since it requires minimum data input, but at the same time limits the types and complexity of building HVAC systems that can be modeled.

5.2. The baselines building envelope design

A buildings baseline design plays an important role in measuring energy savings potential for a new buildings, as well as possible energy retrofits in existing buildings. Standard reference building design was developed to comply with IECC 2009 as an active energy code in the State of Texas with reference to a federal energy code ASHRAE Standard 90.1-2007.

For Valley Park office building simulation, the same geographical location is used as was used for WERC. The building construction process is going to be finished in 2011. The building has one story above the ground. The building azimuth is 45° east of north. The net area of the building is 25,774 ft^2. The building construction is of medium weight mass with a concrete floor and walls, and it is designed as an office building. The architectural drawings of the building were used for a DOE-2.1E building simulation. Electricity is the only one available energy source at the building site.

Table 5.3 shows detailed information about construction material for the Valley Park office building. The properties of the construction material was estimated from the DOE-2.1 material library (LBNL 1993a) corresponding to the actual materials in the building design and energy code compliance. The inside film resistance was defined by using the default value of 0.68 for all vertical wall surfaces. For a roof with upward heat flow, an inside-fi1m-resistance for air of 0.61 Hr-ft^2-°F/Btu was used (LASL 1980).

Table 5.3. Valley Park office building construction material thermal properties

Items	U-NAME		Description	Thermal Properties				
	Construction	Materials		Thickness	Conductivity	Density	Specific heat	Resistance
				ft	Btu-ft/Hr-ft^2-°F	Lb/ft^3	Btu/Lb-°F	Hr-ft^2-°F/Btu
Roof	ROOF-1	RG02	1 inch Roof Gravel or Slag	0.0833	0.834	55	0.4	0.1
		CB56	12 inch Light Weight Hollow Concrete	1	0.4405	49	0.2	2.27
		IN35	2 inch expanded polystyrene	0.1667	0.02	1.8	0.29	8.33
		IN35	2 inch expanded polystyrene	0.1667	0.02	1.8	0.29	8.33
		WD11	3/4 inch hard wood	0.0625	0.0916	45	0.3	0.68
Exterior wall	WALL-EXT	IN33	1 inch polystyrene expanded	0.0833	0.02	1.8	0.29	4.16
		CB27	6 inch medium weight concrete filled concrete block	0.5	0.4443	119	0.2	1.13
		CM02	1.75 inch mortar cement	0.1458	0.4167	116	0.2	0.35

Table 5.3. (Continued)

Items	U-NAME		Description	Thermal Properties				
	Construction	Materials		Thickness	Conductivity	Density	Specific heat	Resistance
				ft	Btu-ft/Hr-ft2-°F	Lb/ft^3	Btu/Lb-°F	Hr-ft^2-°F/Btu
Interior wall	WALL-INT	GP04	3/4 inch Light Weight Aggregate Gypsum Plaster	0.0625	0.133	45	0.2	0.47
		BK01	4 inch Common Brick	0.3333	0.4167	120	0.2	0.8
		GP04	3/4 inch Light Weight Aggregate Gypsum Plaster	0.0625	0.133	45	0.2	0.47
Slab-on-grade: unheated	WALL-U	GP05	1 inch Light Weight Aggregate Gypsum Plaster	0.0833	0.133	45	0.2	0.63
		IN42	3/4 inch Expanded Polyurethane	0.0625	0.0133	1.5	0.38	4.67
		CB56	12 inch Light Weight Hollow Concrete Block	1	0.4405	49	0.2	2.27

All building opaque envelope elements parameters are compliant with IECC 2009 (Table 5.4).

Table 5.4. Valley Park office building baseline model opaque envelope elements code compliance validation

Opaque Element	Assembly Maximum U value	Simulated U value
	Btu/Hr-ft^2-°F	Btu/Hr-ft^2-°F
Roofs insulation entirely above deck	U-0.048	0.048
Walls, above grade, mass	U-0.151	0.149
Floors, mass	U-0.107	-
Slab on grade, unheated	F-0.730	0.032
Opaque doors, swinging*	U-0.700	-
Opaque doors, nonswinging*	U-1.450	-

The building has windows with a single pane glazing and reflective coating. The building fenestrations were simulated with a glass conductance of 0.879 Btu/Hr-ft^2-°F and a fenestration total conductance of 0.750 Btu/Hr-ft^2-°F. The IECC 2009 requirements for a solar heat gain coefficient (SHGC) depend upon the value of the projection factor.

The projection factor is the ratio of the horizontal depth of the external shading projection divided by the sum of the height of the fenestration and the distance from the top of the fenestration to the bottom of the farthest point of the external shading projection, in consistent units.

The projection factor can be calculated by the formula:

$$\mathrm{PF} = \mathrm{A} / B \,, \tag{5.1}$$

where: A - distance measured horizontally from the furthest continuous extremity of any overhang, eave, or permanently attached shading device to the vertical surface of the glazing;

B - distance measured vertically from the bottom of the glazing to the underside of the overhang, eave, or permanently attached shading device.

$$\mathrm{PF} = \frac{A}{B} = \frac{4\,ft}{10.2\,ft} = 0.392\,. \tag{5.2}$$

The buildings vertical glazing has a total surface of 2,886 ft^2 or 4.5% of a total wall area. According to (IECC 2009), the SHGC should be below 0.33 for a projection factor between 0.25 and 0.5.

DOE-2.1E uses the Shading Coefficient (SC) as an input value. According to ASHRAE Standard 90.1-2007 section 5.8.2.5, the SC could be obtained from the solar heat gain coefficient by dividing it by 0.86. The simulated value of SC for the windows is 0.384.

The entrance doors are constructed from glass with a reflective coating and a metal frame. For the door glass simulation, glass conductance of 1.450 Btu/Hr-ft^2-°F and SC 0.384 were used.

Space zoning of the Valley Park office building was defined as four inner and four perimeter zones. Figure 5.1 shows a plan view and space zoning used for the simulation of the building.

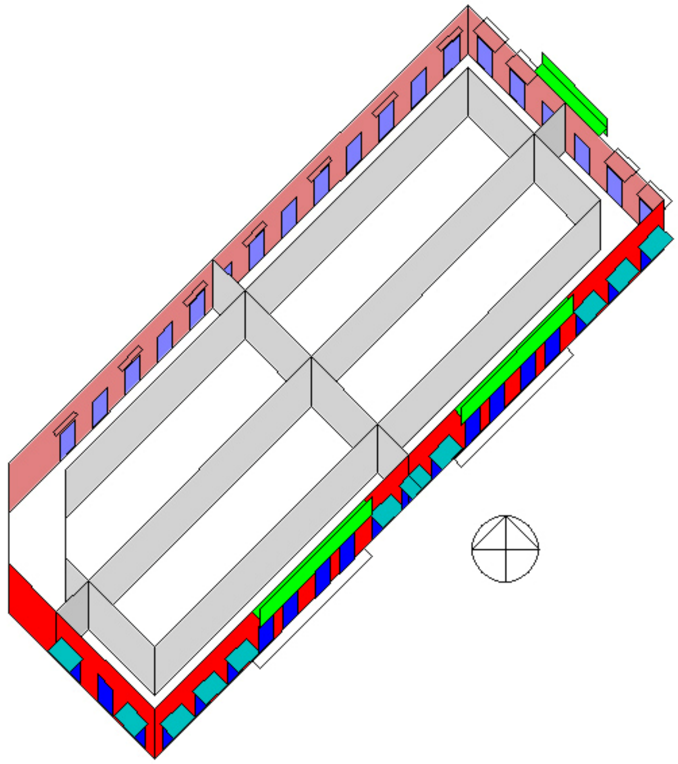


Figure 5.1. Valley Park office building floor plan

Table 5.5 specifies the Valley Park office building space conditions. The building occupancy and people density were estimated based on ESL office and construction project design assumptions. The maximum building office area occupancy is estimated at 100 ft^2/person, according to the building design documentation. It is assumed that the maximum occupancy occurs between 8:00am and 11:00am in the morning and from 3:00pm to 6:00pm in the afternoon. For the weekends, it is assumed that the occupancy is 10% from 9:00 am till 5:00 pm and 5% the rest of the occupied time. The building is assumed to be unoccupied from 1:00 am to 6:00 am over the year.

Table 5.5. Space conditions for the Valley Park office building

DOE-2 Key word	Units	Value	Description
PEOPLE-SCHEDULE	-	OCCUPY-1	Based on average ESL office area occupancy schedule
AREA/PERSON	ft^2/ person	100	Based on the building design assumption
PEOPLE-HG-LAT	Btu/hr-person	200	Moderately active office work (ASHRAE 2009)
PEOPLE-HG-SENS	Btu/hr-person	250	Moderately active office work(ASHRAE 2009)
LIGHTING-SCHEDULE	-	LITEQP-1	Based on ESL office area average energy consumption
LIGHTING-TYPE	-	REC-FLUOR-RV	Recessed fluorescent RV -vented to return air
LIGHT-TO-SPACE		0.80	The fraction, if any, of the lighting energy that is added to the space energy balance as a sensible heat gain. Based on observation
LIGHTING-W/SQFT	W/ft^2	0.9	Based on IECC 2009
EQUIP-SCHEDULE	-	LITEQP-1	Based on the WERC whole-building energy consumption profile
EQUIPMENT-W/SQFT	W/ft^2	0.8	The maximum equipment energy use. Based on the measured value in ESL office area
INF-METHOD	-	AIR-CHANGE	
AIR-CHANGES/HR	-		Assuming building pressurization
FLOOR-WEIGHT	Lb/ft^2	0 (Auto)	The composite weight of the floor, furnishings and interior walls of a space divided by the floor area of the space. Automatic calculated based on construction materials

Lighting and equipment schedules were determined based on ESL average energy consumption for weekdays, Saturdays, Sundays and holidays. The maximum electrical density of office equipment is estimated to be 0.8 W/ft^2 according to the ESL office area measured value. The occupancy schedule was developed based on both observations and ESL typical lighting schedules. Lighting equipment density was assumed to be 1.0 W/ft^2 with 20% of the energy rejected to the return air stream according to IECC 2009. According to ASHRAE Standard 90.1-2007, automatic lighting controls that utilize occupancy sensors for buildings with areas more than 5,000 ft^2 reduce power density by 10%. Automated lighting shutoff is required by IEEC 2009 and could be provided by occupancy sensors that turn off lighting after 30 min after the occupant leaves the space. This reduces the lighting equipment power density to 0.9 W/ft^2.

No infiltration was assumed in the DOE-2 simulation for the office building since the HVAC is always on and the building is assumed to be pressurized. The building construction weight was estimated by custom-weighting factors based on simulated building construction materials.

5.3. The baseline HVAC system design

As a baseline design HVAC system a packaged variable volume, variable temperature system (PVVT) was chosen. This is a hybrid system that cools by the direct expansion of a refrigerant and may optionally heat with gas, HHW or an electric resistance heater (Figure 5.2). Optional components of the systems are shown in the dashed boxes.

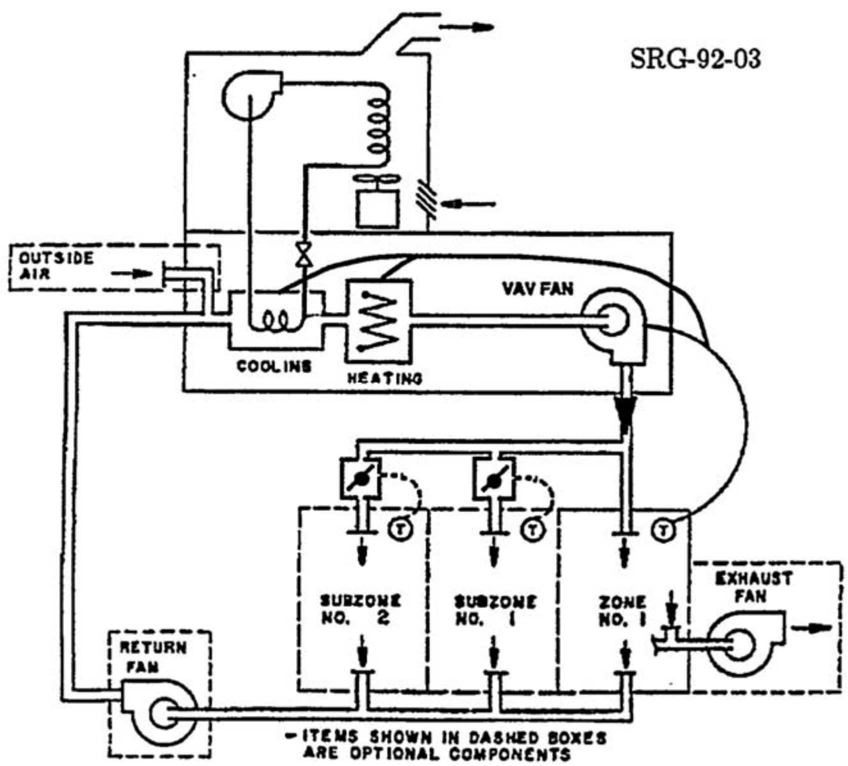


Figure 5.2. Packaged variable temperature, variable volume system (LBNL 1993b)

This unit is normally considered a commercial unit that provides variable volume air to the zone. This forced-air packaged unit may be either a unitary system (rooftop unit or outside-the-wall unit) or it may be a split unit (partially inside and partially outside) (LASL 1980; LBNL 1993b). It may or may not require ducting. In a basic configuration this system includes a refrigeration compressor, an air cooled condenser with a fan discharging heat to the outdoors, an evaporator with a fan supplying cooled air to the indoors, a filter (not shown), and a thermostat. The system does not have zone reheat, which makes it different from a packaged single zone air conditioner with heating and subzone reheating options (PSZ). This system adjusts the volume flow into the zones to match the heating or cooling requirements of the control zone. The system is intended to be a single zone system that could serve multiple zones operating in the same mode (heating or cooling). This system has a variable speed compressor and a variable volume supply air fan (LBNL 1993b). The system can provide outside ventilation air to the zones and economizer cooling. Supply (evaporator) fans can be specified as either cycling or a continuously running. It may be either a blow through or a draw through fan, with the fan motor either inside or outside the airstream. The condenser fan operates automatically on demand (LBNL 1993b). An exhaust air fan and a return air fan or a return-exhaust air fan may be optionally specified. The thermostat may be specified with night setback and night cycle control.

The design thermostat setpoint is 70ºF for heating and 75ºF for cooling with a night setback temperature setpoint of 55ºF for heating and 99ºF for cooling. Night setback is used between 1:00am and 6:00am all year. The exhaust air energy recovery system is not included in the baseline building design since the minimum outdoor air supply is less than 70% of the design supply air capacity (ASHRAE 2007). For a baseline, the design air economizer is not used since it’s not required in a hot and humid climate.

The estimated amount of building cooling load is 300 kBtu/hr. The air cooling of the condenser and resistance heating were used as a baseline design. According to this assumptions electronically operated unitary air conditioning systems with air-cooled evaporator and electric resistance heating sections require a baseline cooling efficiency of 10.0 EER, as of Jan, 1 2010 (0.3412 EIR).

According to ASHRAE 62.1-2004, the required design building outdoor airflow is:

$$V_{bz} = 5\frac{cfm}{person}\cdot 258\,person + 0.06\frac{cfm}{ft^2}\cdot 25{,}774\,ft^2 = 2{,}836cfm \qquad (5.1)$$

The outdoor airflow is distributed equally between all conditioned zones. Valley Park office building total supply airflow varies between 1 and 0.15 cfm/ft^2.

Simulated supply air fans have variable speed, and hourly power consumption per unit of supply air moved at 0.000685 kW/cfm, which is below the IEEC 2009 requirement of (0.001119 kW/cfm) for a variable volume system. The air temperature rise along the supply fan is assumed to be 2.117ºF based on DOE-2.1E default values. The supply air fans are positioned in the airflow and could be loaded between 30 and 110% of nominal load (Table 5.6).

Table 5.6. Valley Park office building HVAC system baseline design

Command	DOE-2.1 Keyword	Set-point for office zone	Description
SPACE-CONDITIONS	-	OFFICE	
ZONE-AIR	CFM/SQFT	1 cfm/ft^2	Minimum air flow rate to the zone per ft^2 of zone area
	OUTSIDE-AIR-CFM	Assigned for each zone. Total 2836 cfm	The minimum flow rate of outside air (0.11 cfm/ft^2)
ZONE-CONTROL	DESIGN-HEAT-T	70ºF	The space temperature that the program uses to calculate the supply air flow rate required to meet peak heating loads for the zone
	DESIGN-COOL-T	75ºF	The space temperature that the program uses to calculate the supply air flow rate required to meet peak cooling loads for the zone
	HEAT-TEMP-SCH	HEAT-SCHED	70ºF with night reset 55ºF
	COOL-TEMP-SCH	COOL-SCHED	75ºF with night reset 99ºF
	THERMOSTAT-TYPE	REVERSE-ACTION	VAV System
ZONE	SIZING-OPTION	ADJUST-LOADS	The size of the supply air fan is determined by the block peak load
	ZONE-TYPE	CONDITIONED	The zone is heated and/or cooled, depending on the type of system selected.

Table 5.6. (Continued)

Command	DOE-2.1 Keyword	Set-point for office zone	Description
SYSTEM-CONTROL	COOLING-SCHEDULE	COOLOFF	Cooling is always available
	HEATING-SCHEDULE	HEATOFF	Heating is always available
	MAX-SUPPLY-T	90 ºF	DOE-2 Default. The highest allowable temperature f or air supplied to the space
	MIN-SUPPLY-T	55 ºF	DOE-2 Default. The lowest allowable temperature f or air supplied to the space
SYSTEM-FANS	FAN-SCHEDULE	FAN-SCHED	Fans are always on
	FAN-CONTROL	SPEED	
	SUPPLY-DELTA-T	2.117ºF	Temperature rise in the air stream across the supply fan.
	SUPPLY-KW	0.000685 kW/cfm	The hourly power consumption o f the supply fan per unit of supply air moved
	MOTOR-PLACEMENT	IN-AIRFLOW	
	MAX-FAN-RATIO	1.1	
	MIN-FAN-RATIO	0.3	
SYSTEM-TERMINAL	MIN-CFM-RATIO	0.15	System simulated in constant air flow operation according to AHU measured values.
SYSTEM EQUIPMENT	COMPRESSOR-TYPE	VARIABLE-SPEED	
	CONDENSER-TYPE	AIR-COOLED	
	DEFROST-TYPE	RESISTIVE	
	DEFROST-CTRL	TIMED	
	COOLING-EIR	0.3412	Corresponding 10.0 EER
	COIL-BF	0.19	The coil bypass factor
SYSTEM	SYSTEM-TYPE	PVVT	Packaged single zone fan system - variable volume - variable temperature
	ECONO-LIMIT-T	60 ºF	Economizer mode is not available for the system
	ECONO-LOW-LIMIT	60 ºF	Economizer mode is not available for the system
	HEAT-SOURCE	ELECTRIC	The heat source for the central heating coils
	RETURN-AIR-PATH	DUCT	Route that return air takes in getting back to the air handling unit

The HVAC system will operate during the night, if the temperature in any zone of the served area falls below throttling range for heating or above throttling range for cooling and the fans are cycled on for that hour. For the baseline design, 19% of the air bypasses the cooling coil and does not come in close enough contact with the coil to be dehumidified. The baseline design return air path is direct, or via a duct to the supply fan with added heat from lights vented to the return air. DOE-2.1E input program code for Valley Park office building baseline design energy consumption model is given in Appendix B.

5.4. The baseline design building energy performance

The baseline design of the Valley Park office building was performed by using Typical Meteorological Year version 3 (TMY3) weather data for College Station Easterwood Airport, TX (U.S. DOE 2010d).

According to the simulation results, the building requires cooling during all of the temperature range, and heating when outside ambient temperature is below 65ºF (Figure 5.3).

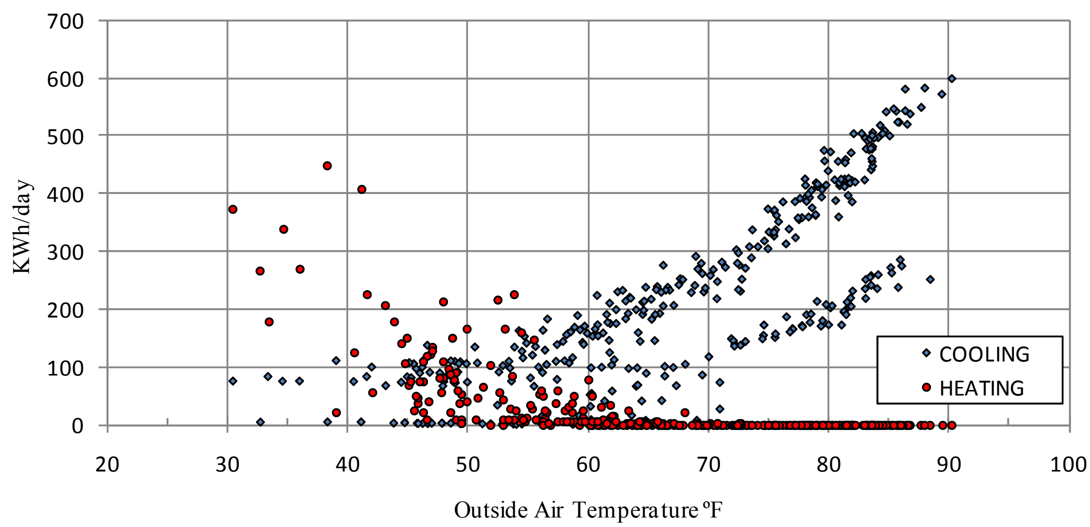


Figure 5.3. Valley Park office building baseline model heating and cooling system electrical energy consumption in kWh/day

The main energy consumption categories are cooling (36%), lighting (29%) and office equipment (26%) (Figure 5.4).

