Climate-Specific Life-Cycle Cost Analysis of Different HVAC Systems

Xiuqing Jin

A thesis

submitted in partial fulfillment of the

requirements for the degree of

Master of Science in Construction Management

University of Washington

2017

Committee:

William Bender

Hyun Woo Lee

Program Authorized to Offer Degree:

Construction Management

University of Washington

Abstract

Climate-Specific Life-Cycle Cost Analysis of Different HVAC Systems

Xiuqing Jin

Chair of the Supervisory Committee:
Professor William Bender
Construction Management

The buildings sector consumes 41.1% of U.S. primary energy, and HVAC system accounts for the major part of building energy consumption. Each type of building has its occupancy schedule and operation preference, and different climate zones offer a broad range of temperature, humidity, wind and solar conditions. When selecting HVAC systems for a new project, designers and engineers should calculate a proper size of the heating and cooling equipment; owners want to know the initial cost and the life-cycle cost of the different options; contractors also need to have a good understanding of HVAC systems to complete the project with a higher quality.

This research involves eQUEST Energy Modeling and Life-Cycle Cost Analysis to compare the energy performance and the overall cost efficiency of different HVAC systems in various climate zones based on a typical educational office building. The selected systems are Variable

Air Volume (VAV) Reheat system, Chilled Beam system, Air Source Heat Pump (ASHP) system, and Ground Source Heat Pump (GSHP) system. The four climate zones are Miami (FL), Phoenix (AZ), Seattle (WA) and Spokane (WA).

The goal of this research is to illustrate a way of selecting the most suitable HVAC system for a project in the specific climate condition. This will be accomplished by using eQUEST Energy Modeling software and developing Life-Cycle Cost Analysis. The life-cycle cost includes the system capital cost, energy cost, system maintenance and replacement cost over a 20-year of life span. The life-cycle cost analysis provides the Present Value (PV) of annual cost and the life cycle cost, and it compares the accumulated cash flow curves of the sixteen models.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1. INTRODUCTION

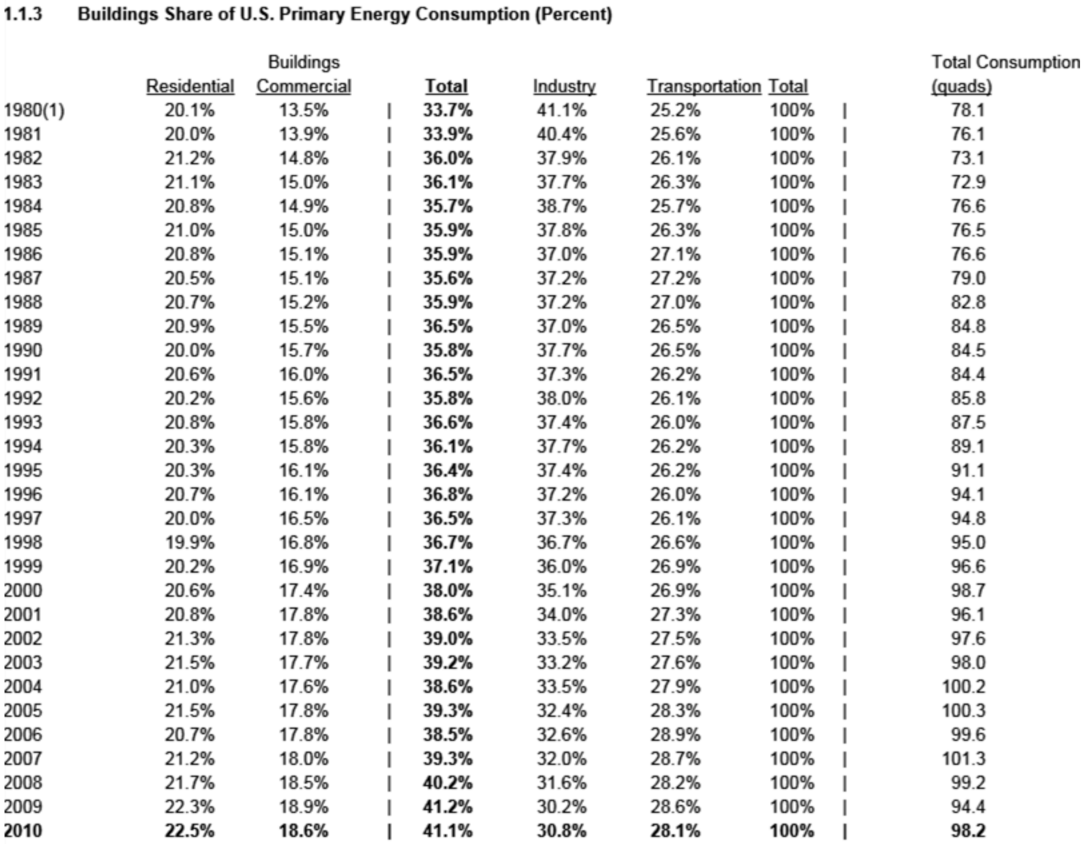
When a building turns into its occupancy phase after the construction completion, the building begins to consistently consume water and energy, which leads to the increasing the carbon dioxide emissions, global warming acceleration, and fossil fuel depletion. Among all the energy consuming sectors in buildings, Heating Ventilating and Air Conditioning (HVAC) systems expend the largest share of the total building energy consumption.