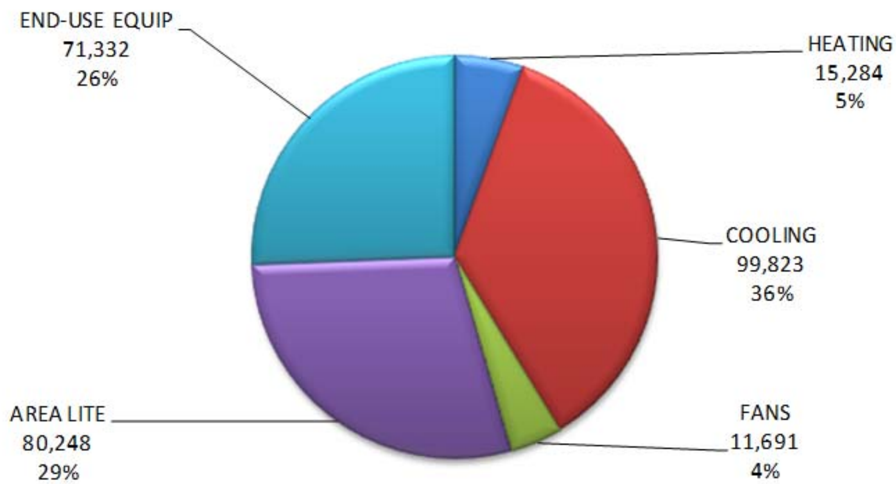


Figure 5.4. Annual Valley Park office building baseline model energy consumption summary in kWh/year

The maximum simulated cooling load of 301 kBtu/hr appears on the 3rd of August 2010 and the maximum heating load of 123.4 kBtu/hr appears on the 7th of January (Table 5.7; Table 5.8).

Table 5.7. Valley Park office building monthly cooling loads summary

Month	Cooling energy	Time of maximum load		Dry-bulb temperature	Wet-bulb temperature	Maximum cooling load
	MMBtu	day	hr	F	F	kBtu/hr
JAN	24.5	14	16	76	64	153.5
FEB	36.7	10	16	81	68	191.0
MAR	45.7	31	16	80	70	201.1
APR	51.2	21	16	88	74	217.7
MAY	94.3	11	15	90	72	248.9
JUN	105.2	2	16	94	81	267.5
JUL	126.5	27	16	99	77	282.1
AUG	125.3	3	16	106	76	301.1
SEP	95.1	10	16	85	77	249.3
OCT	65.2	6	15	91	74	248.6
NOV	32.5	2	16	80	71	194.5
DEC	28.7	2	16	68	57	146.4

Table 5.8. Valley Park office building monthly heating loads summary

	Heating energy	Time of maximum load		Dry-bulb temperature	Wet-bulb temperature	Maximum heating load
Month	MMBtu	day	hr	F	temp	kBtu/hr
JAN	23.592	7	7	24	22	123.365
FEB	6.895	1	6	43	42	103.637
MAR	5.499	17	7	37	35	68.328
APR	2.602	13	6	39	35	56.165
MAY	0.011	19	6	51	47	5.006
JUN	0	-	-	-	-	0
JUL	0	-	-	-	-	0
AUG	0	-	-	-	-	0
SEP	0	-	-	-	-	0
OCT	0.671	15	6	48	47	31.241
NOV	14.26	19	6	40	36	81.395
DEC	17.265	14	7	30	30	83.151

Total site energy consumption is 950.1 MMBtu or 36.9 $\text{kBtu/ft}^2\text{ - yr}$ and total source energy is 2,851 MMBtu. The total annual baseline building energy consumption is 278,377 kWh.

5.5. Decoupled HVAC system simulation

According to the purpose of this study, selected opportunities to improve HVAC systems in a hot and humid climate were simulated based on literature review conclusions. According to the literature review findings, a decoupled system concept that includes splitting of the HVAC system into two components: a DOAS and an air conditioning system, is one of the most efficient measures. A decoupled system can increase humidity control and decrease energy consumption by eliminating supply air reheat requirements and excessive cooling. A decoupled system concept was simulated based on Valley Park office building baseline design by using two direct expansion cooling system operating in parallel.

According to ASHRAE (2007), if the component or the system included in the proposed design cannot be modeled explicitly, it should be substituted by a thermodynamically similar component model that can approximate the expected performance of the component. To perform the simulation of the decoupled system concept in DOE-2.1E two component models were simulated.

The first component that simulates the DOAS is based on an almost adiabatic building envelope model with no fenestration or internal heat gain (lighting and equipment load). The dedicated outdoor air system operates in a constant volume mode and supplies $0.11\,\mathrm{cfm/ft^2}$ of OA equally all over the building (Figure 5.5).

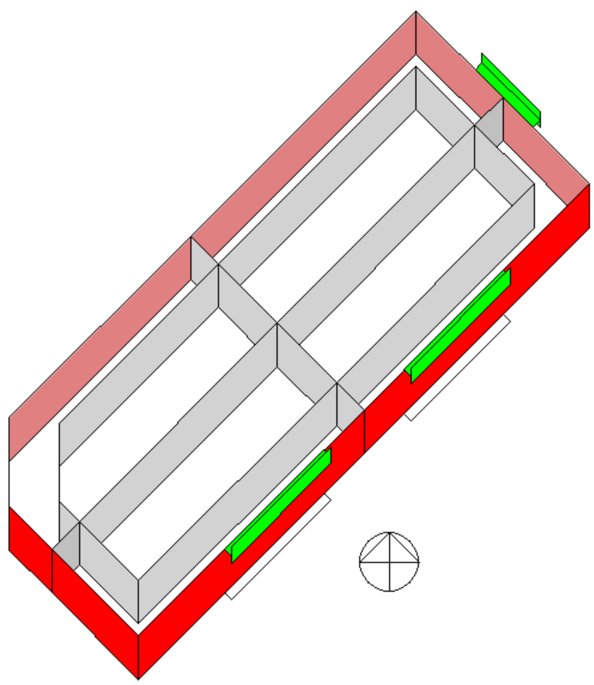


Figure 5.5. Valley Park office building envelope design for dedicated outdoor air system simulation model

The thermal conductivity for exterior walls was set to the minimum level of $0.0001\ \mathrm{Btu/Hr\text{-}ft^2\text{-}{}^\circ F}$, which is the lowest allowed value of conductivity for exterior walls in DOE-2.1E (Table 5.9). All interior walls in the DOAS model were assumed to be adiabatic to prevent any energy transfer between zones. The building composite weight of the floor, furnishings and interior walls of a space in the baseline building model was estimated automatically, based on specified materials. In the dedicated outdoor air system model value of building floor-weight is estimated to be 70 lb/ft^2 (standard ASHRAE weighting factor for medium weight construction) (LASL 1980).

Table 5.9. Decoupled HVAC system parameters comparison

DOE-2 Keyword	Units	Baseline HVAC system	DOAS	Air conditioning system	Description
ROOF-1	Btu/Hr-ft^2-°F	0.048	0.0001	0.048	Roofs insulation entirely above deck
WALL-EXT	Btu/Hr-ft^2-°F	0.149	0.0001	0.149	Walls, above grade, mass
WALL-U	Btu/Hr-ft^2-°F	0.032	0.0001	0.032	Slab on grade, unheated
AREA/PERSON	ft^2/person	100	-	100	
NUMBER-OF-PEOPLE	Person	-	0	-	
LIGHTING-W/SQFT	W/ft^2	0.9	0	0.9	
EQUIPMENT-W/SQFT	W/ft^2	0.8	0	0.8	
FLOOR-WEIGHT	Lb/ft^2	Auto	70	Auto	
CFM/SQFT	cfm/ft^2	1	0.11	0.89	Minimum air flow rate to the zone per ft^2 of zone area
OUTSIDE-AIR-CFM	Cfm	2836	2836	0	
MIN-SUPPLY-T	°F	55	55	55	
SUPPLY-CFM	Cfm	Auto	2840	Auto	The design capacity of the system air supply fan
MIN-CFM-RATIO		0.15	1	0.15	

According to the DOAS building energy performance summary total electricity consumption is 24,216 kWh. Of this amount of energy, 42% is used for heating, 36% for cooling and 22% for ventilation fans.

The simulated air conditioning system has the same design as a baseline model except no OA is provided to the building. According to the simulation report, 80% of all HVAC system energy consumption is used for cooling, 11% for ventilation fans and 8% for heating.

The dedicated outdoor air system and the air conditioning system operate in parallel, which eliminates the need for reheat of DOAS supply air, since OA is reheated by mixing with conditioned supply air. At the same time, this process will reduce the simultaneous cooling requirements of the air conditioning system. To evaluate the performance of a decoupled system, the total heating energy consumption was estimated to be equal to the air conditioning system's heating consumption. The total cooling system's energy consumption was estimated as the sum of the cooling energy consumption for both components minus reheat energy used in the DOAS and divided by the COP of the air conditioning system (2.9308). This change will provide a balance between the reheat energy provided for the DOAS and the simultaneous cooling energy consumption provided by the air conditioning system. In a parallel operation of two systems, both components are compensating each other (Table 5.10).

Table 5.10. Decoupled HVAC system model energy consumption comparison with baseline design in kWh/year

Category	Baseline	DOAS	Air conditioning system		Decoupled HVAC system		Reduction of energy consumption compared with baseline	
Evaporator coil temperature	55ºF	55ºF	55ºF	57ºF	55ºF/ 55ºF	55ºF/57ºF	55ºF/ 55ºF	55ºF/ 57ºF
HEATING	15,284	10,075	8,485	8,463	8,485	8,463	6,800	6,821
COOLING	99,823	8,774	81,994	79,259	87,331	84,596	12,491	15,227
FANS	11,691	5,367	11,551	13,328	16,918	18,696	-5,227	-7,005
AREA LITE	80,248	0	80,248	80,248	80,248	80,248	0	0
END-USE EQUIP	71,332	0	71,332	71,332	71,332	71,332	0	0
TOTAL LOAD	278,377	24,216	253,609	252,630	264,314	263,335	14,063	15,042

A parametrical study was provided to estimate the optimum supply air temperature of the air conditioning system. According to the obtained results, an additional efficiency increase could be provided by the air conditioning system evaporator temperature increase from 55ºF (required for dehumidification in the baseline system) to 57ºF (Table 5.11). This value is influenced by an increase of the supply air fans energy consumption caused by higher supply air flows, which is required to provide the same amount of cooling. An air conditioning system evaporator temperature increase above 57ºF abates savings achieved in the refrigeration cycle efficiency improvement.

Table 5.11. Minimum supply air temperature influence on air conditioning system energy consumption level

MIN-SUPPLY-T, F	Total Energy, kWh/year	Difference, kWh/year
54	254,443	833
55	253,609	0
56	252,941	-668
57	252,630	-979
58	252,636	-974
59	252,888	-721
60	253,491	-118

The optimization of the air conditioning system evaporator temperature will decrease the decoupled HVAC system energy consumption by 979 kWh/year to 263,335 kWh/year (Figure 5.6).

The analysis of energy consumption for the baseline model and the decoupled system shows that the heating energy consumption decreased by 45 %, the cooling energy consumption decreased by 15 %, and the supply air fan energy consumption increased by 60 %.

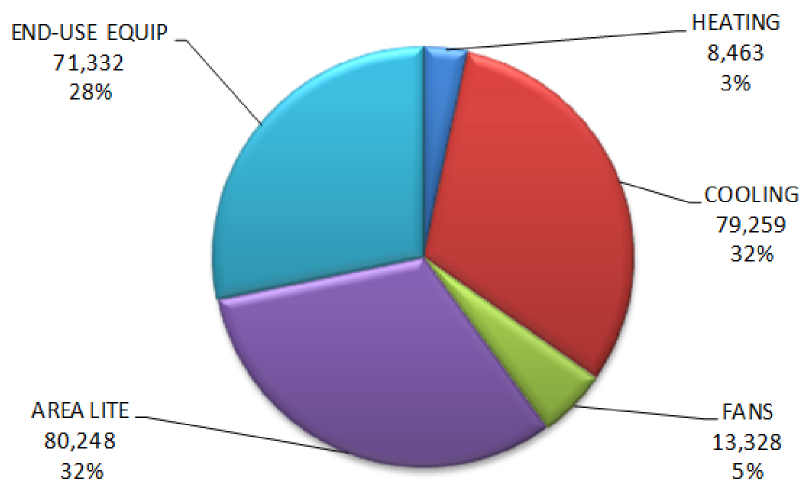


Figure 5.6. Valley Park office building with decoupled HVAC system annual energy consumption summary in kWh/year

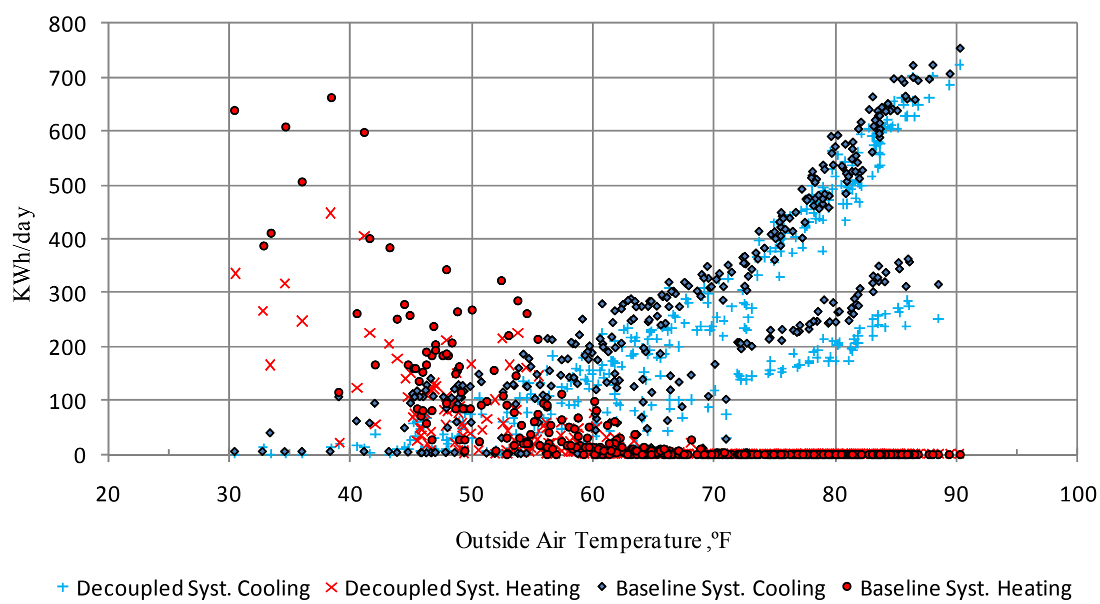


Figure 5.7. Valley Park office building baseline model and decoupled HVAC system model electrical energy consumption

The overall improvement in HVAC system energy consumption is 15,042 kWh/year or 5% from the total building energy consumption or 12% from the baseline HVAC system energy consumption level (Figure 5.7).

5.6. The baseline HVAC system additional efficiency improvements analysis

For the Valley Park office building baseline energy consumption model, following possible measures to increase the energy efficiency of the HVAC system were selected: air side economizer, increase of cooling unit electric input ratio, air source heat pump. In addition a natural gas heat pump was considered to evaluate possible energy efficiency improvement.

In HVAC systems, the water side economizer and OA economizer are normally mutually exclusive. The system should only have one of these options (LBNL. 1993b). For the HVAC improvements modeling, air side economizer was chosen. The outside air economizer mode simulates an economizer that returns the OA damper to a minimum if the OA enthalpy is higher than the return air enthalpy or if the OA temperature is higher than DRYBULB-LIMIT (or ECONO-LIMIT-T) (which defaults to the return air temperature). The economizer is simulated to operate between 18ºF (lowest hourly temperature from the weather file) and 70ºF. The application of the economizer decreased the cooling energy consumption by 3% and the total building energy consumption by 1.1% from the baseline level (Table 5.12).

Table 5.12. Improved Valley Park office building models annual electrical energy consumption in kWh

Category	Baseline	Economizer	Efficiency increase to 12.8 EER	Air-to-air heat pump
HEATING	15,284	15,293	15,284	6,483
COOLING	99,823	96,852	78,052	87,261
FANS	11,691	11,693	11,691	11,691
AREA LITE	80,248	80,248	80,248	80,248
END-USE EQUIP	71,332	71,332	71,332	71,332
TOTAL Electric LOAD	278,377	275,417	256,607	257,015

The baseline model design uses a 10.0 EER. The cooling energy efficiency could be increased to a 12.8 EER (3.75 COP) by using, for example, 4 RT rooftop unit manufactured by Carrier (Carrier 2010).

The Centurion 50PD rooftop units, single package, single-zone variable airflow rooftop units with electric cooling and an optional electric heating could provide a 15.2 SEER and equipped with a variable capacity compressor and a VFD indoor fan motor. The rooftop unit compressor is able to provide any capacity between 15 and 100% by capacity modulation. Based on the simulation results in DOE-2.1E an EER increase from 10.0 to 12.8 provides 21.8% of cooling energy savings or 7.8% of total building energy consumption level.

The application of the air source heat pump was simulated by using standard value for the energy input ratio 0.37 (LBNL 1993a) that corresponds to 2.7 COP. A change of heat source for the central heating coils from the electric heater to an air-to-air heat pump reduced the heating energy consumption by 57.6% and the cooling energy consumption by 12.6%. The overall building energy consumption was reduced by 7.7% (Figure 5.8).

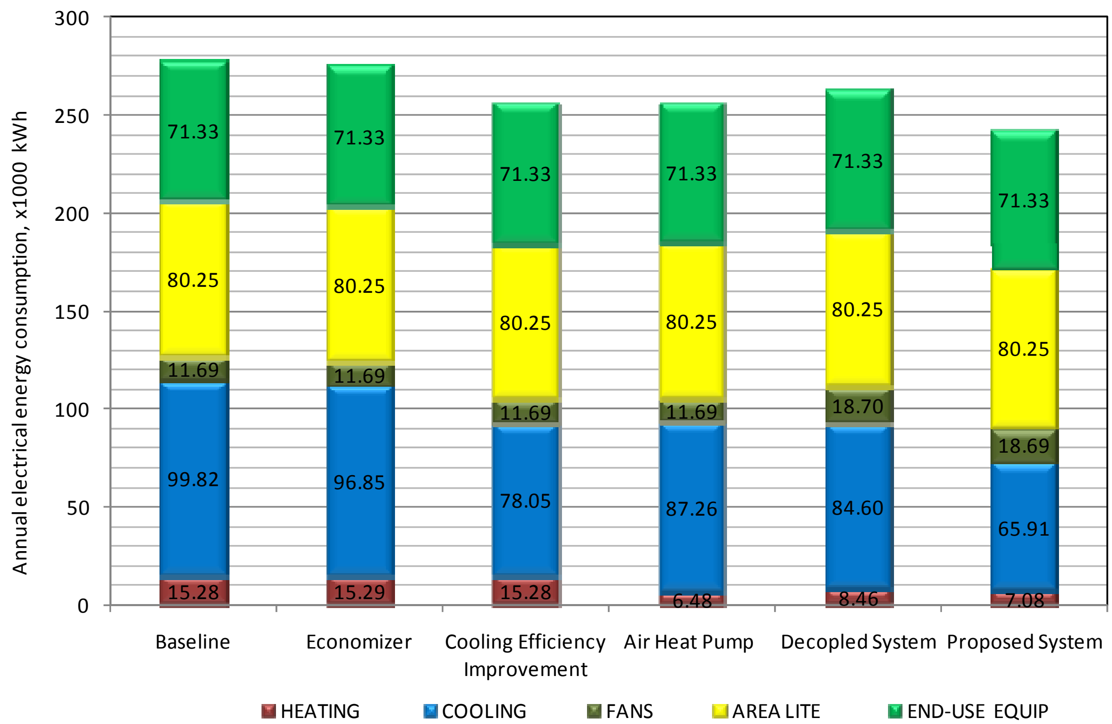


Figure 5.8. Valley Park office building baseline and improved models electrical energy consumption

In addition to energy efficient improvements that could potentially be implemented in Valley Park office building, the application of a natural gas heat pump was considered to evaluate possible energy consumption and power reduction potential. The building was simulated with a natural gas engine driven heat pump that is used for both heating and cooling. The speed of the natural gas engine was adjusted for capacity modulation. Application of this system reduced the total building electrical energy consumption by 37%. The electrical energy consumption for cooling was reduced by 89.7%, heating by 93.5% and fans by 0.6% and the overall HVAC system electrical energy consumption decreased by 82%. The annual natural gas consumption was estimated to be 2,096 therm (210 MMBtu).

Considering the highest wellhead market price for natural gas in the U.S. equal to \$4.75/MMBtu (EIA 2010) and the electrical energy price on campus in College Station \$0.113/kWh, the annual price of consumed energy could be reduced from \$31,457 to \$20,708 (\$19,712 used for electrical energy and \$995.6 for natural gas). The additional important benefit provided by this system is the reduction of hourly maximum electrical energy demand from 114 kW to 56 kW (56% reduction). This opportunity is highly important for the condition of highly wind energy dominated electrical energy generation in Texas.

5.7. Valley Park office building improved system design

Based on simulation results for the Valley Park office building, it is recommended to include in the optimum design the decoupled HVAC system described in Chapter V, section 5.5. Additional recommendations for optimum design include: air source heat pump and cooling unit with increased energy efficiency. An air-side economizer mode cannot be implemented simultaneously with a decoupled HVAC system since the ventilation system operates at 100% OA and is not designed to provide a sufficient amount of OA for complete building cooling. A natural gas heat pump was not considered, since natural gas is not available at the building location.

A decoupled HVAC system included the DOAS, with a cooling efficiency 12.8 EER and the air conditioning system with an evaporator coil temperature of 57ºF, cooling system efficiency of 12.8 EER and air heat pump. Applications of the proposed measures reduced the energy consumption by 35,117 kWh/year or 12.61 % from a total building energy consumption (Figure 5.8). The cooling energy consumption was reduced by 34 %, heating energy consumption reduced by 54 % and supply fans energy consumption increased by 60 % (Table 5.13).

Table 5.13. Valley Park office building model with decoupled HVAC system and improved HVAC system components annual energy consumption in kWh

Category	Baseline design	DOAS	Air Conditioning system	Decoupled HVAC system	Reduction of energy consumption compared with the baseline
Evaporator coil temperature	55ºF	55ºF	57ºF	55ºF/57ºF	55ºF/57ºF
HEATING	15,284	10,075	7,078	7,078	8,206
COOLING	99,823	6,928	61,666	65,908	33,914
FANS	11,691	5,366	13,328	18,695	-7,003
AREA LITE	80,248	0	80,248	80,248	0
END-USE EQUIP	71,332	0	71,332	71,332	0
TOTAL LOAD	278,377	22,369	233,647	243,260	35,117

Overall, the consumed electrical energy was reduced from 10.80 kWh/ft^2-yr to 9.43 kWh/ft^2-yr. In both cases an interior gain from lighting and office equipment is 5.88 kWh/ft^2-yr. The simulation of the decoupled HVAC system shows that the supply air fans consume 0.45 kWh/ft^2-yr. This value could be substantially reduced by chilled beam applications instead of the forced air circulation system for air conditioning. The proposed system with a change from forced air circulation to chilled beams/displacement ventilation can decrease the energy consumption level to 8.98 kWh/ft^2-yr. This energy consumption level includes 5.88 kWh/ft^2-yr of interior heat gain for office equipment and lighting system. The total simulated energy efficiency improvement of the proposed HVAC system is 43.33 % compared to the baseline design.

CHAPTER VI

VALLEY PARK OFFICE BUILDING THEORETICAL MINIMUM ENERGY CONSUMPTION ESTIMATION

To estimate the maximum limit of energy consumption reduction potential, a theoretical analysis of the Valley Park office building was performed. Energy in the office building is consumed by three main components: HVAC system (cooling and heating loads), lighting system and building equipment.

The HVAC system energy consumption is determined by heating and cooling loads, rates of energy input or removal required to maintain an indoor environment at a desired temperature and humidity condition (ASHRAE 2009). Heating and cooling loads depend on external (outside air) and internal conditions (number of people, lighting and equipment energy consumption). The amount of variables that influence building energy consumption are numerous interrelated and often hard to estimate precisely.

Main sources of a building space heat gain are (ASHRAE 2009):

- solar radiation through transparent surfaces,
- heat conduction through exterior walls and roofs,
- heat conduction through ceilings, floors, and interior partitions,
- heat generated in the space by occupants, lights, and appliances,
- energy transfer through direct-with-space ventilation and infiltration of outdoor air,
- miscellaneous heat gains.

There are two types of space heat gains: sensible (added by conduction, convection and radiation) and latent (from water vapor emission). To maintain the temperature and humidity level, space heat gains should be extracted from the space at the same time that they are generated in order to keep a constant temperature and humidity level, assuming no time delay effects on loads appear. In other words, space heating and cooling loads should be balanced.

The Valley Park office building energy consumption minimum limit estimation is based on humid air parameters with a pressure at the sea level (14.696 psia) and normal temperatures and moisture ratios. The calculation method for the office building air conditioning zone includes some level of assumption that makes the results approximate. This calculation is based on the following assumptions:

- air in the thermal zone can be modeled as well mixed (constant properties throughout the zone);
- all zone surfaces (walls, floor, etc.) are diffuse radiating isothermal surfaces with uniform long-wave and short-wave irradiation;
- all exterior elements that border with ambient air are adiabatic;
- equipment used in the building is operating at the maximum efficiency;
- thermal energy storage of the building is negligible;
- the building has no transparent surfaces (windows);
- light to the building is provided from the ideal "white" light source;
- the building has neither moisture migration nor infiltration (exfiltration) through the building envelope;
- cooling and heating equipment operates at the maximum possible efficiency of the Carnot heat engine;
- hydronic system pumps energy consumptions is neglected;
- calculations are provided on a daily basis with the assumption of steady state process and constant properties of air and water vapor.

6.1. Interior heat loads

The main source of building internal heat gain are people, lights, motors, appliances, and equipment.

According to ASHRAE (2009a), during moderately active office work, representative rates at which heat gain generated by a person are 250 Btu/h of sensible heat and 200 Btu/h of moisture (latent heat). The building is assumed to be occupied according to the same schedule that was used in the previous chapter for the Valley Park office building simulation.

The main source of heat gain from lighting in actual office areas comes from light-emitting elements and additional lighting system components like ballasts or other equipment used in lighting systems. Since thermal energy storage is assumed to be zero, the distinction between convective and radiative components is not made in these calculations. The building is equipped with ideal light systems that distribute the light flux equally all over the office zone area. Application of this system eliminates energy consumptions related with an invisible lighting spectrum and side energy consumption that are produced by lighting equipment.

According to The Illuminating Engineering Society of North America (Chen. 1999), illuminance requirements for an office area are 20-50 foot-candles (200-500 lux) (upper limit for performance of visual task of high contrast or large size or lower limit for performance of visual task of medium contrast or small size). There is no straight forward way to recalculate the illuminance requirements in lux ($\mathrm{lm/m^2}$) (photometric unit) to corresponding irradiance (radiometric unit) that is based on physical power. In radiometric units all wavelengths are weighted equally, while photometric units take into account the fact that the human eye's visual system is more sensitive to some wavelengths than others. To balance this difference, every wavelength is given a different weight according to luminosity function (McCluney 1994). Based on a luminous efficacy of radiation and assumption that light source is an ideal “white” light source from a black body with temperature 5800 K and truncated to 400–700 nm, an ideal lighting system efficiency limit is 251 lm/W (IES Lighting Handbook 2000).

This value is three times higher than 80 lm/W sunlight efficiency (for Class G star with surface emission temperature 5800 K) or two times higher than 124 lm/W spiral tube fluorescent lamp with electronic ballast (Panasonic 2010; U.S. DOE 2010e).

This luminous efficacy value is estimated according to the equation (Arecchi, Messadi, and Koshel 2007):

$$F(\lambda_1, \lambda_2) = 683 \int_{\lambda_1}^{\lambda_2} \Phi_\lambda(\lambda) \cdot V(\lambda) \cdot d\lambda \tag{6.1}$$

where: 683 – constant from the definition of the candela;

$\lambda_1 = 400nm$; $\lambda_2 = 700nm$ - borders of visible spectrum range;

$\Phi_\lambda(\lambda)$ - spectral emissive power (Planck distribution), in the band between λ_1 and λ_2;

$V(\lambda)$ - photopic luminous function.

Based on luminous emittance of 50 foot-candles (500 lux) energy provided by the lighting system to the office space is 2 W/m^2 or 0.186 W/ft^2. Energy that is consumed by a lighting system will be transferred to the conditioning zone area and become part of internal heat gain.

The main application of electric motors in an office building is fan drivers for building forced air circulation systems. Application of a decoupled HVAC system, as the most advanced principle, will require fan energy just for ventilation system to supply required amount of OA. Based on the DOAS simulation results for Valley Park office building, current level of supply air fan and motors efficiency as well as air distribution system, estimated total electrical power of OA supply system motors is 1,945 W. Energy that is consumed by electric motors will be transferred to the conditioning zone area and become part of internal heat gain.

Office appliances and equipment energy consumption estimation should consider all of the most common appliances (electrical, gas, or steam). Any possible estimate of this category is very subjective, caused by a big variety of appliances, applications, schedules, uses, and installations. The main categories of office equipment that should be considered are following: personal desktop computers, monitors, printers, copy machines, microwaves, water coolers and refrigerators.

Based on the assumption that 1 GHz is sufficient computational power, theoretical energy consumption limit could be estimated as an energy consumed by an iPhone 4 (2.5W based on battery nominal parameters 5Vx0.5A). Minimum energy consumption by a computer monitor could be estimated based on light flux provided by existing monitors. Assuming a monitor area is 0.14 m^2 (monitor with 23 in diagonal), light flux 250 cd/m^2 (Samsung 2010) and ideal "white" light source energy efficiency 251 lm/W, energy consumption of an ideal monitor is:

$$P_{monitor} = 250\frac{cd}{m^2}\cdot\frac{4\pi\cdot lm}{cd}\cdot\frac{W}{251lm}\cdot 0.14\mathrm{m}^2 = 1.752W \qquad (6.2)$$

Assuming each personal computer has two monitors and one computational unit, the total energy consumption will be 6 W during office hours and zero during the unoccupied time. All consumed energy will be transferred to the conditioning zone area and become part of the internal heat gain.

Laser printers and copiers that are mostly used in offices are required to maintain a fusser roller temperature at 400°F during printing and to maintain a high temperature to avoid delays in printer request during standby mode (RISO 2010). This requirement makes laser printers one of biggest energy consumers in the office at current level of technologies. Energy Star® certification for printers is aimed to decrease idle energy consumption and does not limit the amount of energy consumed for printing (NRC 2010). Assuming that liquid inkjet printing technology is to be used for printing, the energy consumption by printers and copy machines could be neglected.

Energy consumption by microwave ovens could be assumed as energy required for heating one glass of water (8.45oz/250ml) from 50°F (10°C) to 212°F (100°C). The amount of energy consumed by a microwave and transferred to the conditioning zone:

$$W_{food} = V\cdot\rho\cdot Cp\cdot\Delta T = 0.25\cdot 10^{-3}m^3\cdot 997\frac{kg}{m^3}\cdot 4.18\frac{kJ}{kgK}\cdot 90K\cdot\frac{kWh}{3600kJ} = 26Wh \qquad (6.3)$$

Assuming one meal per day for each person, heat gain from microwave (food heating) is 26Wh/person-day. This value also includes possible use of coffee machines.

Energy consumed by water coolers could be estimated as an energy required to cool $10^{-3}m^3$/peron-day of water from 75°F (24°C) to 59°F (15°C).

$$Q_{water} = V \cdot \rho \cdot Cp \cdot \Delta T = 10^{-3}m^3 \cdot 997\frac{kg}{m^3} \cdot 4.18\frac{kJ}{kgK} \cdot 9K \cdot \frac{kWh}{3600kJ} = 10Wh \quad (6.4)$$

The Carnot cycle efficiency for water cooling system is:

$$COP_R = \frac{1}{T_H / T_L - 1} = \frac{1}{297K / 288\mathrm{K} - 1} = 32 \quad (6.5)$$

Heat gain by a water cooler to the space per person per day:

$$W_{water\_gain} = Q_{water}\left(\frac{1}{COP_R} + 1\right) = 10Wh\left(\frac{1}{32} + 1\right) = 10.3Wh \quad (6.6)$$

Electrical energy consumption by a water cooler per person per day:

$$W_{water} = \frac{Q_{water}}{COP_R} = \frac{10Wh}{32} = 0.312Wh \quad (6.7)$$

Assuming that a refrigerator container walls are adiabatic, energy consumption by refrigeration equipment could be neglected.

Based on the described assumptions, estimated energy consumption in the building is 113 kWh/ day for weekdays (Table 6.1) and 47 kWh/day for weekends and holidays (Table 6.2).

Table 6.1. Valley Park office building theoretical minimum electrical energy consumption during weekday

Category	Measurement unit	Amount	Time	Energy Consumption		Energy use per category
			hr/day	Electricity	Dimension	Wh/day
People	Person	258	10			
Lighting	ft^2 of floor area	25,774	12	0.19	W/ft^2	57,528
Ventilation system motors	Building	1	18	1,945	W/building	35,010
Personal computer	PC	258	9	6.00	W/ PC	13,932
Food reheating	cup	258		26	Wh/day- cup	6,708
Water cooler	glass	258		0.31	Wh/day glass	80
Total						113,258

Table 6.2. Valley Park office building theoretical minimum electrical energy consumption during weekend and holiday

Category	Measurement unit	Amount	Time	Unit Energy Consumption		Energy use per category
			hr/day	Electricity	Dimension	Wh/day
People	Person	258	1.35			
Lighting	ft^2 of floor area	25,774	4	0.19	W/ft^2	19,176
Ventilation system motors	Building	1	8	1,945	W/building	15,560
Personal computer	PC	258	4	6.00	W/ PC	6,192
Food reheating	cup	258		26	Wh/day- cup	6,708
Water cooler	glass	258		0.31	Wh/day glass	80
Total						47,716

Based on the described energy consumption level in the building, the interior heat gain for a weekday is 1,040 kBtu/day of sensible load and 516 kBtu/day of latent load (Table 6.3). For weekends and holidays, the interior heat gain is four times lower (Table 6.4).

Table 6.3. Valley Park office building theoretical minimum interior heat gain during weekday

Category	Measurement unit	Amount	Time	Interior heat gain per measurement unit			Heat gain per category, kBtu/day	
			hr/day	Sensible	Latent	Dimension	Sensible	Latent
People	Person	258	10	250	200	Btu/hr-person	645	516
Lighting	ft^2 of floor area	25,774	12	0.64		Btu/hr- ft^2	196	
Ventilation system motors	Building	1	18	6,636.30		Btu/hr-building	119	
Personal computer	PC	258	9	20.50		Btu/hr- PC	48	
Food reheating	cup	258		88.70		Btu/day- cup	23	
Water cooler	glass	258		35.14		Btu/day- glass	9	
Total							1,040	516

Table 6.4. Valley Park office building theoretical minimum interior heat gain during weekend and holiday

Category	Measurement unit	Amount	Time	Interior heat gain per measurement unit			Heat gain per category, kBtu/day	
			hr/day	Sensible	Latent	Dimension	Sensible	Latent
People	Person	258	1.35	250	200	Btu/hr-person	87	70
Lighting	ft^2 of floor area	25,774	4	0.64		Btu/hr-ft^2	65	
Ventilation system motors	Building	1	8	6,636.30		Btu/hr-building	53	
Personal computer	PC	258	4	20.50		Btu/hr- PC	21	
Food reheating	cup	258		88.70		Btu/day- cup	23	
Water cooler	glass	258		35.14		Btu/day- glass	9	
Total							259	70

In this calculation time, usage of each category per day was used to balance occupancy level. This value corresponds to the same average level that was used in the baseline building model simulation.

6.2. Air conditioning, ventilation and infiltration loads

Energy transfer through direct-with-space ventilation and infiltration of outdoor air depends on building pressurization level and ventilation requirements. According to ASHRAE 62.1-2004, Valley Park office building requires 2,836 cfm of OA. This amount of air also keeps the building pressurized and avoids OA infiltration. In an idealized case, no infiltration or exfiltration should occur in the building.

Valley Park office building's idealized HVAC system is based on the decoupled principle (Figure 6.1). Outside air is provided by DOAS through an energy recovery system. The energy recovery system is bypassed in case if operation is not desirable. This system is idealized and can provide 100% recovery of available energy from return air flow to the supply air flow (air parameters of both flows at the exit from energy recovery are the same). The energy recovery system is a passive device and requires no energy input for operation. Cooling energy for air conditioning to the Valley Park office building is provided by an idealized hydronic system with heat exchanger located at the ceiling level and includes a hydronic economizer capable to provide free cooling if the return air dry-bulb temperature is above OA wet-bulb temperature. The air conditioning system's heating energy is provided by an idealized hydronic heating system with a heat exchanger located at the floor level. Both the cooling and heating systems operate with a maximum potential efficiency of the Carnot cycle and exchange energy with OA.

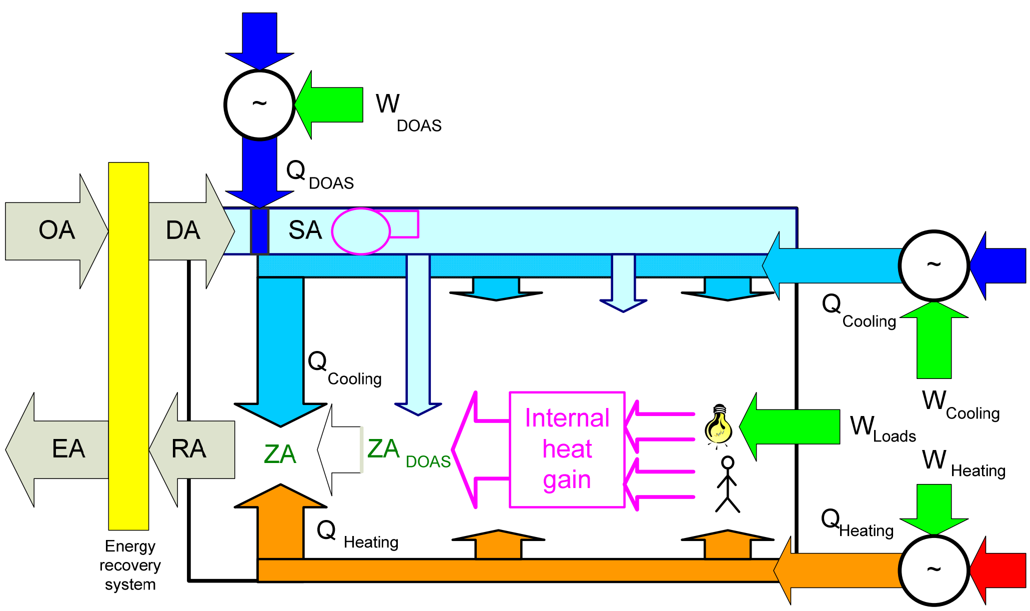


Figure 6.1. Valley Park office building idealized HVAC system energy transfer balance

Parameters of outside air used in calculation are obtained from the Typical Meteorological Year file Version 3 that was also used in the previous building simulations.

The building energy balance was estimated based on the iterative principle. Outside air parameters are used as input parameters of the system on the first iteration of calculation as discharged air parameters. For the second iteration, discharged air properties were balanced by energy recovered from the return air flow estimated on the first iteration step.

6.2.1. First iteration

Outside air is supplied to the building by the DOAS according to the operation schedule (18 hr/day during weekdays and 8 hr/day during weekends and holidays). Supply air (SA) parameters to the conditioned zone are fixed over the year with a dry-bulb temperature of 55°F and a relative humidity of 100% unless the ventilation cooling system is deactivated. The ventilation cooling system should be deactivated if humidity ratio is not to exceed 0.009184 lb_w/lb_a. In this case, supply air could be provided to the building without dehumidification. In case the outside air humidity ratio exceeds 0.009184 lb_w/lb_a, the supply air flow is conditioned to the water vapor saturation at the dry-bulb temperature of 55°F. This requires a cooling coil surface temperature to be 55°F. Based on this control strategy, the dedicated outdoor air cooling system will be disabled 147 days per year or 40% of the time considering OA as an input to the DOAS cooling coil.

When the dedicated outdoor air cooling system is enabled, the cooling energy provided by the system per day could be estimated as follows:

$$Q_{OA} = t \cdot \dot{V} \cdot \rho \cdot \left(h_{OA} - h_{SA}\right), \tag{6.8}$$

where: t - time of ventilation system operation, min /day;

$\dot{V}$ - Volume of outside air, ft^3/min;

ρ - Density of air, 0.075 lb/ft^3;

h_{OA} - Enthalpy of heat recovery discharge air (equal to the OA temperature during the first iteration), Btu/lb$_m$;

h_{SA} - Enthalpy of supply air, Btu/lb$_m$.

For the interior heat gains (supply air fans, lighting system and other) are added to the supply air flow after the cooling coil. During the weekday, internal heat gains are estimated to be 1,040 kBtu of sensible and 516 kBtu of latent load. During the weekends and holidays, the building interior heat gain is 259 kBtu of sensible and 70 kBtu of latent load.

According to the sensible load definition, the supply air temperature by the air ventilation system to the zone could be estimated as follows:

$$T_{ZA\_DOAS} = T_{SA} + \frac{Q_{sen}}{\rho \cdot C_P \cdot \dot{V} \cdot t}, \tag{6.9}$$

where: T_{ZA_DOAS} - Temperature of the air provided by DOAS in the conditioned zone after internal building heat gain was added, °F;

T_{SA} - Supply air temperature after the cooling coil, °F;

Q_{sen} - Sensible component of interior heat load, Btu/day;

C_P - Specific heat of dry air, 0.24 Btu/(lb$_m$·°F).

According to the latent load definition, the humidity ratio of supply air after latent component of internal building heat gain was added could be estimated according to the formula:

$$\omega_{ZA\_DOAS} = \omega_{SA} + \frac{Q_{lat}}{\rho \cdot h_{fg} \cdot \dot{V} \cdot t}, \tag{6.10}$$

where: ω_{ZA_DOAS} - humidity ratio of the air provided by DOAS to the zone, lb$_w$/lb$_a$;

ω_{SA} - supply air humidity ratio after the DOAS cooling coil, lb_w/lb_a;

Q_{lat} - Sensible component of interior heat load, Btu/day;

h_{fg} - Heat of water vaporization, 970.3 Btu/lb_m.

To maintain comfort requirements, cooling or heating should be provided to the air supplied by the DOAS at state of $ZA_{\_DOAS}$ to reach required zone air condition ZA. The zone air required condition is defined by dry-bulb air temperature setpoint of 75°F. Since an air conditioning system should avoid water condensation on cooling surfaces, it does not provide dehumidification and humidity ratio of the zone air stay constant. The surface temperature of the air conditioning system cooling coil is estimated to be slightly above the dew point temperature of $ZA_{\_DOAS}$ air.

According to the first law of thermodynamics energy balance for conditioned zone:

$$\dot{E}_{Return\_Air} - \dot{E}_{Supply\_Air} = \Delta U - \Delta KE - \Delta PE, \tag{6.11}$$

where ΔU - change of internal energy;

ΔKE - change of kinetic energy;

ΔPE - change of potential energy.

The change of kinetic and potential energy can be neglected, considering that supply air velocity and elevation change is small. The change of air internal energy is equal to the amount of heating or cooling provided by the air conditioning system to the office area.

The heating and cooling energy of water vapors in the zone could be neglected since condensation does not occur and the product of water specific heat and humidity ratio is significantly smaller compared with the specific heat of air. The required change of internal energy (cooling or heating) of the zone air can be estimated based on dry air properties.

$$Q_{Cooling/heating} = t \cdot \dot{V} \cdot \rho \cdot C_P \cdot \left(T_{ZA\_DOAS} - T_{ZA\_setpoint}\right), \quad (6.12)$$

where: $T_{ZA\_setpoint}$ - zone air temperature setpoint, 75°F.

If this value has a negative sign, heating is required; if positive – cooling.

Based on the previous assumption, the amount of OA supplied to the building is equal to the amount of return air. Since no energy or mass is transferred to the air at a state described since zone air (ZA) before air becomes return air, all air properties of return air will remain the same.

6.2.2. Second iteration

To reduce energy consumption of the HVAC system, the energy recovery system could be used. Assuming the mass of return dry air is equal to the mass of dry air supplied to the building and 100% efficiency of heat recovery, OA parameters after the heat recovery system (discharge air) should be equal to exhaust air parameters in case if energy recovery system is enabled. The energy recovery system should be enabled if the enthalpy of return air is below the enthalpy of OA. An example of calculation results for the week day May 26 is provided in Table 6.5.