When selecting HVAC systems for a new project, the owner often prefers to use the one that is not only able to provide a pleasant indoor environment but also has a reasonable utility cost. The performance of each HVAC system depends on a variety of factors. There is no best HVAC system for all buildings. Every project has its unique feature and its particular challenges. Different types of buildings have different occupancy schedules, operation preferences; various climate zones offer a broad range of temperature, humidity, wind and solar conditions. Designers and engineers should consider all the information on a case-by-case basis to determine the most suitable HVAC system. It is also important for contractors to have a good understanding of HVAC systems to complete the project with better quality.

The scope of this research is restricted to a typical educational office building, in four climate zones, with four different HVAC systems. The research involves building energy modeling and life-cycle cost analysis method to calculate the annual energy consumption and the lifetime cost of various HVAC systems in the selected climate zones based on a typical educational office building. The goal of the research is to determine if the proposed method will help decide the most energy-efficient and economically sound HVAC system for the educational office building in different climate zones.

1.1 BUILDING ENERGY CONSUMPTIONS

Building Energy accounts for the largest portion of total energy consumption in the United States. According to U.S. Department of Energy, as shown in Figure 1.1-1, 41.1% of U.S. primary energy was consumed by the building sector, compared to 30.8% by the industrial sector and 28.1% by the transportation sector in 2010. (U.S. Department of Energy, 2011)



1.1.3 Buildings Share of U.S. Primary Energy Consumption (Percent)

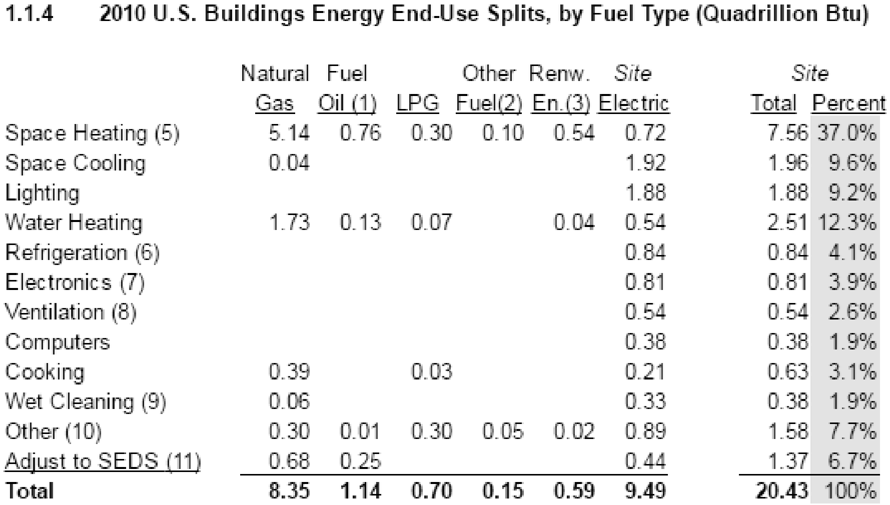
	Buildings						
	Residential	Commercial	Total	Industry	Transportation	Total	Total Consumption (quads)
1980(1)	20.1%	13.5%	**33.7%**	41.1%	25.2%	100%	78.1
1981	20.0%	13.9%	**33.9%**	40.4%	25.6%	100%	76.1
1982	21.2%	14.8%	**36.0%**	37.9%	26.1%	100%	73.1
1983	21.1%	15.0%	**36.1%**	37.7%	26.3%	100%	72.9
1984	20.8%	14.9%	**35.7%**	38.7%	25.7%	100%	76.6
1985	21.0%	15.0%	**35.9%**	37.8%	26.3%	100%	76.5
1986	20.8%	15.1%	**35.9%**	37.0%	27.1%	100%	76.6
1987	20.5%	15.1%	**35.6%**	37.2%	27.2%	100%	79.0
1988	20.7%	15.2%	**35.9%**	37.2%	27.0%	100%	82.8
1989	20.9%	15.5%	**36.5%**	37.0%	26.5%	100%	84.8
1990	20.0%	15.7%	**35.8%**	37.7%	26.5%	100%	84.5
1991	20.6%	16.0%	**36.5%**	37.3%	26.2%	100%	84.4
1992	20.2%	15.6%	**35.8%**	38.0%	26.1%	100%	85.8
1993	20.8%	15.8%	**36.6%**	37.4%	26.0%	100%	87.5
1994	20.3%	15.8%	**36.1%**	37.7%	26.2%	100%	89.1
1995	20.3%	16.1%	**36.4%**	37.4%	26.2%	100%	91.1
1996	20.7%	16.1%	**36.8%**	37.2%	26.0%	100%	94.1
1997	20.0%	16.5%	**36.5%**	37.3%	26.1%	100%	94.8
1998	19.9%	16.8%	**36.7%**	36.7%	26.6%	100%	95.0
1999	20.2%	16.9%	**37.1%**	36.0%	26.9%	100%	96.6
2000	20.6%	17.4%	**38.0%**	35.1%	26.9%	100%	98.7
2001	20.8%	17.8%	**38.6%**	34.0%	27.3%	100%	96.1
2002	21.3%	17.8%	**39.0%**	33.5%	27.5%	100%	97.6
2003	21.5%	17.7%	**39.2%**	33.2%	27.6%	100%	98.0
2004	21.0%	17.6%	**38.6%**	33.5%	27.9%	100%	100.2
2005	21.5%	17.8%	**39.3%**	32.4%	28.3%	100%	100.3
2006	20.7%	17.8%	**38.5%**	32.6%	28.9%	100%	99.6
2007	21.2%	18.0%	**39.3%**	32.0%	28.7%	100%	101.3
2008	21.7%	18.5%	**40.2%**	31.6%	28.2%	100%	99.2
2009	22.3%	18.9%	**41.2%**	30.2%	28.6%	100%	94.4
2010	**22.5%**	**18.6%**	**41.1%**	**30.8%**	**28.1%**	**100%**	**98.2**