Table 6.5. Valley Park office building idealized HVAC system simulation results for 26th of May

Category	Parameter	Units	Iteration 1 (without heat recovery)	Iteration 2 (with heat recovery)
Outside air	Twb	°F	69.7	69.7
	Tdb	°F	79.5	79.5
	ω	lb_w/lb_a	0.013310	0.013310
	h	Btu/lb	33.7	33.7
Heat recovery			Disabled	Enabled
Heat recovery discharge air	h	Btu/lb	33.7	28.3
	ω	lb_w/lb_a	0.013310	0.011334
	Tdb	°F	79.5	77.25
Air flow	Flow	Cfm	2836	2836
	Operation length	Hr	18	18

Table 6.5. (Continued)

Category	Parameter	Units	Iteration 1 (without heat recovery)	Iteration 2 (with heat recovery)
Supply air	Tdb	°F	55	55
	ω	lb_w/lb_a	0.009184	0.009184
	h	Btu/lb	23.2	23.2
DOAS cooling energy provided	Q	Btu/day	2,412,018	1,164,660
DOAS cooling electrical energy consumption	W_{DOAS}	kWh/day	20	9.73
ZA_{DOAS}	Tdb	°F	73.87	73.87
	ω	lb_w/lb_a	0.009358	0.009358
Return Air	Tdb	°F	75	75
	ω	lb_w/lb_a	0.009358	0.009358
	RH		0.507	0.507
	h	Btu/lb	28.26	28.26
Air conditioning cooling energy provided	Q_{CL}	Btu/day	0	0
Air conditioning cooling electrical energy consumption	W_{CL}	kWh/day	0	0
Air conditioning heating energy provided	Q_{HT}	Btu/day	62,234	62,234
Air conditioning heating electrical energy consumption	W_{HT}	kWh/day	0 ($T_{RAdb} < T_{OAdb}$)	0 ($T_{RAdb} < T_{OAdb}$)

Simulation results show that the proposed HVAC system for Valley Park office building with energy recovery system will provide 242.7 MMBtu/year of cooling and 28.4 MMBtu/year of heating energy (Table 6.6). Most of the building cooling energy is provided by DOAS and heat recovery (Figure 6.2).

Table 6.6. Valley Park office building idealized HVAC system annual energy balance

Energy flow	Units	Iteration 1 (without heat recovery)	Iteration 2 (with heat recovery)
DOAS cooling energy provided	Btu	461,046,648	196,423,005
DOAS cooling electrical energy consumption	**kWh**	**4,584**	**1,765**
Air conditioning cooling system energy provided	Btu	16,374,216	46,314,567
Mechanically supplied	Btu	0	0
Naturally supplied	Btu	16,374,216	46,314,567
Air conditioning cooling system electrical energy consumption	**kWh**	**0**	**0**
Air conditioning heating system energy provided	Btu	61,062,045	28,379,875
Mechanically supplied	Btu	45,603,369	12,921,198
Naturally supplied	Btu	15,458,676	15,458,676
Air conditioning heating system electrical energy consumption	**kWh**	**617**	**86**
Total electrical energy consumption	**kWh**	**5201**	**1851**

The Cooling and heating systems operate with the maximum Carnot cycle efficiency. The exterior heat exchanger of the dedicated outdoor air cooling system is water cooled to the OA wet-bulb temperature. The ideal dedicated outdoor air cooling system annual COP is 32.6 (SEER 111.3) in case that the heat recovery system is available during the year.

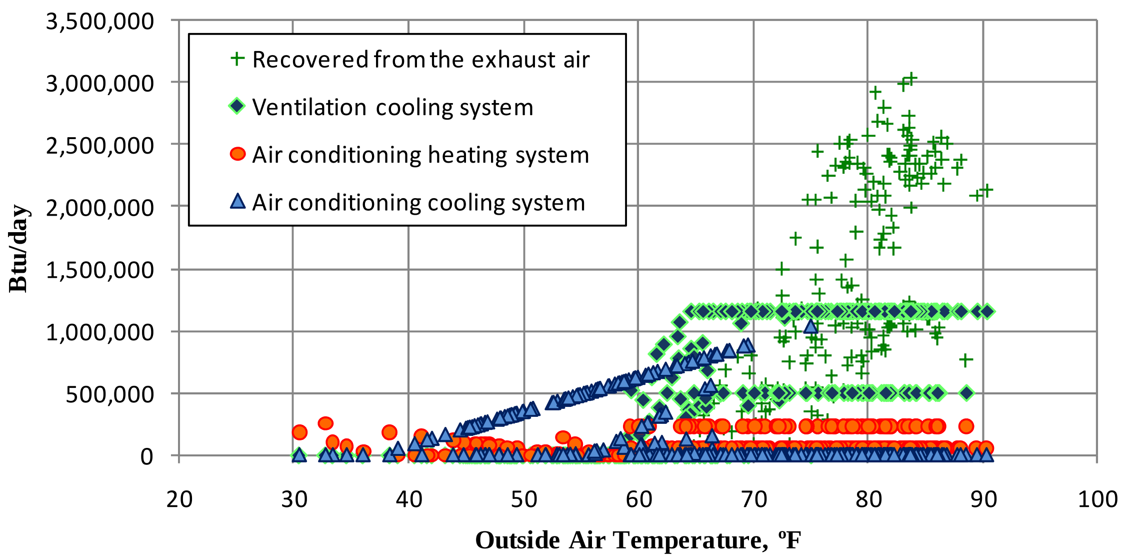


Figure 6.2. Valley Park office building theoretical minimum daily HVAC system thermal energy transfer

The air conditioning cooling system performance is based on OA wet-bulb temperatures and return air dry-bulb temperatures. In the climate of College Station and the simulated level of interior heat gain in time periods when the air conditioning cooling system is required, the OA wet-bulb temperature is below the return air dry-bulb temperature (75°F) and a water side economizer can be used. These conditions allow the mechanical compression cycle cooling for air conditioning systems to be eliminated (Figure 6.3).

The air conditioning heating system performance (heat pump) is based on OA dry-bulb temperature and return air dry-bulb temperature. In the hot and humid climate of College Station, the Valley Park office building air conditioning system required heating 113 days per year. The ideal air conditioning heating system annual COP is 44 (SEER 150) in case if the heat recovery system is available during the year.

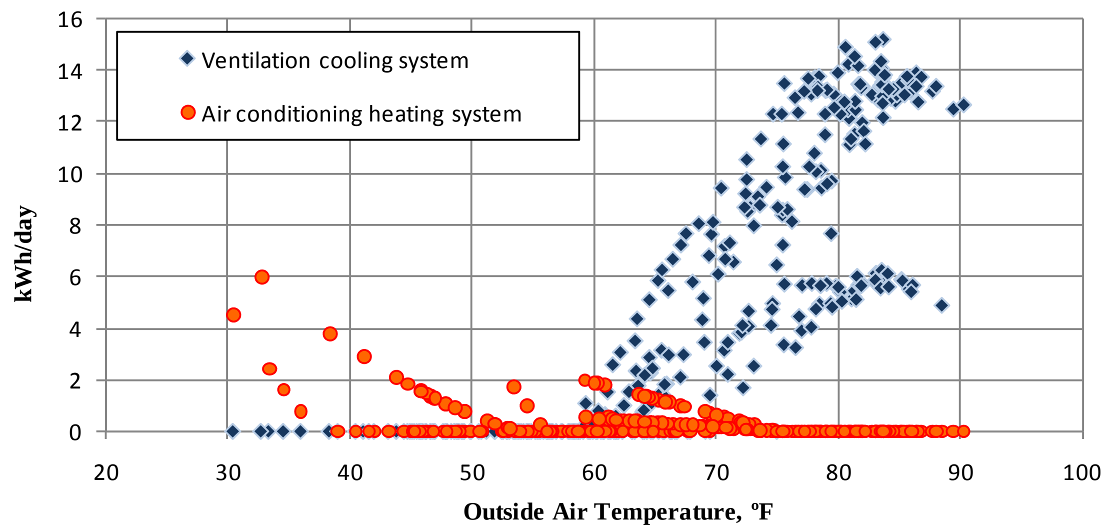


Figure 6.3. Valley Park office building theoretical minimum daily HVAC system electrical energy consumption

The estimation of cooling and heating efficiency is based on the average operation efficiency over the year when the system is used.

6.3. Theoretical energy consumption limit

The theoretical minimum energy consumption model results for Valley Park office building (Table 6.7) show that the lighting system will be still the biggest energy consumer with 46% from total energy consumption (Figure 6.4). Another big energy consumption category is the ventilation system fan motors (30%). This value is determined by supply air fan motors' efficiency as well as the supply fan and air distribution system design.

Table 6.7. Valley Park office building theoretical annual energy consumption limit

Category	Category	Building electrical load		
		kWh/year	kWh/year-ft^2	kBtu/year-ft^2
Interior loads	Lighting	16,664	0.647	2.206
	Personal computer	4,211	0.163	0.557
	Food reheating	2,448	0.095	0.324
	Water cooler	29	0.001	0.004
	Total	23,352	0.906	3.091
HVAC system	Dedicated outdoor air cooling system	1,765	0.068	0.234
	Air conditioning cooling system	0	0.000	0.000
	Air conditioning heating system	86	0.003	0.011
	Ventilation system motors	10,581	0.411	1.401
	Total	12,432	0.482	1.646
Total		35,784	1.388	4.737

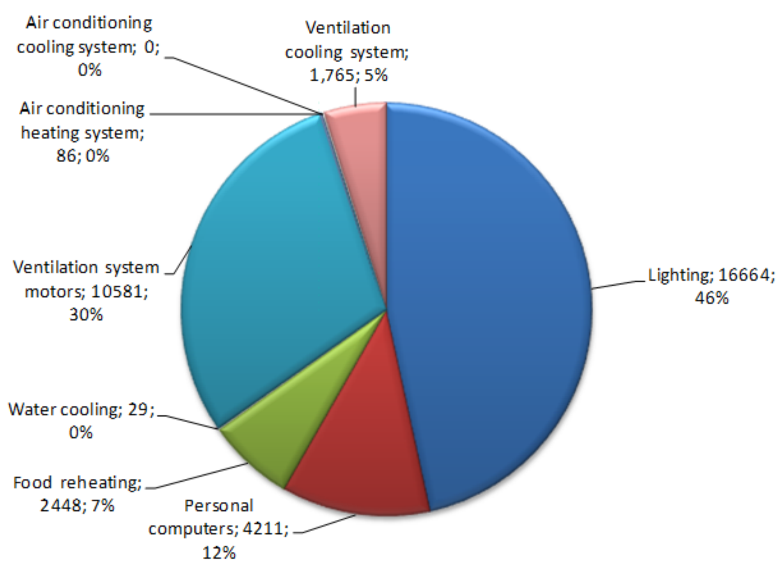


Figure 6.4. Valley Park office building annual theoretical minimum energy consumption balance in kWh

According to the calculation results, the theoretical energy consumption minimum level for the Valley Park office building is 1.4 $\mathrm{kWh/ft^2 yr}$ or 4.7 $\mathrm{kBtu/ft^2 yr}$. The estimated energy consumption limit shows that the baseline design for Valley Park office building efficiency is 13 % when compared with theoretical amount of energy used to provide required services (36.9 $\mathrm{kBtu/ft^2 yr}$).

The estimated energy consumption for an improved HVAC system (with displacement ventilation) is six times more than theoretical minimum limit (0.482 $\mathrm{kWh/ft^2 yr}$). For the Valley Park office building with an improved HVAC system, the building energy efficiency was estimated as 16% (29.6 $\mathrm{kBtu/ft^2 yr}$) of theoretical limit.

CHAPTER VII

SUMMARY AND RECOMMENDATIONS

7.1. Summary

This thesis describes HVAC system design and control strategies that can be used in hot and humid climates. According to the literature review, one of the most efficient concepts is the decoupled HVAC system that combines a DOAS and an air conditioning system. Air conditioning in this case could be provided by a displacement or mechanically air conditioning system.

According to the reviewed case studies, the most common strategies for energy efficient office buildings are: energy efficient building envelope construction materials; decoupled HVAC system; lighting control systems that include continuous dimming, daylighting and occupancy control); air-to-air energy recovery enthalpy wheel systems and economizer mode application; CO_2 sensors control of outside air supply; ground source heating and cooling system; photovoltaic systems.

Based on a real-world case study, the energy saving potential in office building located in College Station, Texas was illustrated. Wisenbaker Engineering Research Center total site energy consumption was 200.4 $\text{kBtu/ft}^2\text{yr}$ (58.73 $\text{kWh/ft}^2\text{yr}$) during the period from 1/1/2008 to 12/31/2008. The estimated lighting system consumption for this period was 8.0 $\text{kWh/ft}^2\text{yr}$ and the estimated office equipment consumption was 5.37 $\text{kWh/ft}^2\text{yr}$. Based on measured energy consumption data, field observation and construction documentation, the WERC building energy consumption model was developed and calibrated. Based on the calibrated model, a series of energy saving measures were proposed: outside air and supply flow adjustment (change from constant to VAV flow), change of HVAC operation schedule (night setback), economizer application, lighting system improvement. Implementation of proposed measures can reduce energy consumption by 46% to 107.9 $\text{kBtu/ft}^2\text{yr}$ (31.6 $\text{kWh/ft}^2\text{yr}$). The cost of energy saving measures implementation (without lighting system optimization) was estimated to be \$160,900, which provides an annual savings of \$362,000 and a simple payback less than six month.

Based on the requirements of valid energy conservation codes, a baseline simulation was created for a new Valley Park office building. The measured lighting and office equipment energy consumption levels in selected office areas of WERC were used as input parameters for this simulation. According to the baseline building model, the simulation site energy consumption level was estimated to be 36.9 $\mathrm{kBtu/ft^2yr}$ (10.8 $\mathrm{kWh/ft^2yr}$). In the baseline building model lighting system energy consumption was estimated at 3.1 $\mathrm{kWh/ft^2yr}$ based on the WERC measured data and office equipment energy consumption was estimated at 2.8 $\mathrm{kWh/ft^2yr}$. The baseline building model packaged variable volume, variable temperature HVAC system's simulated energy consumption was 4.92 $\mathrm{kWh/ft^2yr}$.

Based on the baseline model, an improved HVAC system was proposed that includes: implementation of decoupled HVAC system, improved efficiency of cooling system for air conditioning and ventilation system from 10 EER to 12.8 EER and air source heat pump application. In addition, the application of an air side economizer and a natural gas heat pump measures was simulated.

According to the independently estimated efficiency improvement for each HVAC component, the highest efficiency increase has a cooling system improvement from 10 EER to 12.8 EER that provided 17.2 % reduction in HVAC system energy consumption. The high energy efficiency increase also shows a decoupled HVAC system concept with forced air conditioning system (12% of HVAC baseline energy consumption savings) and air source heat pump (16.8% of HVAC baseline energy consumption savings). The least efficient measure was the economizer operation that provided just 2.2% of HVAC baseline energy consumption savings.

The simulated model of Valley Park office building HVAC system with natural gas heat pump has an electrical energy consumption of 0.868 $\mathrm{kWh/ft^2yr}$ and 0.081 $\mathrm{term/ft^2yr}$ of natural gas. Based on simulation results, the main advantage of natural gas heat pump system is reductions in the building electrical energy consumption by 82% and demand level by 56%.

The Valley Park office building simulation with an improved HVAC system showed a decrease of site energy consumption by 12.7 % compared with the baseline design or 27 % HVAC system energy consumption reduction. The building site energy consumption for this model decreased to $9.43\ \mathrm{kWh/ft^2yr}$ ($32.2\ \mathrm{kBtu/ft^2yr}$) with an HVAC system portion of $3.55\ \mathrm{kWh/ft^2yr}$.

Based on the Valley Park office building with an improved HVAC system simulation model, the displacement air conditioning system can provide an additional energy efficiency improvement when compared with forced air circulation conditioning. This improvement was estimated at $2.8\ \mathrm{kWh/ft^2yr}$, which reduces the entire building energy consumption level to $8.68\ \mathrm{kWh/ft^2yr}$ ($29.6\ \mathrm{kBtu/ft^2yr}$).

To estimate the maximum energy efficiency potential, a theoretical model for Valley Park office building energy consumption was developed. All simulations were performed on a daily basis with all systems operating at a maximum efficiency level. The lighting system energy performance was estimated based on an ideal “white” light source from a black body with a temperature of 5800 K and truncated to 400–700 nm. The estimated lighting system energy consumption decreased to $0.65\ \mathrm{kWh/ft^2yr}$. Based on a series of assumptions, the office equipment energy consumption was estimated at $0.26\ \mathrm{kWh/ft^2yr}$.

A proposed idealized model of the Valley Park office building HVAC system includes DOAS and a displacement air cooling system mounted on the ceiling and a hydronic heating system mounted at the floor level. All cooling and heating systems operate at the maximum Carnot cycle efficiency. A water cooled system is used for cooling system condenser heat rejection. The ideal energy recovery system is used to recover energy from the building exhaust air flow to the supplied outside air flow.

According to the estimation, the theoretical minimum energy consumption for Valley Park office building is $1.4\ \mathrm{kWh/ft^2yr}$. Based on this analysis, the energy efficiency for Valley Park office building with an improved HVAC system (including displacement ventilation) was estimated at 16 %.

7.2. Recommendations

The energy reduction potential for WERC was estimated based on the measured lighting and office equipment energy consumption levels in selected office areas. The implementation of energy saving measures will require a further detailed investigation of the AHU supply air fans and the operation and control sequence of the VAV boxes.

The Valley Park office building energy consumption model included a condenser air cooling system. Comparisons of the simulated building model with the theoretical minimum energy consumption model shows a big potential in improvement of energy efficiency in commercial buildings. Further studies of energy efficiency improvement potentials in commercial buildings should include condenser water cooling systems and hybrid geothermal heat pump systems.

REFERENCES

Ameen A., K. Mahmud. 2005. Desiccant dehumidification with hydronic radiant cooling system for air-conditioning applications in humid tropical climates. *ASHRAE Transactions* 111(2): 225-237

Arecchi A.V., T. Messadi, R.J. Koshel. 2007. *Field guide to illumination*. SPIE field guides. Volume FG II. Bellingham, WA: SPIE Press. 139p.

ASHRAE 2004. *ANSI/ASHRAE/IESNA standard 62.1-2004: Ventilation for acceptable indoor air quality*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE 2007. *ANSI/ASHRAE/IESNA standard 90.1-2007: Energy standard for buildings except low-rise residential buildings. I-P edition.* Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE 2008. *ASHRAE handbook: HVAC systems and equipment.* Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE 2009. *ASHRAE handbook: Fundamentals.* Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Bahman, A., W. Chakroun, N. Ghaddar, R. Saade, K. Ghai, 2009. Performance comparison of conventional and chilled ceiling/displacement ventilation systems in Kuwait. *ASHRAE Transactions* 115(1):587-594

Besant, R., C.J. Simonson. 2000. Air-to-air energy recovery. *ASHRAE Journal* 42(5): 31-38

Bradley, D.E., D.M. Utzinger. 2007. Enhanced and use of combined simulation tools in the assessment of hybrid natural/mechanical ventilation systems. *ASHRAE Transactions* 113(2): 144-153

Brandemehl, M.J., T. Katejanekarn. 2004. Dehumidification characteristics of commercial building applications. *ASHRAE Transactions* 110(2): 65-76

Braun, J.E., S.A. Klein, J.W. Mitchell. 1989. Effectiveness models for cooling towers and cooling coils. *ASHRAE Transactions* 95(2): 164-174

Cabeza, L., M. Ibanez, C. Sole, J. Roca, M. Nogues, S. Hiebler, H. Mehling. 2006. Use of phase-change materials in solar domestic hot water tanks. *ASHRAE Transactions* 112(1): 495-508

Carrier 2010. *The Centurion 48/50PD advanced product data.* http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48pd-01apd.pdf. Accessed: 07.16.2010

Castro, J., A. Oliva, C.D. Pérez-Segarra, C. Oliet. 2008. Developments in the design of a new air-cooled, hot-water-driven H_2O-LiBr Absorption Chiller. *ASHRAE Transactions* 114(1): 288-289

Chen, K. 1999. *Energy management in illuminating systems.* New York: CRC Press. 158p.

Colpan, C.O., I. Dincer, F. Hamdullahpur. 2008. Exergy analysis of SOFC-based cogeneration system for buildings. *ASHRAE Transactions* 114(1):108-115

COMcheck. 2010. *Technical Support Document for Version 3.8.0 of the COMcheck Software.* http://www.energycodes.gov/comcheck/documents/comcheck_tsd_380.pdf. Accessed: 08/03/2010

Conroy, C.L., S.A. Mumma. 2001. Ceiling radiant cooling panels as a viable distributed parallel sensible cooling technology integrated with dedicated outdoor air systems. *ASHRAE Transactions* 107(1): 578-585

Dieckmann, J., K. McKenney, J. Brodrick. 2009. Emerging technologies: Energy-efficient dehumidification. *ASHRAE Journal* 51(8): 78-80

Dieckmann, J., K.W. Roth, J. Brodrick. 2004. Emerging technologies: Radiant ceiling cooling. *ASHRAE Journal* 46(6): 42-43

Dieckmann, J., K.W. Roth, J. Brodrick. 2003. Emerging technologies: Air-to-air energy recovery heat exchangers. *ASHRAE Journal* 45(8): 57-58

Durkin, T.H., J.B. Rishel. 2003. Dedicated heat recovery. *ASHRAE Journal* 45(10): 18-23

EIA 2009. *Annual energy review.* U.S. Energy Information Administration. http://www.eia.doe.gov/emeu/aer/pdf/aer.pdf. Accessed: 5/10/2010

EIA 2010. *Natural gas weekly update. Overview (For the Week Ending Wednesday, September 1, 2010).* U.S. Energy Information Administration. http://www.eia.doe.gov/oog/info/ngw/ngupdate.asp#Prices. Accessed: 08/07/2010

El Mankibi, M. 2009. Hybrid ventilation control design and management. *ASHRAE Transactions* 2009(1): 3-9

Farzam, S., G. Todesco. 2010. Sustainability in cold climates. *ASHRAE Journal* 52(1): 20-28

Ghaddar, N., R. Saadeh, K. Ghali, A. Keblawi. 2008. Design charts for combined chilled ceiling displacement ventilation system. *ASHRAE Transactions* 114(2): 574-587

Google Maps. 2010. *Digital globe. Map data.* http://maps.google.com/maps?f=s&utm_campaign=en&utm_source=en-ha-na-us-bk-gm&utm_medium=ha&utm_term=google%20map. Accessed: 7/10/2010

Haberl, J.S., S. Cho. 2004. *Literature review of uncertainty of analysis methods (DOE-2 program).* ESL-TR-04/11-01. College Station: Energy Systems Laboratory, Texas A&M University. http://repository.tamu.edu/handle/1969.1/2072. Accessed: 04/05/2010

Hackel, A., G. Nellis, S. Klein. 2009. Optimization of cooling-dominated hybrid ground-coupled heat pump systems. *ASHRAE Transactions* 115(1): 565-580

Huang, Y. J. 2000. *DrawBDL version 3.0*. Joe Haung and Association. http://www.drawbdl.com/. Accessed: 05/24/2010

IECC 2009. *International energy conservation code*, Falls Church, VA: International Code Council, Inc.