Figure 1.1-1 Building energy consumption 2010 (U.S. Department of Energy, 2011)

Building energy consumption is based on a variety of factors, such as the building types, orientation, detail layout, climate, building envelope material, occupancy schedule, as well as design and operation of the HVAC systems.

HVAC systems consume the largest amount of energy in the total building energy consumption, which will largely contribute to the greenhouse gas emission and the fossil fuel depletion.

According to the U.S. Department of Energy, shown in Figure 1.1-2, the top three end uses are space heating (37%), water heating (12%) and space cooling (10%). Together with ventilation (3%) and refrigeration (4%), HVAC systems account for 66% of the total energy consumed by the building sector. (U.S. Department of Energy, 2012)



1.1.4 2010 U.S. Buildings Energy End-Use Splits, by Fuel Type (Quadrillion Btu)

	Natural	Fuel		Other	Renw.	*Site*	*Site*	
	Gas	Oil (1)	LPG	Fuel(2)	En.(3)	Electric	Total	Percent
Space Heating (5)	5.14	0.76	0.30	0.10	0.54	0.72	7.56	37.0%
Space Cooling	0.04					1.92	1.96	9.6%
Lighting						1.88	1.88	9.2%
Water Heating	1.73	0.13	0.07		0.04	0.54	2.51	12.3%
Refrigeration (6)						0.84	0.84	4.1%
Electronics (7)						0.81	0.81	3.9%
Ventilation (8)						0.54	0.54	2.6%
Computers						0.38	0.38	1.9%
Cooking	0.39		0.03			0.21	0.63	3.1%
Wet Cleaning (9)	0.06					0.33	0.38	1.9%
Other (10)	0.30	0.01	0.30	0.05	0.02	0.89	1.58	7.7%
Adjust to SEDS (11)	0.68	0.25				0.44	1.37	6.7%
Total	**8.35**	**1.14**	**0.70**	**0.15**	**0.59**	**9.49**	**20.43**	100%

Figure 1.1-2 Buildings Energy End-Use Splits (U.S. Department of Energy, 2012)

Energy use of the space cooling and heating is directly related to the heating and the cooling load of each part of a building. Cooling load means the amount of energy needed to be discharged per unit time to maintain the designed temperature and humidity within a space. (Burdick, 2011) Heating load means the amount of energy needed to be added per unit time to provide the desired level of temperature and humidity within a space.

Both heating load and cooling load include the sensible load and the latent load. The sensible load is the amount of heat exchanged by increasing or decreasing the temperature of the air without any phase transitions. Latent load, on the other hand, is the heat amount that leads to the phase transition of the air without any temperature changes, such as humidifying and dehumidifying of the air.

1.2 ENERGY MODELING

Building Energy Modeling is a computer-based simulation process that can predict the energy consumption of buildings. (Energy-Models.com, n. d.) It focuses on the energy consumption of a building, which mainly includes space heating, space cooling, air conditioning, lighting, hot water heating, and other equipment.

In the early schematic design stage of each project, many options are available for designers, such as the materials of building envelopes, opening sizes and locations, floor plans, lighting layouts, shading designs, and the different sizes of chillers and boilers.

Before Energy Modeling software became available, engineers had to calculate the peak heating load and the peak cooling load of the building. After that, the engineers can decide whether the wall assemblies are thick enough for insulation and whether the selected cooling equipment size is sufficient to cool the building on a hot summer day. However, it is not sufficient to calculate the annual heating and cooling energy consumption of the building.

Distinct from the lighting and equipment energy usage calculations, space heating and cooling energy consumption is much more complicated to calculate. Due to the outside air condition is changing all the time, the heating and the cooling load would not remain at the same value, so it requires numerous iterations of calculations to get the heating and cooling energy consumption of a project.

When performing an energy modeling, it allows engineers to import the climate information into the simulation program. As most of the weather data is in hourly times, energy simulation results always have 8,760 (=24h*365d) sets of data, which means the heating and cooling load are calculated hourly in the simulation program.

Other than the climate information, building energy simulation requires for more details and assumptions of the project. For example, the building structure, detail layouts, room functions, U-value and Solar Heat Gain Coefficient (SHGC) of envelope materials, occupancy schedules, and the specific HVAC systems.

Building Energy Modeling can provide information for architects and designers to make right decisions so that the building is not only comfortable but also energy efficient. Energy modeling can be used to calculate the payback period of green strategies such as photovoltaic solar panels, wind turbines, high-performance glazing systems, and geothermal heat pump systems. An energy model is also able to help the designers know what the optimal systems look like.

1.3 LIFE-CYCLE COST ANALYSIS

Life-Cycle Cost Analysis is a process of evaluating the economic performance of the total cost of ownership of a product over its entire lifetime. Life-Cycle Cost Analysis evaluates the initial investment, operation, and maintenance cost over the whole life cycle.

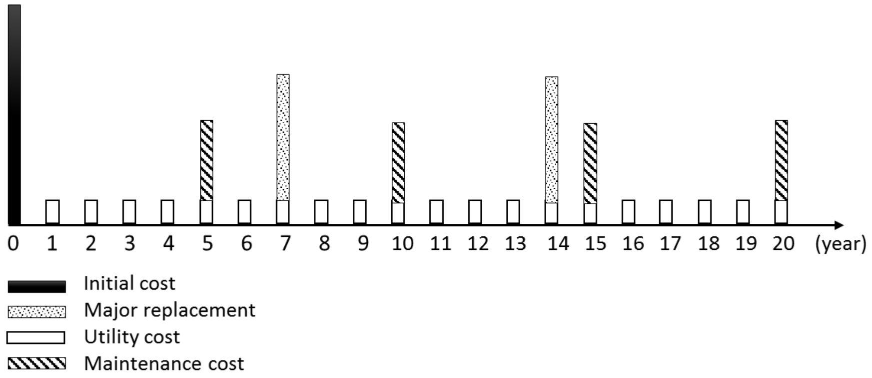
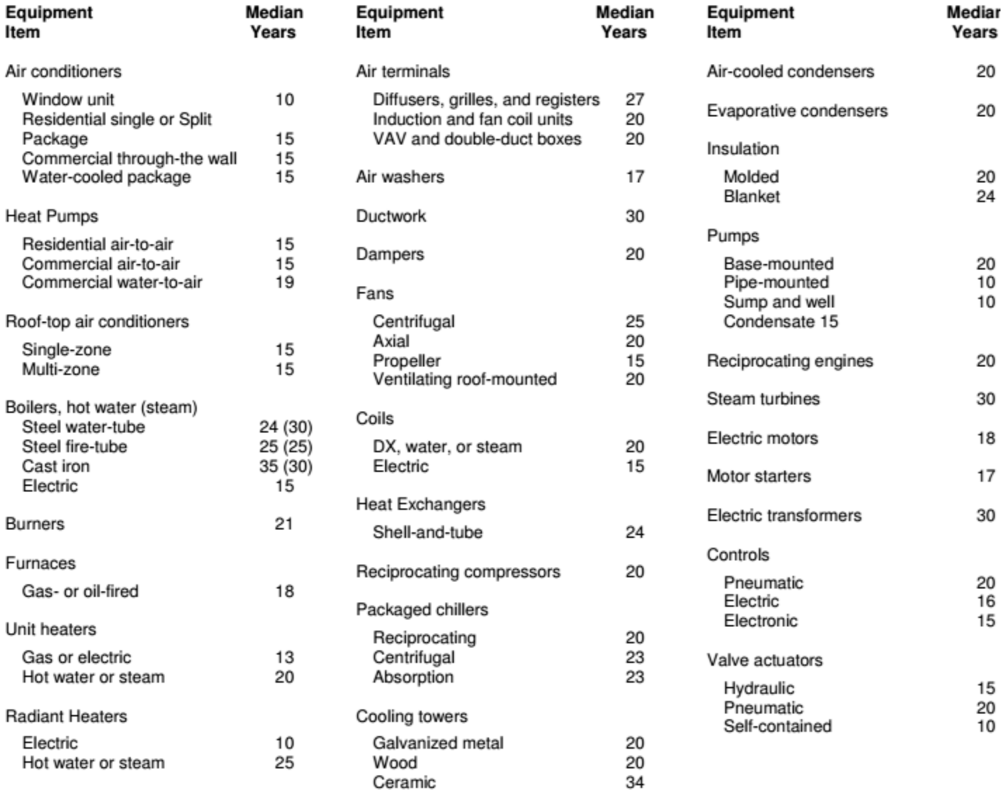


Figure 1.3-1 Life-cycle cost of HVAC Systems

Most of the life-cycle cost analysis in the field of AEC are comparing simple tools or building materials, but there is limited research done on the life cycle cost analysis of HVAC systems. Considering HVAC system is one of the features, which has the largest impact on the life-cycle cost, this reach focuses on the life-cycle cost analysis of HVAC systems.