IES Lighting Handbook. 2000. *The IESNA lighting handbook. Reference and application.* Ninth edition. New York: Illuminating Engineering Society of North America

Jarnagin, R.E. 2010. *Heat recovery from air conditioning units. Fact sheet EES-26* http://www.wec.ufl.edu/extension/gc/harmony/documents/EH126.pdf. Accessed: 05/24/2010

Jeong J.W., S.A. Mumma. 2006. Designing a dedicated outdoor air system with ceiling radiant cooling panels. *ASHRAE Journal* 48(10): 56-66

Kilkis, B. 2006. From heating to hybrid HVAC panel - a trial of exergy-efficient innovations. *ASHRAE Transactions* 112(1): 343-349

Klein, S.A. 1973. *TRNSYS - A transient simulation program. Rept. 3*. Madison, WI: Solar Energy Laboratory, University of Wisconsin-Madison

Komor, P. 1997. Space cooling demands from office plug loads. *ASHRAE Journal* 39(12): 41-44

Landsberg, D., D.K. McLellan, C. Kurkjian. 2008. Geothermal heat rejection systems for data centers. *ASHRAE Transactions* 114(1): 37-43

Larsen, T.S., R.H.W. Bindels, L. Michalak, M. Milewski, P.V. Nielsen. 2007. Air distribution in rooms with a fan-driven convector. *ASHRAE Transactions* 113(1): 342-357

LASL 1980. *DOE-2 reference manual. Part 1. Version 2.1*. Los Alamos, NM: Los Alamos Scientific Laboratory

LBNL 1993a. *DOE-2 BDL summary. Version 2.1E. LBL-34946.* Berkeley, CA: Lawrence Berkeley National Laboratory

LBNL 1993b. *DOE-2 supplement. Version 2.1E.* Berkeley, CA: Lawrence Berkeley National Laboratory

Liebel, B., J. Brodrick 2005. Emerging technologies: Squeezing the watts out of fluorescent lighting. *ASHREA Journal* 47(11):52-54

Liesen, R.J. Variable heat recovery in double bundle electric chiller. *SimBuild 2004, IBPSA - USA national conference*, Boulder, CO, August 4-6, 2004, http://simulationresearch.lbl.gov/dirpubs/SB04/sb03.pdf. Accessed: 04.24.2010

Liu, M. D.E. Claridge, N. Bensouda, K. Heinemeier, S. Uk Lee, G. Wie. 2003. *High performance commercial building systems. Manual of procedures for calibrating simulations of building systems. Element 5. Project 2.3.* Berkeley, CA: California Energy Commission. Lawrence Berkeley National Laboratory

Liu, M., D.E. Claridge, W.D. Turner. 2002. *Continuous Commissioning^{sm} guidebook: Maximizing building energy efficiency and comfort.* Federal Energy Management Program (FEMP), U.S. Department of Energy, http://www.eere.energy.gov/femp/pdfs/ccg01_covers.pdf. Accessed: 04.24.2010

Loudermilk, K. 2005. Displacement with induction: conditioning our classrooms in accordance with ANSI/ASA S12.60. *ASHRAE Transactions* 111(1): 740-747

Loudermilk, K. 2009. Designing chilled beans for thermal comfort. *ASHRAE Journal* 51(10): 58-64

Malhotra, M. 2009. *An analysis of off-grid, off-pipe housing in six U.S. climates.* Ph.D. Dissertation; College Station: Texas A&M University

McCluney, W.R. 1994. *Introduction to radiometry and photometry*. Boston: Artech House. 404p.

McDougall, T., K. Nordmeyer, C. Klaassen. 2006. Low-energy building case study: IAMU office and training headquarters. *ASHRAE Transactions* 112(1): 312-320.

McQuay International. 2002. *Condenser water heat recovery.* http://www.mcquay.com/mcquaybiz/literature/lit_systems/Flyers/CondWater-HeatRec.pdf. Accessed: 04.09.2010

Mercer, K., J. Braun. 2005. Evaluation of a ventilation heat pump for small commercial buildings. *ASHRAE Transactions* 111(1): 890-900

Morrow, A., P. Savarino. 2006. *UTC Power Pure Comfort 240 Ritz-Carlton San Francisco.* Oak Ridge, TN: Oak Ridge National Laboratory. http://www.ms.ornl.gov/maw06/pdfs/presentations/Day2/Morrow.pdf. Accessed: 2/4/2010

Mossolly, M., K.Ghali, N. Ghaddar, L. Jensen. 2008 Optimized operation of combined chilled ceiling displacement ventilation system using genetic algorithm. *ASHRAE Transactions* 114(2): 541-554

Mumma S. A. 2009. 30% Surplus OA. Does it use more energy? *ASHRAE Journal* 51(6): 24-36

Mumma S. A. 2010. *Dedicated outdoor air systems and building pressurization*. http://doas-radiant.psu.edu/DOAS_Pressurization_Paper.pdf. Accessed: 5/12/2010

Mumma, S.A., J.W. Jeong. 2005. Field experience controlling a dedicated outdoor air system. *ASHRAE Transactions* 111(2): 433-442

Mushtaq, A. 2003. *Systematic time-based study for quantifying the uncertainty of uncalibrated models in building energy simulations.* M.S. Thesis; College Station: Texas A&M University

NRC 2010. *Energy Star® Qualified office equipment. Printers.* Natural Recourses Canada. Office of Energy Efficiency. http://oee.nrcan.gc.ca/residential/personal/office/printers.cfm?attr=4. Accessed: 08/20/2010

O'Neal, D. L., J. A. Gonzalez, W. Aldred, 1994. A simplified procedure for sizing vertical ground coupled heat pump heat exchangers for residences in Texas, *Proceedings of the 9th Symposium on Improving Building Systems in Hot and Humid Climates.* May 19-20, 1994. Arlington, TX. http://repository.tamu.edu/handle/1969.1/6648. Accessed: 2/10/2010

Olesen, B., M. de Cari, M. Scarpa, M. Koschez. 2006. Dynamic evaluation of the cooling capacity of thermo-activated building systems. *ASHRAE Transactions* 112(1): 350-357

Panasonic 2010. *Panasonic spiral fluorescent ceiling lamp EcoNavi used for residential lighting*. (In Japanese) http://panasonic.co.jp/corp/news/official.data/data.dir/jn100609-1/jn100609-1.html. Accessed: 08/16/2010

Petrov, A. Y., J. B. Berry, A. Zaltash. 2006. Commercial integrated energy systems provide data that advance combined cooling, heating, and power. IMECE 2006-14932. *Proceeding of the 2006 ASME International Mechanical Engineering Congress and Exposition.* November 5-10, 2006. Chicago, IL. http://www.ornl.gov/sci/engineering_science_technology/eere_research_reports/der_chp/ies/gas_turbine_chp/imece2006_14932/imece2006_14932.pdf. Accessed: 3/15/2010

Phillips, D., M. Beyers, J.Good. 2009. How high can you go? *ASHRAE Journal* 51(9): 26-36

Pless, S., P. Torchellini, J. Petersen. 2006. Energy performance evaluation of a low-energy academic building. *ASHRAE Transactions* 112(1): 295-311

Plesser, S., M.N. Fisch, E. Gruber, B. Schlomann. 2008. *Monitoring electricity consumption in the tertiary sector- a project within the Intelligent Energy Europe program*. http://repository.tamu.edu/handle/1969.1/90787. Accessed: 7/22/2010

Ramamoorthy, M. H. Jin, A. Chiasson, J.D. Spitler. 2001. Optimal sizing of hybrid ground-source heat pump systems that use a cooling pond as a supplemental heat rejecter – A System simulation approach. *ASHRAE Transactions* 107(1): 26-38

Rees, S.J., P. Haves. 2001. *A nodal model for displacement ventilation and chilled ceiling systems in office spaces*. http://www.inive.org/members_area/medias/pdf/Inive%5CIBPSA%5CUFSC729.pdf. Accessed: 1/25/2010

Reilly, S., T.S. Kraft, V. Olgyay. 2007. Jack Evans police headquarters: 24-hour low energy use. *ASHRAE Transactions* 113(1): 51-55

RISO 2010. *Power consumption: the hidden costs of copiers and printers*. https://us.riso.com/pls/portal/docs/page/riso07/customer%20support/information%20resources/whitepapers/power_consumption_white_paper.pdf. Accessed: 08/20/2010

RMI 2010. *Rocky Mountain Institute. Buildings.* http://www.rmi.org/rmi/Buildings. Accessed: 06/15/2010

RSMeans Electrical cost data. 30th annual edition. 2007. Kingston, MA: Construction Publishers & Consultants. 566p.

RSMeans Mechanical cost data. 30th annual edition. 2007. Kingston, MA: Construction Publishers & Consultants. 728p.

Samsung 2010. *23" Touch of color HDTV enabled LCD monitor*. http://www.samsung.com/us/computer/monitors/LS23ELNKF/ZA-features. Accessed: 08/20/2010

Schweigler, C., S. Hiebler, C. Keil, H. Köbel, C. Kren, H. Mehling. 2007. Low-temperature heat storage for solar heating and cooling applications. *ASHRAE Transactions* 113(1): 89-96

Sekhar S.C. 2007. A review of ventilation and air-conditioning technologies for energy-efficient healthy buildings in tropics. *ASHRAE Transactions* 113(1):426-434

Sekhar, S. C., K. W. Tham, U. Maheswaran, K. W. Cheong. 2004. Development of energy efficient single coil twin fan air-conditioning system with zonal ventilation control. *ASHRAE Transactions* 110(2): 204-217

Sekine K., R. Ooka, M. Yokoi, Y. Shiba, S. Hwang. 2007. Development of a ground-source heat pump system with ground heat exchanger utilizing the cast-in-place concrete pile foundations of building. *ASHRAE Transactions* 113(1): 558-566

Shapiro, I. 2009 Energy audits in large commercial office buildings. *ASHRAE Journal* 51(1): 18-27

Simmonds, P., I. Chambers, B. Meholomakulu, C. Simmonds. 2006a. Applied performance of radiant ceiling panels for cooling. *ASHRAE Transactions* 112(1): 368-376

Simmonds, P., B. Mehlomakulu, T. Ebert. 2006b. Radiant cooled floors- operation and control dependant upon solar radiation. *ASHRAE Transactions* 112(1): 358-367

Steinbock, J., D. Eijadi, T. McDougall. 2007. Net zero energy building case study: Science House. *ASHRAE Transactions* 113(1): 26-35

Takahashi, H., 2006. *GASEX2006 Country report – city gas industry in Japan.* July 25, 2006. http://www.gas.or.jp/english/letter/images/06/pdf/04-japan_country_report.pdf. Accessed: 04/14/2010

Texas A&M Utilities Rate 2010. *Utility rates*. Physical Plant Department. Texas A&M University.

http://utilities.tamu.edu/index.php?option=com_docman&task=doc_download&gid=9&Itemid=. Accessed: on 07/14/2010

Tsay, Y.S., S. Kato, R. Ooka, M. Koganei, N.Shoda. 2006. Study on the applicability of combining a desiccant cooling system with a heat pump in a hot and humid climate. *ASHRAE Transactions* 112(1): 189-194

U.S. DOE 2001. *Assessment of hybrid geothermal heat pump systems. DOE/EE-0258.* New technology demonstration program. Federal Energy Management Program. U.S. Department of Energy. http://www1.eere.energy.gov/femp/pdfs/hyhgp_tir.pdf. Accessed: 6/4/2010

U.S. DOE 2009. *Buildings energy data book.* Washington, DC: Office of Energy Efficiency and Renewable Energy. U.S. Department of Energy. http://buildingsdatabook.eren.doe.gov/docs%5CDataBooks%5C2009_BEDB_Updated.pdf. Accessed: 5/10/2010

U.S. DOE 2010a. *Building energy software tools directory.* Building technologies program. Energy efficiency and renewable energy. U.S. Department of Energy. http://www.eere.energy.gov/buildings/tools_directory/. Accessed: 04/22/2010

U.S. DOE 2010b. *Building energy software tools directory. DOE-2.* Building technologies program. Energy efficiency and renewable energy. U.S. Department of Energy. http://apps1.eere.energy.gov/buildings/tools_directory/software.cfm/ID=34/pagename_submenu=energy_simulation/pagename_menu=whole_building_analysis/pagename=subjects. Accessed: 04/05/2010

U.S. DOE 2010c. *Contrasting the capabilities of building energy performance simulation programs.* U.S. Department of Energy. http://apps1.eere.energy.gov/buildings/tools_directory/pdfs/contrasting_the_capabilities_of_building_energy_performance_simulation_programs_v1.0.pdf. Accessed: 03/05/2010

U.S. DOE 2010d. *Typical Meteorological Year. Version 3.* U.S. Department of Energy. http://doe2.com/Download/Weather/TMY3/States_O-W/TX_College_Station_Easter.bin. Accessed: 04/05/2010

U.S. DOE 2010e. *How to buy an energy-efficient fluorescent tube lamp*. Federal Energy Management Program. Energy Efficiency and Renewable Energy. U.S. Department of Energy. http://www1.eere.energy.gov/femp/procurement/eep_fluortube_lamp.html. Accessed: 08/16/2010

USA Today 2009. Brighter skies for solar. *ASHRAE Journal* 51(2): 13

Visual DOE 4.0 software. 2005. San Francisco, CA: Architectural Energy Cooperation. http://www.archenergy.com/. Accessed: 04/15/2010

Wagner, T.C., N.P. Leslie, R.S. Sweetser, T.K. Stovall. 2009. Combined heating and power using microturbines in a major urban hotel. *ASHRAE Transactions* 115(1): 208-219

Walker, C.E., L.R. Glicksman, L.K. Norfold. 2007. Tale of two low-energy designs: Comparison of mechanically and naturally ventilated office buildings in temperate climates. *ASHRAE Transactions* 113(1): 36-50

Weidner, S., J. Doerger, M. Walsh. 2009. Cooling with less air. *ASHRAE Journal* 51(12): 34-40

Wilkins, C., M.H. Hosni. 2000. Heat gain from office equipment. *ASHRAE Journal* 42(6): 33-39

Williamson, S. H. 2010. *Seven ways to compute the relative value of a U.S. dollar amount, 1774 to present, measuring worth.* April 2010. http://www.measuringworth.com/uscompare/. Accessed: 07/14/2010

Xia, Y. and S.A. Mumma. 2006. Ceiling radiant cooling panels employing heat-conducting rails: deriving the governing heat transfer equations. *ASHRAE Transactions* 112(1): 34-41

Xu, P., J. Huang, R. Jin, G. Yang. 2007. Measured energy performance of a US-China demonstration energy-efficient office building. *ASHRAE Transactions* 113(1): 56-64

Yahagi, M., K. Imai, Y. Taniguchi. 2006. Development of high-efficient gas engine heat pumps. *23rd World Gas Conference*, Amsterdam, The Netherlands, June 2006. www.igu.org/html/wgc2006/pdf/paper/add10722.pdf. Accessed: 03/11/2010

Yavuzturk, C., J. D. Spitler. 2000. Comparative study of operating and control strategies for hybrid ground-source heat pump systems using a short time step simulation model. *ASHRAE Transactions* 106(2): 192-209

Yuan X., Q. Chen, L. R. Glicksman. 1998. A critical review of displacement ventilation. *ASHRAE Transactions* 104(1A): 78-90

Zaltash, A., E. Vineyard, R. Linkous. 2008. Laboratory evaluation: Performance of a 10 RT gas engine-driven heat pump. *ASHRAE Transactions* 114(2): 224-231

APPENDIX A

DOE-2.1E WISENBAKER ENGINEERING RESEARCH CENTER

CALIBRATED SIMULATION MODEL

```
INPUT LOADS  ..

TITLE           LINE-1 * Wisenbaker ERC, College Station  *
                LINE-2 *-Calibrated Model-*
                LINE-3 *DOE-2.1E *
                LINE-4 *Custom weighting factors*
                LINE-5 * Oleksandr Tanskyi *    ..

                RUN-PERIOD         JAN 1 2008 THRU DEC 31 2008     ..
                ABORT              ERRORS                          ..
                DIAGNOSTIC         WARNINGS  NO-ECHO    NARROW     ..
                LOADS-REPORT       SUMMARY =(LS-B, LS-F, LS-D)
                                   VERIFICATION = (LV-B,LV-D)      ..
                BUILDING-LOCATION  LATITUDE=30.61  LONGITUDE=96.3
                                   ALTITUDE=325
                                   TIME-ZONE=6  AZIMUTH=-135.0     ..


$                            BUILDING DESCRIPTION

$                       CONSTRUCTION AND GLASS-TYPES

ROO-1  = LAYERS = MAT =(RG01,IN72,CB12,GP01) INSIDE-FILM-RES = 0.61  ..
$ RG0l - 1/2 inch Roof Gravel or Slag R=0.05
$ IN72 - 1 inch Roof Insulation, Preformed R=2.78
$ CB12 – Concrete Filled Concrete Block, 8 inch Heavy Weight R=0.88
$ GP01 - 1/2 inch Gypsum or Plaster Board R= 0.45

WA-1-2 =LAYERS =MAT = (BK02,IN01,GP01)  ..
$ BK02 - 8 inch Common Brick R= 1.60
$ IN01 - Batt, R-7+ Mineral Wool/Fiber R= 7.53
$ GP01 - 1/2 inch Gypsum or Plaster Board R= 0.45

WA-LAB =LAYERS =MAT= (AS01,HB01,IN21,HB01,AS01)  ..
$ AS01 - Aluminum or Steel Siding R=0
$ HB01 - Hard Board, 3/4 inch R= 1.15
$ IN21 - Preformed Mineral Board R= 3.04
$ HB01 - Hard Board, 3/4 inch R= 1.15
$ AS01 - Aluminum or Steel Siding R=0

UND-W1 =LAYERS =MAT=(GP05,IN42,CB12)   ..
$ GP05 -  1 inch Light Weight Aggregate Gypsum Plaster R= 0.63
$ IN42 -  3/4 inch Polyurethane, Expanded R= 4.67
$ CB12 – Hollow Concrete Block, 8 inch Heavy Weight R=0.88
```

```
FL1 = LAYERS =MAT=(AV01,GP05,CB12,GP05)   ..
$ AV01 - Asbestos-Vinyl Tile R= 0.05
$ GP05 - 1 inch Light Weight Aggregate Gypsum Plaster R= 0.63
$ CB12 - Concrete Filled Concrete Block, 8 inch Heavy Weight R=0.88
$ GP05 - 1 inch Light Weight Aggregate Gypsum Plaster R= 0.63

WALL-1         =CONSTRUCTION       LAYERS=WA-1-2   ..
ROOF-1         =CONSTRUCTION       LAYERS=ROO-1    ..
FLOOR-1        =CONSTRUCTION       LAYERS=FL1      ..
WALL-U         =CONSTRUCTION       LAYERS=UND-W1   ..
LAB-1          =CONSTRUCTION       LAYERS=WA-LAB   ..

W-1            =GLASS-TYPE       GLASS-TYPE-CODE = 9  PANES = 1    ..
                                 $ SINGLE PANE WITH REFLECTIVE COATING
DOORS          =GLASS-TYPE         GLASS-TYPE-CODE = 5        ..
                                 $ SINGLE PANE UNCOATED

$                          BUILDING SHADES

      BD11 = BUILDING-SHADE
      X = 60  Y = 145  Z = 0
      HEIGHT = 40
      WIDTH = 40
      AZIMUTH = 180
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD12 = BUILDING-SHADE
      X = 60  Y = 145  Z = 0
      HEIGHT = 40
      WIDTH = 120
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD13 = BUILDING-SHADE
      X = 100  Y = 145  Z = 0
      HEIGHT = 40
      WIDTH = 120
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD21 = BUILDING-SHADE
      X = 195  Y = 270  Z = 0
      HEIGHT = 40
      WIDTH = 70
      AZIMUTH = 180
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..
```

```
      BD22 = BUILDING-SHADE
      X = 195  Y = 270  Z = 0
      HEIGHT = 40
      WIDTH = 115
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD23 = BUILDING-SHADE
      X = 265  Y = 270  Z = 0
      HEIGHT = 40
      WIDTH = 115
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD31 = BUILDING-SHADE
      X = 330  Y = 150  Z = 0
      HEIGHT = 40
      WIDTH = 70
      AZIMUTH = 180
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD32 = BUILDING-SHADE
      X = 330  Y = 150  Z = 0
      HEIGHT = 40
      WIDTH = 250
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BD33 = BUILDING-SHADE
      X = 400  Y = 150  Z = 0
      HEIGHT = 40
      WIDTH = 250
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

$                              SHADING SCHEDULE

    B-SH-1 = SCHEDULE            THRU DEC 31 (ALL) (1,24)  (1)   ..

$                              OCCUPANCY SCHEDULE

OC-1          =DAY-SCHEDULE       (1,5)   (0.05)
                                  (6,7)   (0.25)
```

```
                                        (8,11)  (1.0)
                                        (12,14) (0.8,0.6,0.8)
                                        (15,18) (1.0)
                                        (19,21) (0.5,0.1,0.1)
                                        (22,24) (0.05)         ..

OC-2          =DAY-SCHEDULE             (1,8) (0.05)
                                        (9,17) (0.10)
                                        (18,24) (0.05)         ..

OC-WEEK       =WEEK-SCHEDULE            (WD) OC-1 (WEH) OC-2   ..

OCCUPY-1      =SCHEDULE                 THRU DEC 31 OC-WEEK    ..

$                                   LIGHTING SCHEDULE

LT-1          =DAY-SCHEDULE             (1,4) (0.702, 0.686, 0.673, 0.665)
                                        (5,8) (0.660, 0.657, 0.657, 0.671)
                                        (9,12) (0.720, 0.849, 0.915, 0.956)
                                        (13,16) (0.973, 0.965, 0.984, 1.00)
                                        (17,20) (1.00, 0.976, 0.873, 0.796)
                                        (21,24) (0.755,0.737,0.728,0.718)
..
LT-2          =DAY-SCHEDULE             (1,4) (0.681,0.669, 0.662, 0.656)
                                        (5,8) (0.649, 0.646, 0.647, 0.649)
                                        (9,12) (0.649, 0.654, 0.667, 0.685)
                                        (13,16) (0.698, 0.706, 0.719, 0.729)
                                        (17,20) (0.732, 0.729, 0.720, 0.707)
                                        (21,24) (0.696, 0.690, 0.685, 0.681)
..
LT-3          =DAY-SCHEDULE             (1,4) (0.673, 0.662, 0.651, 0.644)
                                        (5,8) (0.644, 0.641, 0.639, 0.638)
                                        (9,12) (0.636, 0.641, 0.651
      0.668)
                                        (13,16) (0.681, 0.692, 0.708, 0.722)
                                        (17,20) (0.727, 0.729, 0.724, 0.714)
                                        (21,24) (0.707, 0.700, 0.698, 0.687)
..
LT-4          =DAY-SCHEDULE             (1,4) (0.671, 0.662, 0.658, 0.650)
                                        (5,8) (0.647, 0.644, 0.642, 0.642)
                                        (9,12) (0.649, 0.671, 0.699, 0.721)
                                        (13,16) (0.734, 0.733, 0.735, 0.752)
                                        (17,20) (0.755, 0.744, 0.723, 0.700)
                                        (21,24) (0.686, 0.677, 0.673, 0.668)
..

LT-WEEK      =WEEK-SCHEDULE             (MON,FRI) LT-1
                                        (SAT) LT-2
                                        (SUN) LT-3
                                        (HOL) LT-4  ..
LIGHTS-1      =SCHEDULE                 THRU DEC 31 LT-WEEK  ..
```

```
$                          OFFICE EQUIPMENT SCHEDULE

EQ-1            =DAY-SCHEDULE       (1,4) (0.702, 0.686, 0.673, 0.665)
                                    (5,8) (0.660, 0.657, 0.657, 0.671)
                                    (9,12) (0.720, 0.849, 0.915, 0.956)
                                    (13,16) (0.973, 0.965, 0.984, 1.00)
                                    (17,20) (1.00, 0.976, 0.873, 0.796)
                                    (21,24) (0.755,0.737,0.728,0.718)
..
EQ-2            =DAY-SCHEDULE       (1,4) (0.681,0.669, 0.662, 0.656)
                                    (5,8) (0.649, 0.646, 0.647, 0.649)
                                    (9,12) (0.649, 0.654, 0.667, 0.685)
                                    (13,16) (0.698, 0.706, 0.719, 0.729)
                                    (17,20) (0.732, 0.729, 0.720, 0.707)
                                    (21,24) (0.696, 0.690, 0.685, 0.681)
..
EQ-3            =DAY-SCHEDULE       (1,4) (0.673, 0.662, 0.651, 0.644)
                                    (5,8) (0.644, 0.641, 0.639, 0.638)
                                    (9,12) (0.636, 0.641, 0.651
      0.668)
                                    (13,16) (0.681, 0.692, 0.708, 0.722)
                                    (17,20) (0.727, 0.729, 0.724, 0.714)
                                    (21,24) (0.707, 0.700, 0.698, 0.687)
..
EQ-4            =DAY-SCHEDULE       (1,4) (0.671, 0.662, 0.658, 0.650)
                                    (5,8) (0.647, 0.644, 0.642, 0.642)
                                    (9,12) (0.649, 0.671, 0.699, 0.721)
                                    (13,16) (0.734, 0.733, 0.735, 0.752)
                                    (17,20) (0.755, 0.744, 0.723, 0.700)
                                    (21,24) (0.686, 0.677, 0.673, 0.668)
..

EQ-WEEK         =WEEK-SCHEDULE      (MON,FRI) EQ-1
                                    (SAT) EQ-2
                                    (SUN) EQ-3
                                    (HOL) EQ-4  ..
EQUIP-1         =SCHEDULE           THRU DEC 31 EQ-WEEK  ..

                  $ INFILTRATION SCHEDULE

INFIL-SCH1       =SCHEDULE           THRU DEC 31 (ALL) (1,24) (0)  ..
INFIL-SCH2       =SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..

                        $ SET DEFAULT VALUES

             SET-DEFAULT FOR SPACE  FLOOR-WEIGHT=0  ..
             SET-DEFAULT FOR EXTERIOR-WALL  CONSTRUCTION=WALL-1  ..
             SET-DEFAULT FOR UNDERGROUND-WALL CONSTRUCTION=WALL-U ..
             SET-DEFAULT FOR WINDOW  HEIGHT=4.0  GLASS-TYPE=W-1  Y=3 ..

                        $ GENERAL SPACE DEFINITION

OFFICE       =SPACE-CONDITIONS   TEMPERATURE = (72)
              PEOPLE-SCHEDULE      =OCCUPY-1
```

```
              AREA/PERSON          =200
              PEOPLE-HG-LAT        =200
              PEOPLE-HG-SENS       =250
$ Moderately active office work
              LIGHTING-SCHEDULE    =LIGHTS-1
              LIGHTING-TYPE        =REC-FLUOR-RV
              LIGHT-TO-SPACE       =0.8
              LIGHTING-W/SQFT      =1.2
              EQUIP-SCHEDULE       =EQUIP-1
              EQUIPMENT-W/SQFT     =0.8
              INF-METHOD           =AIR-CHANGE
              AIR-CHANGES/HR       =0.0
$ Assuming pressurization
              INF-SCHEDULE         =INFIL-SCH1  ..

LAB           =SPACE-CONDITIONS    TEMPERATURE = (72)
              PEOPLE-SCHEDULE      =OCCUPY-1
              NUMBER-OF-PEOPLE     =10
              PEOPLE-HEAT-GAIN     =800       $ Light physical work
              LIGHTING-SCHEDULE    =LIGHTS-1
              LIGHTING-TYPE        =REC-FLUOR-NV
              LIGHT-TO-SPACE       =1.0
              LIGHTING-W/SQFT      =1.2
              EQUIP-SCHEDULE       =EQUIP-1
              EQUIPMENT-W/SQFT     =0.8
              INF-METHOD           =AIR-CHANGE
              AIR-CHANGES/HR       =2.0      $ The loading dock door is
                                             $ open most of the time
              INF-SCHEDULE         =INFIL-SCH2   ..

                          $   SPECIFIC SPACE DETAILS

$                                GROUND FLOOR

SPACE-F0       =SPACE            SPACE-CONDITIONS = OFFICE
                                 AREA = 39504  VOLUME = 440380   ..

   WALL-F0-F  =  EXTERIOR-WALL         HEIGHT = 10 WIDTH = 360
                                       X = 0 Y = 0 Z = -10
                                       AZIMUTH = 180               ..
         W-F0 =  WINDOW                WIDTH = 350  HEIGHT = 6
                                       X = 5   Y = 2  SETBACK = 2  ..
   WALL-F0-R  =  EXTERIOR-WALL         HEIGHT = 10 WIDTH = 120
                                       X = 360 Y = 0 Z = -10
                                       AZIMUTH = 90                ..
         DOOR-F0-R = WINDOW            HEIGHT = 7 WIDTH = 7
                                       X = 59 Y = 0
                                       GLASS-TYPE = DOORS
                                       SETBACK = 0                 ..
   WALL-F0-L  =  EXTERIOR-WALL         HEIGHT = 10 WIDTH = 120
                                       X = 0 Y = 120 Z = -10
                                       AZIMUTH = 270               ..
         DOOR-F0-L = WINDOW            HEIGHT = 7 WIDTH = 7
                                       X = 59 Y = 0
```

```
                                           GLASS-TYPE = DOORS
                                           SETBACK = 0                ..
   WALL-F0-B1-1  = EXTERIOR-WALL           HEIGHT = 5 WIDTH = 170
                                           X = 170 Y = 120 Z = -5
                                           AZIMUTH = 0                ..
   WALL-F0-B1-2  = UNDERGROUND-WALL        HEIGHT = 5 WIDTH = 170
                                           X = 170 Y = 120 Z = -10
                                           AZIMUTH = 0                ..
   WALL-F0-B2  = UNDERGROUND-WALL          HEIGHT = 10 WIDTH = 120
                                           X =290 Y = 120 Z = -10
                                           AZIMUTH = 0                ..
   WALL-F0-B3-1  = EXTERIOR-WALL           HEIGHT = 5 WIDTH = 70
                                           X =360 Y = 120 Z = -5
                                           AZIMUTH = 0                ..
   WALL-F0-B3-2  = UNDERGROUND-WALL        HEIGHT = 5 WIDTH = 70
                                           X =360 Y = 120 Z = -10
                                           AZIMUTH = 0                ..
   FLOOR-F0  = UNDERGROUND-FLOOR           HEIGHT=120  WIDTH=360
                                           X =0 Y = 120 Z = -10
                                           AZIMUTH = 180
                                           CONSTRUCTION = WALL-U
                                           TILT = 180                 ..

         $                               FLOOR 1

SPACE-F1       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 39504  VOLUME = 474048  ..

   WALL-F1-F  =  EXTERIOR-WALL             HEIGHT = 12 WIDTH = 360
                                           X = 0 Y = 0 Z = 0
                                           AZIMUTH = 180   ..
         W-F1 = WINDOW                     WIDTH = 350  HEIGHT = 6
                                           X = 5   Y = 2
                                           SETBACK = 2  ..
   WALL-F1-R  =  EXTERIOR-WALL             HEIGHT = 12 WIDTH = 120
                                           X = 360 Y = 0 Z = 0
                                           AZIMUTH = 90   ..
         W-F1-R  =WINDOW                   WIDTH = 40   X = 42   Y = 2
                                           SETBACK = 2    ..
   WALL-F1-L  =  EXTERIOR-WALL             HEIGHT = 12 WIDTH = 120
                                           X = 0 Y = 120 Z = 0
                                           AZIMUTH = 270   ..
         W-F1-L   =WINDOW                  WIDTH = 40
                                           X = 42   Y = 2
                                           SETBACK = 2    ..
   WALL-F1-B1  = EXTERIOR-WALL             HEIGHT = 12 WIDTH = 170
                                           X = 170 Y = 120 Z = 0
                                           AZIMUTH = 0   ..
         W-F1-B1  =WINDOW                  WIDTH = 160  HEIGHT = 5
                                           X = 5   Y = 2
                                           SETBACK = 2  ..
     WALL-F1-B3  = EXTERIOR-WALL           HEIGHT = 12 WIDTH = 70
                                           X =360 Y = 120 Z = 0
                                           AZIMUTH = 0     ..
```

```
      W-F1-B3  =WINDOW              WIDTH = 60  HEIGHT = 5
                                    X = 5   Y = 1
                                    SETBACK = 2  ..
   FLOOR-F1  = INTERIOR-WALL        HEIGHT=120  WIDTH=360
                                    X =0 Y = 120 Z = 0
                                    AZIMUTH = 180
                                    NEXT-TO SPACE-F0
                                    CONSTRUCTION = FLOOR-1
                                    TILT = 180         ..

  $                              FLOOR 2

SPACE-F2       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 39504  VOLUME = 474048  ..

   WALL-F2-F  =  EXTERIOR-WALL        HEIGHT = 12 WIDTH = 360
                                      X = 0 Y = 0 Z = 12
                                      AZIMUTH = 180  ..
        W-F2 = WINDOW                 WIDTH = 350  HEIGHT = 6
                                      X = 5   Y = 2
                                      SETBACK = 2  ..
   WALL-F2-R  =  EXTERIOR-WALL        HEIGHT = 12 WIDTH = 120
                                      X = 360 Y = 0 Z = 12
                                      AZIMUTH = 90   ..
        W-F2-R = WINDOW               WIDTH = 40
                                      X = 42   Y = 2
                                      SETBACK = 2    ..
   WALL-F2-L  =  EXTERIOR-WALL        HEIGHT = 12 WIDTH = 120
                                      X = 0 Y = 120 Z = 12
                                      AZIMUTH = 270   ..
        W-F2-L =WINDOW                WIDTH = 40
                                      X = 42   Y = 2
                                      SETBACK = 2    ..
   WALL-F2-B1  = EXTERIOR-WALL        HEIGHT = 12 WIDTH = 170
                                      X = 170 Y = 120 Z = 12
                                      AZIMUTH = 0   ..
        W-F2-B1  =WINDOW              WIDTH = 160  HEIGHT = 5
                                      X = 5   Y = 2
                                      SETBACK = 2  ..
      WALL-F2-B2  = EXTERIOR-WALL     HEIGHT = 12 WIDTH = 80