High-efficiency HVAC systems, such as chilled beams and geothermal heat pumps, often require higher initial investments. However, over the long-term, those systems can be cost efficient over the conventional ones with lower initial cost. The energy price may not be the same in different locations; a high-performing system for one climate zone may be inefficient for another. Through comparing the life cycle cost of the HVAC systems for each location, stakeholders can see the whole financial conditions of all compared systems and easily make an appropriate decision for the project.

When developing a life cycle cost analysis, it is important to set a reasonable life span for the analysis. According to ASHRAE Equipment Life Expectancy Chart, Figure 1.3-2, the median value of the life expectancy for most HVAC equipment ranges from 15 years to 25 years. (ASHRAE, 2013) Considering the longevity of HVAC systems, this research uses a 20-year life span.



Equipment Item	Median Years
Air conditioners	
Window unit	10
Residential single or Split Package	15
Commercial through-the wall	15
Water-cooled package	15
Heat Pumps	
Residential air-to-air	15
Commercial air-to-air	15
Commercial water-to-air	19
Roof-top air conditioners	
Single-zone	15
Multi-zone	15
Boilers, hot water (steam)	
Steel water-tube	24 (30)
Steel fire-tube	25 (25)
Cast iron	35 (30)
Electric	15
Burners	21
Furnaces	
Gas- or oil-fired	18
Unit heaters	
Gas or electric	13
Hot water or steam	20
Radiant Heaters	
Electric	10
Hot water or steam	25

Equipment Item	Median Years
Air terminals	
Diffusers, grilles, and registers	27
Induction and fan coil units	20
VAV and double-duct boxes	20
Air washers	17
Ductwork	30
Dampers	20
Fans	
Centrifugal	25
Axial	20
Propeller	15
Ventilating roof-mounted	20
Coils	
DX, water, or steam	20
Electric	15
Heat Exchangers	
Shell-and-tube	24
Reciprocating compressors	20
Packaged chillers	
Reciprocating	20
Centrifugal	23
Absorption	23
Cooling towers	
Galvanized metal	20
Wood	20
Ceramic	34

Equipment Item	Mediar Years
Air-cooled condensers	20
Evaporative condensers	20
Insulation	
Molded	20
Blanket	24
Pumps	
Base-mounted	20
Pipe-mounted	10
Sump and well	10
Condensate 15	
Reciprocating engines	20
Steam turbines	30
Electric motors	18
Motor starters	17
Electric transformers	30
Controls	
Pneumatic	20
Electric	16
Electronic	15
Valve actuators	
Hydraulic	15
Pneumatic	20
Self-contained	10

Figure 1.3-2 ASHRAE Equipment Life Expectancy Chart (ASHRAE, 2013)

The costs of different HVAC systems is decided according to the previous case studies, papers, and cost data books such as RSMeans building construction cost data 2016. According to U.S. Department of Commerce, the National Institute of Standards and Technology, (Lavappa and Kneifel, 2016) the DOE discount and inflation rates of 2016 are as shown in the Table 1.3-1.

Table 1.3-1 DOE discount and inflation rates ((Lavappa and Kneifel, 2016))

Real rate (excluding general price inflation)	3.0 %
Nominal rate (including general price inflation)	2.6 %
Implied long-term average rate of inflation	-0.4 %

The electricity cost of the systems used in this analysis is collected from the energy modeling result through the eQUEST simulation software.

1.4 RESEARCH QUESTION

The goal of this research is to illustrate a way of selecting the most suitable HVAC system for a project in the specific climate condition. This will be accomplished by using eQUEST Energy Modeling and Life-Cycle Cost Analysis to compare the energy performance and the overall cost efficiency of four HVAC systems in four climate zones based on a typical educational office building. The selected systems are Variable Air Volume (VAV) Reheat system, Chilled Beam system, Air Source Heat Pump (ASHP) system, and Ground Source Heat Pump (GSHP) system. The four climate zones are Miami (FL), Phoenix (AZ), Seattle (WA) and Spokane (WA).

The eQUEST Energy Modeling simulation compares the building load characteristics of the four climate zones, and the total energy consumption of the four systems in each climate zones. The life-cycle cost includes capital cost, energy cost, maintenance and replacement cost over a 20-year of life span. The life-cycle cost analysis provides the Present Value (PV) of annual cost and the life cycle cost, and it compares the accumulated cash flow curves of the sixteen models.

CHAPTER 2. INDOOR ENVIRONMENT CONTROL

Long before people began to use mechanical ways to cool and heat their living spaces, our ancestors had adopted many sustainable approaches to the building construction to resist against the weather. (Zhu, 2010)

Other than the most basic building elements, such as the roofs overhead and the walls around to protect from the rain and wind, the builders at that time knew the ways to improve and adjust to the climate.

The early constructions in the ancient times appeared to be at the proper orientations, able to take advantage of the solar heat and daylight, has enough openings for natural ventilation, and with proper thermal mass. Those approaches, which enable the buildings to require little energy for space heating and cooling, are what we call “passive approaches” today.

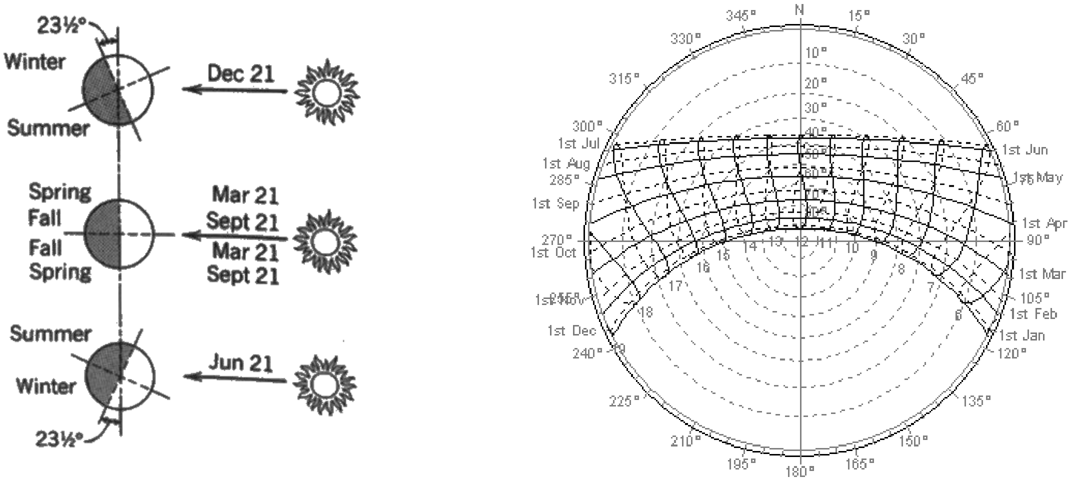
2.1 PASSIVE APPROACHES

2.1.1 Solar Radiation

The sun provides a significant amount of heat and daylight with its radiation to earth.

The amount of solar radiation that reaches the Earth’s surface is called “Insolation,” which is from Incident Solar Radiation. Insolation level is defined by the amount of solar radiation received per unit area (kWh/m^2).

The Earth’s seasons, dates, and hours are depended by the changes of the insolation, as shown in the figure 2.1-1 (a) (Stein, B., Reynolds, Grondzik, and Kwok, 2006, p. 150) and Figure 2.1-1 (b). (Ecotect Solar Tool)



(a) (b)

Figure 2.1-1 (a) Tile of the Earth's axis in the plane of the ecliptic results in the seasonal variations (Stein, B., Reynolds, Grondzik, and Kwok, 2006, p. 150); (b) Sun position variations in the sun-path diagram (Autodesk Ecotect Analysis, Version 2011)

For the north hemisphere, the sun is toward the south, so it has a larger altitude angle during summer and a smaller altitude angle during winter, Figure 2.1-2. (Stein, B., Reynolds, Grondzik, and Kwok, 2006, p. 151)

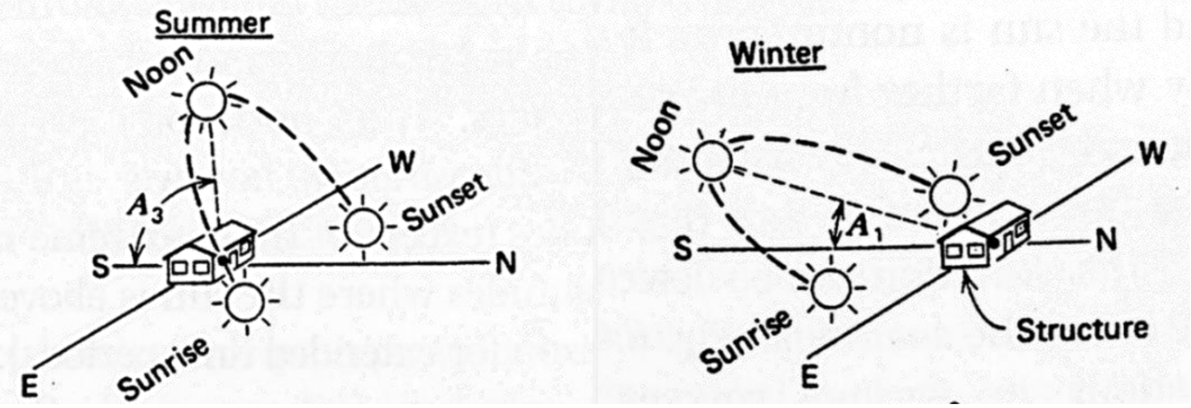


Figure 2.1-2 Approximate positions of the sun on a summer day and a winter day at a mid-northern latitude about 45° (Stein, B., Reynolds, Grondzik, and Kwok, 2006, p. 151)