                                      X =250 Y = 120 Z = 12
                                      AZIMUTH = 0  ..
   WALL-F2-B3  = EXTERIOR-WALL        HEIGHT = 12 WIDTH = 70
                                      X =360 Y = 120 Z = 12
                                      AZIMUTH = 0     ..
        W-F2-B3  = WINDOW             WIDTH = 60  HEIGHT = 5
                                      X = 5   Y = 1
                                      SETBACK = 2  ..
   FLOOR-F2  = INTERIOR-WALL          HEIGHT=120  WIDTH=360
                                      X =0 Y = 120 Z = 12
                                      AZIMUTH = 180
                                      NEXT-TO SPACE-F1
                                      CONSTRUCTION = FLOOR-1
                                      TILT = 180         ..
```

```
$                                FLOOR 3

SPACE-F3       =SPACE            SPACE-CONDITIONS = OFFICE
                                 AREA = 39504  VOLUME = 474048 ..

      WALL-F3-F  =  EXTERIOR-WALL    HEIGHT = 12 WIDTH = 360
                                     X = 0 Y = 0 Z = 24
                                     AZIMUTH = 180  ..
           W-F3 = WINDOW     WIDTH = 350  HEIGHT = 6
                                     X = 5   Y = 2
                                     SETBACK = 2  ..
      WALL-F3-R  =  EXTERIOR-WALL    HEIGHT = 12 WIDTH = 120
                                     X = 360 Y = 0 Z = 24
                                     AZIMUTH = 90   ..
                 W-F3-R  =WINDOW     WIDTH = 40
                                     X = 42   Y = 2
                                     SETBACK = 2    ..
      WALL-F3-L  =  EXTERIOR-WALL    HEIGHT = 12 WIDTH = 120
                                     X = 0 Y = 120 Z = 24
                                     AZIMUTH = 270   ..
            W-F3-L =WINDOW           WIDTH = 40
                                     X = 42   Y = 2
                                     SETBACK = 2    ..
      WALL-F3-B1  = EXTERIOR-WALL    HEIGHT = 12 WIDTH = 360
                                     X = 360 Y = 120 Z = 24
                                     AZIMUTH = 0   ..
           W-F3-B = WINDOW           WIDTH = 350  HEIGHT = 6
                                     X = 5   Y = 2
                                     SETBACK = 2  ..
      FLOOR-F3  = INTERIOR-WALL      HEIGHT=120  WIDTH=360
                                     X =0 Y = 120 Z = 24
                                     AREA = 43200
                                     AZIMUTH = 180
                                     NEXT-TO SPACE-F2
                                     CONSTRUCTION = FLOOR-1
                                     TILT = 180         ..
     ROOF-3      =ROOF               HEIGHT=120  WIDTH=360
                                     X=0   Y=0   Z=36
                                     AZIMUTH = 180
                                     TILT=0  GND-REFLECTANCE=0
                                     CONSTRUCTION = ROOF-1   ..

      $                            FLOOR A1

SPACE-AF1       =SPACE            SPACE-CONDITIONS = OFFICE
                                  AREA = 4800  VOLUME = 57600  ..

      WALL-AF1-F2  = INTERIOR-WALL     HEIGHT = 12 WIDTH = 120
                                       X =290 Y = 120 Z = 0
                                       AZIMUTH = 0
                                       CONSTRUCTION = FLOOR-1
                                       NEXT-TO SPACE-F1     ..
      WALL-AF1-L  =  EXTERIOR-WALL     HEIGHT = 12 WIDTH = 40
```

```
                                        X = 170 Y = 160 Z = 0
                                        AZIMUTH = 270  ..
    W-AF1-L     =WINDOW                 WIDTH = 8  HEIGHT = 8
                                        X = 30   Y = 0
                                        SETBACK = 0    ..
  WALL-AF1-R  =  EXTERIOR-WALL          HEIGHT = 12 WIDTH = 40
                                        X = 290 Y = 120 Z = 0
                                        AZIMUTH = 90   ..
    W-AF1-R     =WINDOW                 WIDTH = 24  HEIGHT = 5
                                        X = 8 Y = 3
                                        SETBACK = 2 ..
   ROOF-AF1    =  ROOF                  HEIGHT=40  WIDTH=80
                                        X =170 Y = 120 Z = 12
                                        AZIMUTH = 180
                                        TILT=0  GND-REFLECTANCE=0
                                        CONSTRUCTION = ROOF-1   ..
  FLOOR-AF1  = UNDERGROUND-FLOOR        HEIGHT=40  WIDTH=120
                                        X =170 Y = 160 Z = 0
                                        AZIMUTH = 180
                                        CONSTRUCTION = WALL-U
                                        TILT = 180         ..
$                          FLOOR A2

SPACE-AF2       =SPACE               SPACE-CONDITIONS = OFFICE
                                     AREA = 1600  VOLUME = 19200   ..

  WALL-AF2-F2  = INTERIOR-WALL         HEIGHT = 12 WIDTH = 40
                                       X =290 Y = 120 Z = 12
                                       AZIMUTH = 0
                                       NEXT-TO SPACE-F2
                                       CONSTRUCTION = FLOOR-1 ..
  WALL-AF2-L  =  EXTERIOR-WALL         HEIGHT = 12 WIDTH = 40
                                       X = 250 Y = 160 Z = 12
                                       AZIMUTH = 270  ..
  WALL-AF2-R  =  EXTERIOR-WALL         HEIGHT = 12 WIDTH = 40
                                       X = 290 Y = 120 Z = 12
                                       AZIMUTH = 90   ..
  FLOOR-AF2  = INTERIOR-WALL           HEIGHT=40  WIDTH=40
                                       X =250 Y = 160 Z = 12
                                       AZIMUTH = 180
                                       NEXT-TO SPACE-AF1
                                       CONSTRUCTION = FLOOR-1
                                       TILT = 180         ..
  ROOF-AF2    =  ROOF                  HEIGHT=40  WIDTH=40
                                       X =250 Y = 120 Z = 24
                                       AZIMUTH = 180
                                       TILT=0  GND-REFLECTANCE=0
                                       CONSTRUCTION = ROOF-1   ..
$                          FLOOR B1

SPACE-B1       =SPACE               SPACE-CONDITIONS = OFFICE
                                    AREA = 6400  VOLUME = 76800  ..

WALL-B1-F1  = INTERIOR-WALL            HEIGHT = 12 WIDTH = 80
```

```
                                    X =250 Y = 160 Z = 0
                                    AZIMUTH = 0
                                    NEXT-TO SPACE-AF1
                                    CONSTRUCTION = FLOOR-1  ..
      WALL-B1-L  =  EXTERIOR-WALL   HEIGHT = 12 WIDTH = 80
                                    X = 170 Y = 240 Z = 0
                                    AZIMUTH = 270  ..
           WIN-B1  =WINDOW          WIDTH = 60  HEIGHT = 5
                                    X = 10   Y = 3
                                    SETBACK = 2    ..
      WALL-B1-B1  = EXTERIOR-WALL   HEIGHT = 12 WIDTH = 80
                                    X =250 Y = 240 Z = 0
                                    AZIMUTH = 0     ..
      FLOOR-B1  = UNDERGROUND-FLOOR HEIGHT=80  WIDTH=120
                                    X =170 Y = 240 Z = 0
                                    AZIMUTH = 180
                                    CONSTRUCTION = WALL-U
                                    TILT = 180          ..
      ROOF-B1 =  ROOF               HEIGHT=80  WIDTH=80
                                    X =170 Y = 240 Z = 12
                                    AZIMUTH = 270
                                    TILT=0  GND-REFLECTANCE=0
                                    CONSTRUCTION = ROOF-1   ..

$                         Section C

SPACE-C1       =SPACE             SPACE-CONDITIONS = LAB
                                  AREA = 3200  VOLUME = 115200
..

WALL-C1-F2  = INTERIOR-WALL           HEIGHT = 12 WIDTH = 40
                                      X =290 Y = 160 Z = 0
                                      AZIMUTH = 0
                                      NEXT-TO SPACE-AF1
                                      CONSTRUCTION = WALL-1   ..
WALL-C2-F2  = INTERIOR-WALL           HEIGHT = 12 WIDTH = 40
                                      X =290 Y = 160 Z = 10
                                      AZIMUTH = 0
                                      NEXT-TO SPACE-AF2
                                      CONSTRUCTION = WALL-1   ..
WALL-C1-L  =  INTERIOR-WALL           HEIGHT = 12 WIDTH = 80
                                      X = 250 Y = 160 Z = 0
                                      AZIMUTH = 90
                                      NEXT-TO SPACE-B1
                                      CONSTRUCTION = WALL-1   ..
WALL-C1-R  = EXTERIOR-WALL            HEIGHT = 36 WIDTH = 80
                                      X =290 Y =160 Z = 0
                                      AZIMUTH = 90
                                      CONSTRUCTION =LAB-1  ..
WALL-C1-B  = EXTERIOR-WALL            HEIGHT = 36 WIDTH = 40
                                      X =290 Y =240 Z = 0
                                      AZIMUTH = 0
                                      CONSTRUCTION =LAB-1     ..
FLOOR-C1  = UNDERGROUND-FLOOR         HEIGHT=40  WIDTH=80
```

```
                                            X =170 Y =240 Z = 0
                                            AZIMUTH = 180
                                            CONSTRUCTION = WALL-U
                                            TILT = 180         ..
WALL-C2-L  = EXTERIOR-WALL                  HEIGHT = 24 WIDTH = 80
                                            X =250 Y =240 Z = 12
                                            AZIMUTH = 270
                                            CONSTRUCTION =LAB-1   ..
WALL-C3-F  = EXTERIOR-WALL                  HEIGHT = 12 WIDTH = 40
                                            X =250 Y =160 Z = 24
                                            AZIMUTH = 180
                                            CONSTRUCTION =LAB-1        ..
      ROOF-C      =  ROOF                   HEIGHT=40  WIDTH=80
                                            X =250 Y = 240 Z = 36
                                            AZIMUTH = 270
                                            TILT=0  GND-REFLECTANCE=0
                                            CONSTRUCTION = ROOF-1   ..

$-----HOURLY REPORTS-----$
PLTSCH  = SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..
PLOTER1 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (4)  ..
PLOTER2 = REPORT-BLOCK
          VARIABLE-TYPE = BUILDING
          VARIABLE-LIST = (1, 2, 19, 20, 37)         ..
LDS-1 = HOURLY-REPORT
        REPORT-SCHEDULE=PLTSCH
        REPORT-BLOCK   = (PLOTER1, PLOTER2)
        OPTION         = PRINT      ..

END  ..
COMPUTE  LOADS   ..

INPUT SYSTEMS   ..

               PARAMETER          HEAT1 = 75.0
                                  COOL1 = 76.0        ..

                 $ SYSTEM TYPE VAVS VARIABLE VOLUME FAN SYSTEM

SYSTEMS-REPORT   SUMMARY=(SS-A,SS-B,SS-C,SS-D,SS-F,SS-J,SS-K,SS-N)
VERIFICATION=(SV-A)                                   ..

                 $ SYSTEMS SCHEDULES

FAN-1      =DAY-SCHEDULE     (1,6)(1)(7,8)(1)(9,18)(1)(19,24)(1)  ..
FAN-2      =DAY-SCHEDULE     (1,24) (1)   ..
FAN-SCHED  =SCHEDULE         THRU DEC 31 (WD) FAN-1 (WEH) FAN-2  ..

           $ HEATING AND COOLING IS AVAILABLE THROUGHOUT THE YEAR
COOLOFF     =SCHEDULE        THRU DEC 31 (ALL) (1,24) (1)  ..
HEATOFF     =SCHEDULE        THRU DEC 31 (ALL) (1,24) (1)  ..
```

```
           $ SPECIFIES THE SET POINT OF THE ZONE HEATING THERMOSTAT
HEAT-1      =DAY-SCHEDULE    (1,8) (HEAT1)   (9,18) (HEAT1)  (19,24)
(HEAT1)  ..
HEAT-2      =DAY-SCHEDULE    (1,24) (HEAT1)   ..
HEAT-WEEK   =WEEK-SCHEDULE   (MON,FRI)  HEAT-1   (WEH)  HEAT-2   ..
HEAT-SCHED  =SCHEDULE        THRU DEC 31   HEAT-WEEK   ..

           $ SPECIFIES THE SET POINT OF THE ZONE COOLING THERMOSTAT
COOL-1      =DAY-SCHEDULE    (1,8) (COOL1)  (9,18) (COOL1)  (19,24)
(COOL1)  ..
COOL-2      =DAY-SCHEDULE    (1,24) (COOL1)   ..
COOL-WEEK   =WEEK-SCHEDULE   (MON,FRI)  COOL-1   (WEH)  COOL-2   ..
COOL-SCHED  =SCHEDULE        THRU DEC 31   COOL-WEEK   ..

R1          =DAY-RESET-SCH   SUPPLY-HI=54   SUPPLY-LO=54
                             OUTSIDE-LO=30  OUTSIDE-HI=55  ..
SAT-RESET   =RESET-SCHEDULE  THRU DEC 31 (ALL) R1  ..

                    $ SYSTEM DESCRIPTION

ZAIR         =ZONE-AIR      CFM/SQFT = 0.5626
                            OUTSIDE-AIR-CFM = 5845          ..

CONTROL      =ZONE-CONTROL  DESIGN-HEAT-T=68  DESIGN-COOL-T=78
                            HEAT-TEMP-SCH= HEAT-SCHED
                            COOL-TEMP-SCH= COOL-SCHED
                            THERMOSTAT-TYPE= REVERSE-ACTION  ..

SPACE-F0     =ZONE          ZONE-AIR=ZAIR
                            SIZING-OPTION=ADJUST-LOADS
                            ZONE-TYPE=CONDITIONED
                            ZONE-CONTROL=CONTROL                 $
39504ft2 $ ..
SPACE-F1     =ZONE          LIKE SPACE-F0              ..
SPACE-F2     =ZONE          LIKE SPACE-F0              ..
SPACE-F3     =ZONE          LIKE SPACE-F0              ..
SPACE-AF1    =ZONE          LIKE SPACE-F0 OUTSIDE-AIR-CFM = 710
$ 4800ft2 $  ..
SPACE-AF2    =ZONE          LIKE SPACE-F0 OUTSIDE-AIR-CFM = 237
$ 1600ft2 $  ..
SPACE-B1     =ZONE          LIKE SPACE-F0 OUTSIDE-AIR-CFM = 947
$ 6400ft2 $  ..
SPACE-C1     =ZONE          LIKE SPACE-F0 OUTSIDE-AIR-CFM = 473
$ 3200ft2 $  ..

S-CONT       =SYSTEM-CONTROL    COOLING-SCHEDULE= COOLOFF
                                HEATING-SCHEDULE= HEATOFF
                                COOL-CONTROL=RESET
                                COOL-RESET-SCH=SAT-RESET
                                MAX-SUPPLY-T=105
                                MIN-SUPPLY-T=53.0   ..

S-FAN        =SYSTEM-FANS       FAN-SCHEDULE=FAN-SCHED
                                FAN-CONTROL=SPEED
```

```
                                SUPPLY-STATIC=4.1
                                SUPPLY-EFF=.70
                                MOTOR-PLACEMENT = IN-AIRFLOW
                                NIGHT-CYCLE-CTRL=CYCLE-ON-ANY
                                RETURN-STATIC=0.9
                                RETURN-EFF=0.70    ..

S-TERM      =SYSTEM-TERMINAL    REHEAT-DELTA-T= 55
                                MIN-CFM-RATIO= 1   ..

SYST-1     =SYSTEM    SYSTEM-TYPE=VAVS
                                $ Variable Volume Fan System w/Opt
Reheat (VAVS)
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      NIGHT-CYCLE-CTRL=CYCLE-ON-ANY
                         ECONO-LIMIT-T=55
                         ECONO-LOW-LIMIT=55
                      RETURN-AIR-PATH=DUCT
                      ZONE-NAMES = (SPACE-F1, SPACE-F2,
                      SPACE-F3, SPACE-AF1, SPACE-AF2,SPACE-B1)   ..

SYST-2     =SYSTEM    SYSTEM-TYPE=VAVS
                                $ Variable Volume Fan System w/Opt
Reheat (VAVS)
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      NIGHT-CYCLE-CTRL=CYCLE-ON-ANY
                         ECONO-LIMIT-T=55
                         ECONO-LOW-LIMIT=55
                      RETURN-AIR-PATH=DUCT
                      ZONE-NAMES = (SPACE-C1)
                                RETURN-STATIC=0
                                MIN-CFM-RATIO= 1      ..

SYST-3     =SYSTEM    SYSTEM-TYPE=VAVS
           $ Variable Volume Fan System w/Opt Reheat (VAVS)
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      NIGHT-CYCLE-CTRL=CYCLE-ON-ANY
                         ECONO-LIMIT-T=55
                         ECONO-LOW-LIMIT=55
                      RETURN-AIR-PATH=DUCT
                      ZONE-NAMES = (SPACE-F0)
                                MIN-CFM-RATIO= 1       ..

PLANT-1  =  PLANT-ASSIGNMENT   SYSTEM-NAMES = (SYST-1, SYST-2, SYST-3)
..

$-----HOURLY REPORTS-----$
PLTSCH2  = SCHEDULE          THRU DEC 31 (ALL) (1,24) (1)  ..
```

```
PLOTER21 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (8)       ..
PLOTER22 = REPORT-BLOCK
          VARIABLE-TYPE = PLANT-1
          VARIABLE-LIST = (1,2,3)          ..
LDS-2 = HOURLY-REPORT
        REPORT-SCHEDULE=PLTSCH2
        REPORT-BLOCK   = (PLOTER21, PLOTER22)
        OPTION         = PRINT      ..

END   ..
COMPUTE SYSTEMS   ..


INPUT PLANT   ..

PLANT-1  =  PLANT-ASSIGNMENT ..

              PLANT-REPORT    SUMMARY=(ALL-SUMMARY,BEPS)
              VERIFICATION = (ALL-VERIFICATION)  ..
CHIL1    =PLANT-EQUIPMENT  TYPE= HERM-CENT-CHLR  SIZE= -999         ..
          PLANT-PARAMETERS  HERM-CENT-COND-TYPE=TOWER      ..
BOIL1   =PLANT-EQUIPMENT  TYPE=HW-BOILER  SIZE=-999    ..
PLANT-COSTS      PROJECT-LIFE=25  DISCOUNT-RATE=5 ..
ENERGY-RESOURCE  RESOURCE=ELECTRICITY  ..
ENERGY-RESOURCE  RESOURCE=NATURAL-GAS  ENERGY/UNIT=100000
                  UNIT-NAME=THERMS   ..

                       $-----HOURLY REPORTS-----$
PLTSCH2  = SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..
PLOTER31 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (1)         ..
PLOTER32 = REPORT-BLOCK
          VARIABLE-TYPE = PLANT
          VARIABLE-LIST = (1,2,3)       ..
                  $  1 Heating load from SYSTEMS (Btu/hr)
                  $  2 Cooling load from SYSTEMS (Btu/hr)
                  $  3 Electric load from SYSTEMS (Btu/hr)
PLOTER41 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (1)         ..
PLOTER42 = REPORT-BLOCK
          VARIABLE-TYPE = END-USE
          VARIABLE-LIST = (6,7,8,9)   ..
LDS-3 = HOURLY-REPORT
         REPORT-SCHEDULE=PLTSCH2
         REPORT-BLOCK   = (PLOTER31, PLOTER32)
         OPTION         = PRINT       ..
END  ..
COMPUTE PLANT   ..
STOP ..
```