Nicolow and Sami (2006) noted that "One of the keys to reducing building energy needs is the configuration of building massing and glazing to take maximum advantage of building orientation." (p. 36) .

Since far ago, in the north part of China, the first feature people have been considered when buying their houses are whether or not the major orientation of the house is south. The reason is that the south side of the building can get most of the solar radiation during winter and can avoid being overheated during summer. If the major orientation is south, it will be very easy to shade the strong summer sun with a horizontal shading device.

However, due to the lack of buildable land area and the increasing demand for more and more office spaces in the downtown area of most large cities, there are many high-rise office buildings in the downtown area, which major orientation is toward east/west. Even worse, some of those buildings have full glass curtain walls installed on the east and west side.

In summer, the sun rises early, and it is quite intense, office spaces with large windows on the east side will be heated up before 9:00 a.m. when the occupants arrive at the offices and get ready to work. In the afternoons at about 3:00 p.m. the position of the sun is on the west, and the radiation enters directly from the west curtain wall into the office spaces, which again increases the cooling load. All those situations above will cost more energy to heat and cool the space.

Solar radiation can provide endless heating and lighting energy, but it is necessary to adjust the amount of solar radiation a building would get; balance the use of shading devices to avoid unnecessary solar heat gain and ensure enough daylight for the occupant spaces.

There are two parts of solar radiation: direct solar radiation and diffuse solar radiation. Direct solar radiation is solar radiation traveling in a straight line from the sun to the earth. On the other hand, diffuse solar radiation is used to describe solar radiation reaching the earth after being scattered from the direct solar beam by molecules or particles in the atmosphere.

2.1.2 Building Envelope

Building envelope acts as a partition, which isolates the indoor air from the outdoor air. It keeps the indoor spaces warm and safe from cold wind and rains. Building loses heat through its envelope and gets the external heat gain, such as solar radiation and higher outdoor air temperature through its envelope. Also, building envelope can provide the occupants with cool shadows in hot summer afternoons.

Building envelope can be divided into two different part: opaque envelope and transparent envelope. (Zhu, 2010) For opaque building envelopes, such as walls, roofs, and ground floors, U-value and R-value are the most important factors; for transparent building envelopes, such as glass curtain walls, windows, and skylights, U-value and SCHG should be considered preferentially.

ASHRAE 90.1-2010 sets requirements for the baseline of each envelope element in different climate zones in detail, which provides designer a guideline in material selection so that the envelope can provide a sound insulation performance.

On account of the heat gain through the solar radiation and the internal heat generation, the temperature inside the building envelope can be increased by a certain amount, so the air temperature within a building is often higher than outside during winter. However, during summer, the envelope is the only path where the building can discharge its unnecessary heat.

If a building has a thick external wall with high R-value and a low window to wall ratio, it will provide an excellent insulation during winter, but it will not be a wise way if the building is in a climate with hot summers.

In some equator regions in Africa, the walls and windows are designed to connect the indoor space with the outdoor space, as shown in the following Figure 2.1-3, (China Zhongyuan Engineering Corp., 2014) where the envelope does not have any insulation effect at all.

The reason why core spaces of office buildings always require for cooling even during cold winters is by the same logic. With all the computers, printers, lights, and people, the inner space has too much internal heat gain, but there is no way to discharge the extra heat.

The building envelope is like the skin of a human body, it not only needs insulation to keep warm, but it also requires for respiration to discharge extra heat. The process to decide a balance between the thermal resistance and the ability to discharge heat of building envelope materials is necessary.

Figure 2.1-3 Building Envelope of Typical Buildings at the University of Dar es Salaam in Tanzania (China Zhongyuan Engineering Corp., 2014)

2.1.3 Natural Ventilation

Through ventilation, the air within the indoor space can be replaced by the fresh outdoor air, which reduces contaminant rate of the indoor air and increases the productivity of the indoor occupants. Natural ventilation during spring, autumn and cool summer night is an effective strategy to reduce cooling load of a building.

The building needs to have enough operable windows so that it can provide an adequate amount of fresh air passively to the space. The operable window area can be decided based on the wind pressure on each orientation, and the required air change rate of each space. For the locations where the wind pressure is low, or a building is not able to have too many operable windows, designers should consider taking advantage of the heat pressure.