APPENDIX B

DOE-2.1E VALLEY PARK OFFICE BUILDING BASELINE SIMULATION MODEL

```
INPUT LOADS  ..

TITLE           LINE-1 * Valley Park Building, College Station  *
                LINE-2 * Baseline HVAC design *
                LINE-3 * DOE-2.1E *
                LINE-4 * IECC 2009 complaint *
                LINE-5 * Oleksandr Tanskyi *                       ..
                RUN-PERIOD         JAN 1 2010 THRU DEC 31 2010     ..
                ABORT              ERRORS                          ..
                DIAGNOSTIC      WARNINGS  NO-ECHO NARROW           ..
                LOADS-REPORT       SUMMARY =( LS-F, LS-D)
                                   VERIFICATION = (LV-B,LV-D)      ..
                BUILDING-LOCATION  LATITUDE=30.61  LONGITUDE=96.3
                                   ALTITUDE=325   AZIMUTH= 315.0
                                   TIME-ZONE=6                     ..

$                         BUILDING DESCRIPTION

$                         CONSTRUCTION AND GLASS-TYPES

ROO-1  =LAYERS =MAT=(RG02, CB56,IN35, IN35, WD11)
                INSIDE-FILM-RES = 0.92  ..
$  R-20ci required for baseline building design
$ RG02 - 1 inch Roof Gravel or Slag R=0.1
$ CB56 - Hollow Concrete Block, 12 inch Light Weight R=2.27
$ IN35 - 2 inch Expanded Polystyrene, R= 8.33
$ IN35 - 2 inch Expanded Polystyrene, R= 8.33
$ WD11 - 3/4 inch Hard Wood, R= 0.68

WA-1 =LAYERS =MAT=(IN33, CB27,CM02)  ..
$ R-5.7 required for baseline building design
$ IN33 - 1 inch Expanded Polystyrene, R= 4.16
$ CB27 - 6 inch Medium Weight Concrete Filled Concrete Block R= 1.13
$ CM02 – 1.75 inch Mortar Cement R= 0.35

WA-2 =LAYERS =MAT=(GP04,BK01,GP04)  ..
$ GP04 -  3/4 inch Light Weight Aggregate Gypsum Plaster R= 0.47
$ BK01 - 4 inch Common Brick R= 0.80
$ GP04 -  3/4 inch Light Weight Aggregate Gypsum Plaster R= 0.47

UND-W1 =LAYERS =MAT=(GP05,IN04,CB56)   ..
$ GP05 -  1 inch Light Weight Aggregate Gypsum Plaster R= 0.63
$ IN04 - Batt, R-24 Mineral Wool/Fiber R= 27.88
$ CB56 - Hollow Concrete Block, 12 inch Light Weight R=2.27
```

```
WALL-EXT       =CONSTRUCTION       LAYERS=WA-1     ..
WALL-INT       =CONSTRUCTION       LAYERS=WA-2     ..
ROOF-1         =CONSTRUCTION       LAYERS=ROO-1    ..
WALL-U         =CONSTRUCTION       LAYERS=UND-W1   ..
W-1            = GLASS-TYPE  PANES = 1
              SHADING-COEF = 0.384  GLASS-CONDUCTANCE = 0.879     ..
DOORS          = GLASS-TYPE  PANES = 1
                SHADING-COEF = 0.384  GLASS-CONDUCTANCE = 1.450   ..

$                          BUILDING SHADES

      BDSH11 = BUILDING-SHADE
      X = -23  Y = 0  Z = 17.7
      HEIGHT = 4
      WIDTH = 57.3
      AZIMUTH = 0
      TRANSMITTANCE = 0
      TILT = 0
      SHADE-SCHEDULE = B-SH-1     ..

      BDSH12 = BUILDING-SHADE
      X = -23  Y = -4  Z = 15
      HEIGHT = 6.5
      WIDTH = 57.3
      AZIMUTH = 0
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BDSH21 = BUILDING-SHADE
      X = 80.3  Y = 0  Z = 17.7
      HEIGHT = 4
      WIDTH = 57.3
      AZIMUTH = 0
      TRANSMITTANCE = 0
      TILT = 0
      SHADE-SCHEDULE = B-SH-1     ..

      BDSH22 = BUILDING-SHADE
      X = 80.3  Y = -4  Z = 15
      HEIGHT = 6.5
      WIDTH = 57.3
      AZIMUTH = 0
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

      BDSH31 = BUILDING-SHADE
      X = 134  Y = 32.6  Z = 17.7
      HEIGHT = 4
      WIDTH = 33.3
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 0
```

```
      SHADE-SCHEDULE = B-SH-1     ..

      BDSH32 = BUILDING-SHADE
      X = 134  Y = 32.6  Z = 15
      HEIGHT = 6.5
      WIDTH = 33.3
      AZIMUTH = 90
      TRANSMITTANCE = 0
      TILT = 90
      SHADE-SCHEDULE = B-SH-1     ..

$                              SHADING SCHEDULE

B-SH-1 = SCHEDULE   THRU DEC 31 (ALL) (1,24)  (1)   ..

                      $ OCCUPANCY SCHEDULE

OC-1           =DAY-SCHEDULE       (1,6)   (0.00)
                                   (7,8)   (0.05)
                                   (9,11)  (1.0)
                                   (12,14) (0.8,0.4,0.8)
                                   (15,18) (1.0)
                                   (19,21) (0.5,0.1,0.1)
                                   (22,24) (0.05)  ..
OC-2           =DAY-SCHEDULE       (1,6) (0.0)
                                   (7,8) (0.05)
                                   (9,17) (0.10)
                                   (18,24) (0.05)            ..
OC-WEEK        =WEEK-SCHEDULE      (WD) OC-1 (WEH) OC-2      ..
OCCUPY-1       =SCHEDULE           THRU DEC 31 OC-WEEK       ..

                       $ LIGHTING AND OFFICE EQUIPMENT SCHEDULE

LTQ-1           =DAY-SCHEDULE      (1,4) (0.161, 0.151, 0.146, 0.145)
                                   (5,8) (0.143, 0.145, 0.175, 0.363)
                                   (9,12) (0.798, 0.988, 1.000, 0.973)
                                   (13,16) (0.928, 0.955, 0.985, 0.962)
                                   (17,20) (0.861, 0.610, 0.397, 0.270)
                                   (21,24) (0.233, 0.219, 0.206, 0.186)
..

LTQ-2           =DAY-SCHEDULE      (1,4) (0.129, 0.126, 0.127, 0.125)
                                   (5,8) (0.127, 0.126, 0.125, 0.129)
                                   (9,12) (0.135, 0.157, 0.171, 0.181)
                                   (13,16) (0.190, 0.197, 0.216, 0.226)
                                   (17,20) (0.224, 0.197, 0.183, 0.175)
                                   (21,24) (0.156, 0.155, 0.147, 0.143)
..

LTQ-3           =DAY-SCHEDULE      (1,4) (0.140, 0.136, 0.138, 0.139)
                                   (5,8) (0.137, 0.133, 0.134, 0.135)
                                   (9,12) (0.134, 0.136, 0.165, 0.185)
                                   (13,16) (0.194, 0.191, 0.188, 0.203)
                                   (17,20) (0.202, 0.204, 0.207, 0.209)
```

```
                                   (21,24) (0.183, 0.170, 0.163, 0.148)
..

LTQ-WEEK       =WEEK-SCHEDULE      (MON,FRI) LTQ-1
                                   (SAT) LTQ-2
                                   (SUN) LTQ-3
                                   (HOL) LTQ-3  ..
LITEQP-1       =SCHEDULE           THRU DEC 31 LTQ-WEEK  ..

                 $ INFILTRATION SCHEDULE

INFIL-SCH1      =SCHEDULE           THRU DEC 31 (ALL) (1,24) (0)  ..

                       $ SET DEFAULT VALUES

  SET-DEFAULT FOR SPACE  FLOOR-WEIGHT=0  ..
  SET-DEFAULT FOR EXTERIOR-WALL  CONSTRUCTION=WALL-EXT HEIGHT = 17.7 ..
  SET-DEFAULT FOR INTERIOR-WALL  CONSTRUCTION=WALL-INT HEIGHT = 17.7 ..
  SET-DEFAULT FOR WINDOW  Y=0.1
                          GLASS-TYPE=W-1
                          WIDTH=7.66
                          HEIGHT=9.66
                          SETBACK = 0 ..

                       $ GENERAL SPACE DEFINITION

OFFICE        = SPACE-CONDITIONS
              PEOPLE-SCHEDULE      =OCCUPY-1
              AREA/PERSON          =100
              PEOPLE-HG-LAT        =200
              PEOPLE-HG-SENS       =250
              LIGHTING-SCHEDULE    = LITEQP-1
              LIGHTING-TYPE        =REC-FLUOR-RV
              LIGHT-TO-SPACE       =.80
              LIGHTING-W/SQFT      =0.9
              EQUIP-SCHEDULE       = LITEQP-1
              EQUIPMENT-W/SQFT     =0.8
              INF-METHOD           =AIR-CHANGE
              AIR-CHANGES/HR       =0.0     $ASSUMING PRESSURIZATION
              INF-SCHEDULE         =INFIL-SCH1  ..

     $                       EXTERIOR ZONE 1

SPACE-E1       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 2624   VOLUME = 39360   ..

WALL-E1-F  = EXTERIOR-WALL         WIDTH = 130
                                   X =-130 Y =0  Z = 0
                                   AZIMUTH = 180   ..
     W-E1-F1 =  WINDOW              X = 4.3
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 7.0      ..
     W-E1-F2 =  WINDOW              X = 21
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
```

```
                 OVERHANG-D = 7.0       ..
     W-E1-F3 =  WINDOW              X = 37.6
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
                 OVERHANG-D = 7.0       ..
     W-E1-F4 =  WINDOW    GLASS-TYPE = DOORS          X = 53.3     ..
     W-E1-F5 =  WINDOW    GLASS-TYPE = DOORS          X = 65.6     ..
     W-E1-F6 =  WINDOW    GLASS-TYPE = DOORS          X = 81.9     ..
     W-E1-F7 =  WINDOW    GLASS-TYPE = DOORS          X = 94.3     ..
     W-E1-F8 =  WINDOW              X = 111.3
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
                 OVERHANG-D = 7.0       ..
     W-E1-F9 =  WINDOW              X = 126.15 WIDTH=3.83
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 5.38
                 OVERHANG-D = 7.0       ..
WALL-E1-L  =  EXTERIOR-WALL        WIDTH = 50
                                   X = -130 Y = 50 Z = 0
                                   AZIMUTH = 270  ..
     W-E1-L1 =  WINDOW              X = 4.3
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
                 OVERHANG-D = 7.0       ..
     W-E1-L2 =  WINDOW              X = 21      ..
     W-E1-L3 =  WINDOW              X = 37.6
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
                 OVERHANG-D = 7.0       ..
HEXAGON1 = POLYGON (0,0) (130,0)  (130,16) (16,16) (16,50) (0,50)   ..
ROOF-E1 =  ROOF                    POLYGON = HEXAGON1
                                   X =-130 Y = 0 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0  CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0              ..
FLOOR-E1  = UNDERGROUND-FLOOR      POLYGON = HEXAGON1
                                   X =-130 Y = 0 Z =0
                                   AZIMUTH = 180 CONSTRUCTION=WALL-U ..

     $                          EXTERIOR ZONE 2

SPACE-E2       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 2624   VOLUME = 39360   ..

WALL-E2-F  =  EXTERIOR-WALL        WIDTH = 130
                                   X =0 Y =0  Z = 0
                                   AZIMUTH = 180   ..
     W-E2-F1 =  WINDOW              X = 0 WIDTH=3.83
                 OVERHANG-A = 0   OVERHANG-B = 0.5  OVERHANG-W = 5.38
                 OVERHANG-D = 7.0       ..
     W-E2-F2 =  WINDOW              X = 11.5
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
                 OVERHANG-D = 7.0       ..
     W-E2-F3 =  WINDOW    GLASS-TYPE = DOORS          X = 28.5     ..
     W-E2-F4 =  WINDOW    GLASS-TYPE = DOORS          X = 40.9     ..
     W-E2-F5 =  WINDOW    GLASS-TYPE = DOORS          X = 57.2     ..
     W-E2-F6 =  WINDOW    GLASS-TYPE = DOORS          X = 69.5     ..
     W-E2-F7 =  WINDOW                                X = 85.2
                 OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
```

```
              OVERHANG-D = 7.0      ..
     W-E2-F8 =  WINDOW              X = 101.8
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
     W-E2-F9 =  WINDOW              X = 118.5
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
WALL-E2-R  =  EXTERIOR-WALL        WIDTH = 50
                                   X = 130 Y = 0 Z = 0
                                   AZIMUTH = 90  ..
     W-E2-R1 =  WINDOW              X = 4.3
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
     W-E2-R2 =  WINDOW              X = 21
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
     W-E2-R3 =  WINDOW   GLASS-TYPE = DOORS          X = 37.6    ..
WALL-E2-L  =  INTERIOR-WALL        WIDTH = 16
                                   X = 0 Y = 0 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E1  ..
HEXAGON2 = POLYGON (130,0) (130,50) (114,50) (114,16) (0,16) (0,0)   ..
ROOF-E2 =  ROOF                    POLYGON = HEXAGON2
                                   X =0 Y = 0 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0   CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0        ..
FLOOR-E2  = UNDERGROUND-FLOOR      POLYGON = HEXAGON2
                                   X =0 Y = 0 Z =0
                                   AZIMUTH = 180 CONSTRUCTION=WALL-U
..

$                          EXTERIOR ZONE 3

SPACE-E3       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 2624   VOLUME = 39360   ..
WALL-E3-F  =  INTERIOR-WALL        WIDTH = 16
                                   X =114 Y =50  Z = 0
                                   AZIMUTH = 180
                                   NEXT-TO SPACE-E2  ..
WALL-E3-R  =  EXTERIOR-WALL        WIDTH = 50
                                   X = 130 Y = 50 Z = 0
                                   AZIMUTH = 90  ..
     W-E3-R1 =  WINDOW   GLASS-TYPE = DOORS          X = 4.3  ..
     W-E3-R2 =  WINDOW              X = 21
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
     W-E3-R3 =  WINDOW              X = 37.6
              OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
              OVERHANG-D = 7.0      ..
WALL-E3-B  =  EXTERIOR-WALL        WIDTH = 130
                                   X = 130 Y = 100 Z = 0
                                   AZIMUTH = 0   ..
     W-E3-B1 =  WINDOW              X = 4.3
```

```
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      ..
     W-E3-B2 =  WINDOW              X = 20.9   ..
     W-E3-B3 =  WINDOW              X = 37.5
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      ..
     W-E3-B4 =  WINDOW              X = 54.1 ..
     W-E3-B5 =  WINDOW              X = 70.7
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      SETBACK = 4     ..
     W-E3-B6 =  WINDOW              X = 87.3 ..
     W-E3-B7 =  WINDOW              X = 103.9
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      ..
     W-E3-B8 =  WINDOW              X = 120.5 ..
ROOF-E3 =  ROOF                    POLYGON = HEXAGON1
                                   X =130 Y = 100 Z = 17.7
                                   AZIMUTH = 0
                                   TILT=0   CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0           ..
FLOOR-E3  = UNDERGROUND-FLOOR      POLYGON = HEXAGON1
                                   X =130 Y = 100 Z =0
                                   AZIMUTH = 0  CONSTRUCTION=WALL-U
..

$                         EXTERIOR ZONE 4

SPACE-E4       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 2647   VOLUME = 39705   ..
WALL-E4-B  =  EXTERIOR-WALL        WIDTH = 104.5
                                   X = 0 Y = 100 Z = 0
                                   AZIMUTH = 0   ..
     W-E4-B1 =  WINDOW              X = 4.3
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      ..
     W-E4-B2 =  WINDOW              X = 20.9   ..
     W-E4-B3 =  WINDOW              X = 37.5
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0    SETBACK = 4       ..
     W-E4-B4 =  WINDOW              X = 54.1 ..
     W-E4-B5 =  WINDOW              X = 70.7
               OVERHANG-A = 1.5   OVERHANG-B = 0.5  OVERHANG-W = 10.75
               OVERHANG-D = 2.0      ..
WALL-E4-R  =  INTERIOR-WALL        WIDTH = 16
                                   X = 0 Y = 84 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E3  ..
WALL-E4-LB  =  EXTERIOR-WALL       WIDTH = 36.0
                                   X = -104.5 Y = 100 Z = 0
                                   AZIMUTH = -45   ..
WALL-E4-L  =  EXTERIOR-WALL        WIDTH = 24.5
                                   X = -130 Y = 74.5 Z = 0
                                   AZIMUTH = 270  ..
WALL-E4-F  = INTERIOR-WALL         WIDTH = 16
```

```
                                   X =-130 Y =50  Z = 0
                                   AZIMUTH = 180
                                   NEXT-TO SPACE-E1  ..
OCT1 = POLYGON (16,0) (16,11.7) (38.3,34) (130,34) (130,50) (24.5,50)
(0,24.5)
(0,0)  ..

ROOF-E4 =  ROOF                    POLYGON = OCT1
                                   X =-130 Y = 50 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0   CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0        ..
FLOOR-E4  = UNDERGROUND-FLOOR      POLYGON = OCT1
                                   X =-130 Y = 50 Z =0
                                   AZIMUTH = 180  CONSTRUCTION=WALL-U
..

$                          INTERIOR ZONE 1

SPACE-I1       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 3876   VOLUME = 58140   ..
WALL-I1-F  = INTERIOR-WALL         WIDTH = 114
                                   X =0 Y =16  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-E1  ..
WALL-I1-L  =  INTERIOR-WALL        WIDTH = 34
                                   X = -114 Y = 16 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E1  ..
ROOF-I1 =  ROOF                    HEIGHT=34  WIDTH=114
                                   X =-114 Y = 16 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0  CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0             ..
FLOOR-I1  = UNDERGROUND-FLOOR      HEIGHT=34  WIDTH=114
                                   X =-114 Y = 16 Z = 0
                                   AZIMUTH = 180  CONSTRUCTION=WALL-U
..

$                          INTERIOR ZONE 2

SPACE-I2       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 3876   VOLUME = 58140   ..
WALL-I2-F  = INTERIOR-WALL         WIDTH = 114
                                   X =114 Y =16  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-E2  ..
WALL-I2-R  =  INTERIOR-WALL        WIDTH = 34
                                   X = 114 Y = 16 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E2  ..
WALL-I2-L  =  INTERIOR-WALL        WIDTH = 34
                                   X = 0 Y = 16 Z = 0
                                   AZIMUTH = 90
```

```
                                   NEXT-TO SPACE-I1  ..
ROOF-I2 =  ROOF                    HEIGHT=34  WIDTH=114
                                   X =0 Y = 16 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0  CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0            ..
FLOOR-I2  = UNDERGROUND-FLOOR      HEIGHT=34  WIDTH=114
                                   X =0 Y = 16 Z = 0
                                   AZIMUTH = 180  CONSTRUCTION=WALL-U
..

$                          INTERIOR ZONE 3

SPACE-I3       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 3876   VOLUME = 58140   ..
WALL-I3-B  = INTERIOR-WALL         WIDTH = 114
                                   X =114 Y =84  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-E3  ..
WALL-I3-R  =  INTERIOR-WALL        WIDTH = 34
                                   X = 114 Y = 50 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E3  ..
WALL-I3-F  = INTERIOR-WALL         WIDTH = 114
                                   X =114 Y =50  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-I2  ..
ROOF-I3 =  ROOF                    HEIGHT=34  WIDTH=114
                                   X =0 Y = 50 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0  CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0             ..
FLOOR-I3  = UNDERGROUND-FLOOR      HEIGHT=34  WIDTH=114
                                   X =0 Y = 50 Z = 0
                                   AZIMUTH = 180  CONSTRUCTION=WALL-U
..

$                          INTERIOR ZONE 4

SPACE-I4       =SPACE              SPACE-CONDITIONS = OFFICE
                                   AREA = 3627   VOLUME = 54405   ..
WALL-I4-B  = INTERIOR-WALL         WIDTH = 91.7
                                   X =0 Y =84  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-E4  ..
WALL-I4-L  =  INTERIOR-WALL        WIDTH = 11.7
                                   X = -114 Y = 50 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-E4  ..
WALL-I4-LB  = INTERIOR-WALL        WIDTH = 31.5
                                   X = -114 Y = 61.7 Z = 0
                                   AZIMUTH = 135
                                   NEXT-TO SPACE-E4  ..
WALL-I4-R  =  INTERIOR-WALL        WIDTH = 34
```

```
                                   X = 0 Y = 50 Z = 0
                                   AZIMUTH = 90
                                   NEXT-TO SPACE-I3  ..
WALL-I4-F  = INTERIOR-WALL         WIDTH = 114
                                   X =0 Y =50  Z = 0
                                   AZIMUTH = 0
                                   NEXT-TO SPACE-I1  ..
PENTAGON1 = POLYGON (0,0)(114,0)(114,34)(22.3,34)(0,11.7)..
ROOF-I4 =  ROOF                    POLYGON = PENTAGON1
                                   X =-114 Y = 50 Z = 17.7
                                   AZIMUTH = 180
                                   TILT=0 CONSTRUCTION = ROOF-1
                                   GND-REFLECTANCE=0 ..
FLOOR-I4  = UNDERGROUND-FLOOR      POLYGON = PENTAGON1
                                   X =-114 Y = 50 Z =0
                                   AZIMUTH = 180   CONSTRUCTION=WALL-U
..

$-----HOURLY REPORTS-----$
PLTSCH  = SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..
PLOTER1 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (4)         ..
PLOTER2 = REPORT-BLOCK
          VARIABLE-TYPE = BUILDING
          VARIABLE-LIST = (1, 2, 19, 20, 37)         ..
 LDS-1 = HOURLY-REPORT
        REPORT-SCHEDULE=PLTSCH
        REPORT-BLOCK   = (PLOTER1, PLOTER2)
        OPTION         = PRINT      ..

END  ..
COMPUTE  LOADS   ..

INPUT SYSTEMS   ..

PARAMETER          HT1 = 70.0    HT2 = 55.0
                   CL1 = 75.0    CL2 = 99.0                ..

$ SYSTEM TYPE: (PVVT)
$ PACKAGED SINGLE ZONE FAN SYSTEM - VARIABLE VOLUME - VARIABLE
TEMPERATURE
       SYSTEMS-REPORT  SUMMARY=(SS-A, SS-B, SS-C, SS-F, SS-J,SS-K, SS-
N)
       VERIFICATION=(SV-A)                                  ..

                 $ SYSTEMS SCHEDULES

FAN-1      =DAY-SCHEDULE     (1,5)(0)(6,7)(-999)(8,18)(1)(19,24)(0)  ..
FAN-2      =DAY-SCHEDULE     (1,24) (0)   ..
FAN-SCHED  =SCHEDULE         THRU DEC 31 (WD) FAN-1 (WEH) FAN-2  ..

           $ HEATING AND COOLING IS AVAILABLE THROUGHOUT THE YEAR
COOLOFF     =SCHEDULE        THRU DEC 31 (ALL) (1,24) (1)  ..
```

```
HEATOFF     =SCHEDULE        THRU DEC 31 (ALL) (1,24) (1)  ..

           $ SPECIFIES THE SET POINT OF THE ZONE HEATING THERMOSTAT
HEAT-1 =DAY-SCHEDULE (1,6) (HT2) (7,8) (HT1) (9,18) (HT1)  (19,24)
(HT1)  ..
HEAT-2 =DAY-SCHEDULE (1,6) (HT2) (7,24) (HT1)   ..
HEAT-WEEK1   =WEEK-SCHEDULE   (MON,FRI)  HEAT-1   (WEH)  HEAT-2   ..
HEAT-3 =DAY-SCHEDULE (1,6) (HT2) (7,8) (HT1) (9,18) (HT1)  (19,24)
(HT1)  ..
HEAT-4 =DAY-SCHEDULE (1,6) (HT2) (7,24) (HT1)   ..
HEAT-WEEK2   =WEEK-SCHEDULE   (MON,FRI)  HEAT-3   (WEH)  HEAT-4   ..

HEAT-SCHED  =SCHEDULE          THRU APR 15   HEAT-WEEK1
                               THRU OCT 31   HEAT-WEEK2
                               THRU DEC 31   HEAT-WEEK1   ..

           $ SPECIFIES THE SET POINT OF THE ZONE COOLING THERMOSTAT
COOL-1  =DAY-SCHEDULE (1,6) (CL2) (7,8) (CL1) (9,18) (CL1) (19,24)
(CL1)  ..
COOL-2  =DAY-SCHEDULE (1,6) (CL2) (7,24) (CL1)   ..
COOL-WEEK1   =WEEK-SCHEDULE   (MON,FRI)  COOL-1   (WEH)  COOL-2   ..
COOL-3 =DAY-SCHEDULE (1,6) (CL2) (7,8) (CL1) (9,18) (CL1) (19,24) (CL1)
..
COOL-4 =DAY-SCHEDULE (1,6) (CL2) (7,24) (CL1)    ..
COOL-WEEK2   =WEEK-SCHEDULE   (MON,FRI)  COOL-3   (WEH)  COOL-4   ..

COOL-SCHED  =SCHEDULE        THRU APR 15   COOL-WEEK1
                             THRU OCT 31   COOL-WEEK2
                             THRU DEC 31   COOL-WEEK1      ..

                            $ SYSTEM DESCRIPTION

ZAIR         =ZONE-AIR      CFM/SQFT =1.0        ..

CONTROL      =ZONE-CONTROL  DESIGN-HEAT-T=70
                            DESIGN-COOL-T=75
                            HEAT-TEMP-SCH= HEAT-SCHED
                            COOL-TEMP-SCH= COOL-SCHED
                            THERMOSTAT-TYPE= REVERSE-ACTION  ..
SPACE-E1     =ZONE          ZONE-AIR=ZAIR
                            SIZING-OPTION=ADJUST-LOADS
                            ZONE-TYPE=CONDITIONED
                            ZONE-CONTROL=CONTROL
                            OUTSIDE-AIR-CFM = 355      ..
SPACE-E2     =ZONE          LIKE SPACE-E1              ..
SPACE-E3     =ZONE          LIKE SPACE-E1              ..
SPACE-E4     =ZONE          LIKE SPACE-E1              ..
SPACE-I1     =ZONE          LIKE SPACE-E1              ..
SPACE-I2     =ZONE          LIKE SPACE-E1              ..
SPACE-I3     =ZONE          LIKE SPACE-E1              ..
SPACE-I4     =ZONE          LIKE SPACE-E1              ..

S-CONT       =SYSTEM-CONTROL    COOLING-SCHEDULE= COOLOFF
                                HEATING-SCHEDULE= HEATOFF
```

```
                                MAX-SUPPLY-T=90
                                MIN-SUPPLY-T=55       ..

S-FAN        =SYSTEM-FANS    FAN-SCHEDULE=FAN-SCHED
                             FAN-CONTROL=SPEED
                             SUPPLY-DELTA-T=2.117
                             SUPPLY-KW= 0.000685
                             MOTOR-PLACEMENT = IN-AIRFLOW
                             MAX-FAN-RATIO = 1.1
                             MIN-FAN-RATIO = 0.3
                             NIGHT-CYCLE-CTRL = CYCLE-ON-ANY         ..

S-TERM       =SYSTEM-TERMINAL   MIN-CFM-RATIO   = 0.15     ..

S-EQU        =SYSTEM-EQUIPMENT  COMPRESSOR-TYPE = VARIABLE-SPEED
                                CONDENSER-TYPE  = AIR-COOLED
                                DEFROST-TYPE    = RESISTIVE
                                DEFROST-CTRL    = TIMED
                                COOLING-EIR     = 0.3412
                                COIL-BF         = 0.19               ..

$ PACKAGED SINGLE ZONE FAN SYSTEM - VARIABLE VOLUME - VARIABLE
TEMPERATURE

SYST-E1     = SYSTEM  SYSTEM-TYPE = PVVT
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      SYSTEM-EQUIPMENT = S-EQU
                          OA-CONTROL = FIXED
                          ECONO-LIMIT-T=60
                          ECONO-LOW-LIMIT=60
                      HEAT-SOURCE = ELECTRIC
                      RETURN-AIR-PATH = DUCT
                      ZONE-NAMES = (SPACE-E1)     ..

SYST-E2     = SYSTEM  SYSTEM-TYPE = PVVT
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      SYSTEM-EQUIPMENT = S-EQU
                          OA-CONTROL = FIXED
                          ECONO-LIMIT-T=60
                          ECONO-LOW-LIMIT=60
                      HEAT-SOURCE = ELECTRIC
                      RETURN-AIR-PATH = DUCT
                      ZONE-NAMES = (SPACE-E2)     ..

SYST-E3     = SYSTEM  SYSTEM-TYPE = PVVT
                      SYSTEM-CONTROL= S-CONT
                      SYSTEM-FANS= S-FAN
                      SYSTEM-TERMINAL= S-TERM
                      SYSTEM-EQUIPMENT = S-EQU
                          OA-CONTROL = FIXED
```

```
                        ECONO-LIMIT-T=60
                        ECONO-LOW-LIMIT=60
                    HEAT-SOURCE = ELECTRIC
                    RETURN-AIR-PATH = DUCT
                    ZONE-NAMES = (SPACE-E3)     ..

SYST-E4     = SYSTEM  SYSTEM-TYPE = PVVT
                    SYSTEM-CONTROL= S-CONT
                    SYSTEM-FANS= S-FAN
                    SYSTEM-TERMINAL= S-TERM
                    SYSTEM-EQUIPMENT = S-EQU
                        OA-CONTROL = FIXED
                        ECONO-LIMIT-T=60
                        ECONO-LOW-LIMIT=60
                    HEAT-SOURCE = ELECTRIC
                    RETURN-AIR-PATH = DUCT
                    ZONE-NAMES = (SPACE-E4)     ..

SYST-I     = SYSTEM  SYSTEM-TYPE = PVVT
                    SYSTEM-CONTROL= S-CONT
                    SYSTEM-FANS= S-FAN
                    SYSTEM-TERMINAL= S-TERM
                    SYSTEM-EQUIPMENT = S-EQU
                        OA-CONTROL = FIXED
                        ECONO-LIMIT-T=60
                        ECONO-LOW-LIMIT=60
                    HEAT-SOURCE = ELECTRIC
                    RETURN-AIR-PATH = DUCT
                    ZONE-NAMES = (SPACE-I1,
                    SPACE-I2, SPACE-I3, SPACE-I4)     ..

PLANT-1  = PLANT-ASSIGNMENT
     SYSTEM-NAMES = (SYST-E1, SYST-E2, SYST-E3, SYST-E4, SYST-I)   ..

$-----HOURLY REPORTS-----$
PLTSCH2  = SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..

PLOTER21 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (8)     ..
                       $   8 - OUTDOOR DRY BULB TEMPERATURE F
PLOTER22 = REPORT-BLOCK
          VARIABLE-TYPE = PLANT-1
          VARIABLE-LIST = (3,5,6,7)         ..
PLOTER23 = REPORT-BLOCK
          VARIABLE-TYPE = END-USE
          VARIABLE-LIST = (1,3)          ..
LDS-2 = HOURLY-REPORT
        REPORT-SCHEDULE=PLTSCH2
        REPORT-BLOCK   = (PLOTER21, PLOTER22, PLOTER23)
        OPTION         = PRINT      ..

END  ..
COMPUTE SYSTEMS     ..
```

```
INPUT PLANT   ..

PLANT-1  =  PLANT-ASSIGNMENT ..

              PLANT-REPORT    SUMMARY=(ALL-SUMMARY,BEPS)
                              VERIFICATION = (ALL-VERIFICATION)  ..
CHIL1    =PLANT-EQUIPMENT  TYPE= HERM-CENT-CHLR  SIZE=-999  ..
          PLANT-PARAMETERS  HERM-CENT-COND-TYPE=AIR  ..

BOIL1    =PLANT-EQUIPMENT  TYPE= ELEC-HW-BOILER  SIZE=-999  ..
WATER1   =PLANT-EQUIPMENT  TYPE= ELEC-DHW-HEATER  SIZE=-999 ..

PLANT-COSTS      PROJECT-LIFE=25  DISCOUNT-RATE=5 ..

ENERGY-RESOURCE  RESOURCE=ELECTRICITY  ..
ENERGY-RESOURCE  RESOURCE=NATURAL-GAS  ENERGY/UNIT=100000
                 UNIT-NAME=THERMS   ..

$-----HOURLY REPORTS-----$

PLTSCH2  = SCHEDULE           THRU DEC 31 (ALL) (1,24) (1)  ..

PLOTER31 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (1)          ..
PLOTER32 = REPORT-BLOCK
          VARIABLE-TYPE = PLANT
          VARIABLE-LIST = (1,2,3)                                  ..
                      $  1 Heating load from SYSTEMS (Btu/hr)
                      $  2 Cooling load from SYSTEMS (Btu/hr)
                      $  3 Electric load from SYSTEMS (Btu/hr)
PLOTER41 = REPORT-BLOCK
          VARIABLE-TYPE = GLOBAL
          VARIABLE-LIST = (1)        ..
PLOTER42 = REPORT-BLOCK
          VARIABLE-TYPE = END-USE
          VARIABLE-LIST = (6,7,8,9)    ..
                                   $  6 Cooling Electric (kW)
                                   $  7 Heat Reject Electric (kW)
                                   $  8 Auxiliary Electric (pumps) (kW)
                                   $  9 Ventilation Electric (kW)
$ LDS-3 = HOURLY-REPORT
$        REPORT-SCHEDULE=PLTSCH2
$        REPORT-BLOCK   = (PLOTER31, PLOTER32)
$        OPTION         = PRINT      ..

END  ..
COMPUTE PLANT   ..
STOP ..
```

VITA

Name :	Oleksandr Tanskyi
Address:	Energy Systems Laboratory, MS 3581 Texas A&M University, College Station, TX 77843-3581
Email Address:	tanskyi@gmail.com

Education:

Dec 2010	M.S., Mechanical Engineering, Texas A&M University, College Station, Texas
Jun 2005	M.S., Electrical Engineering, National Technical University of Ukraine, Kyiv, Ukraine
Jun 2003	B.S., Electrical Engineering, National Technical University of Ukraine, Kyiv, Ukraine

Scholarships and Awards:

2008-2010	Fulbright Student Program scholarship
2002-2004	Joint Stock Company “Kyivenergo” scholarship for academic excellence
1999-2005	National Technical University of Ukraine scholarship for academic excellence

Professional Experience:

Since 05/2010	Texas Engineering Experiment Station. Energy System Laboratory. Graduate Assistant – Research. Analysis of HVAC systems control and retro commissioning of buildings
01/2006 - 12/2006	Denki (Kyiv, Ukraine). Research. Development of the State Standard of Ukraine
09/2005 - 05/2008	Diakom (Kyiv, Ukraine). Electrical Engineering. Design of electrical energy supply and building control systems