An example of utilizing heat pressure in natural ventilation is solar chimney, or stack effect, as shown in the following Figure 2.1-4. (Solar chimney, 2017)

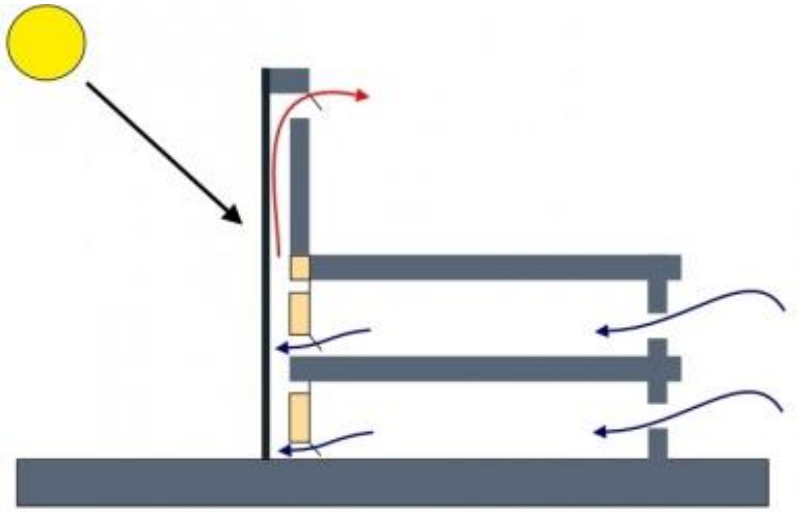


Figure 2.1-4 Solar Chimney Stack Effect (Solar chimney, 2017)

Solar chimneys are higher than the other part of the building structures, and they are always designed for the orientation where it can get the most sunlight. When the sun begins to radiate, the air inside the chimney is getting hotter. As hot air rises, the air inside the chimney will rise and get out through the outlet on the top of the stack, which reduces the air pressure of the building and draws more air in the building. As the top of the chimney is heated by the sun, causing cooler air to be drawn in through the shaded windows, which allows fresh air to move throughout the building when the weather is hot.

2.1.4 Summary

Passive approaches such as building orientation, solar radiation, and ventilation are the cheapest strategies to adjust the built environment and human comfort if adopted early during schematic design. Those strategies can create a better indoor environment and cut down the energy consumption effectively.

The effectiveness of improving indoor environment by using those strategies are limited. In those extreme climate zones, passive approaches can only help make a small improvement to the indoor environment. Active approaches to indoor environment control allow people move to and live in severer climate conditions.

2.2 ACTIVE APPROACHES

2.2.1 Electrical Lighting and Control

Daylight from the solar radiation a natural source of lighting. However, it is not available during the night, and it varies time to time. The luminous environment involves the consideration of illuminance (generally, 100-2000 Lux is appropriate for reading and working), luminance level (maximum of 3000 cd/m^2), and the visual comfort.

Electrical lighting provides the occupants additional illuminance during nights and overcast days. Also, when the sun causes too much glare, occupants can put down a dark curtain and adjust the light level inside with electrical lighting, which is an easy way to adjust illuminance and luminance requirement, especially in the Arctic region.

Regarding the energy conservation and green strategies, modern buildings also utilize LED lightings, occupancy sensors, and daylight sensors to detect the presence of the occupants and the amount of daylight level. When there is no person in the space or when the natural daylight can provide enough daylight, the electrical lights will be automatically turned off.

2.2.2 Heating and Cooling

Due to the solar radiation and the internal heat generation, the indoor air temperature is always higher than the outside. Mechanical heating technology enabled people to move to those extreme climates.

The indoor fireplaces, Korean 'Ondol' (500-400 BCE), Roman Hypocaust (100 BCE), Charcoal Brazier (1000-1400), the Japanese Hibachi (1603-1867), and radiators (19th century) (Sami, 2015) noted the evolution of active heating. Mechanical heating technology comes earlier than mechanical cooling. The primary reason is that the difference between the outdoor air temperature and the desired comfortable temperature is larger during winter. Heating has become a necessity for survival. Substantial areas of North America and North Asia where the summers are so mild and cool, but the winters are so cold that heating systems are installed but not cooling systems.

The first thing of mechanical cooling is the electric fan invented by Schuyler Wheeler, in 1886 (Sami, 2015). Since then, electric fans have become a principal way of home cooling and has been widely used all around the world. In 1922, the centrifugal chiller was invented (Sami, 2015), which stimulated the development of architecture, construction, engineering, and accelerated the expansion of cities.

Moisture control is an essential process of mechanical systems. Heating and cooling involve sensitive heat change and latent heat change.

According to the Figure 2.2-1, increasing the dry-bulb temperature of the air causes the relative humidity to decrease, which means heating dehumidifies the indoor air; on the other hand, cooling, which decreases the dry-bulb temperature of the air will cause the relative humidity of the indoor air to increase. Inappropriate moisture control will result in poor thermal comfort and even condensation issues, which can cause a mold problem.

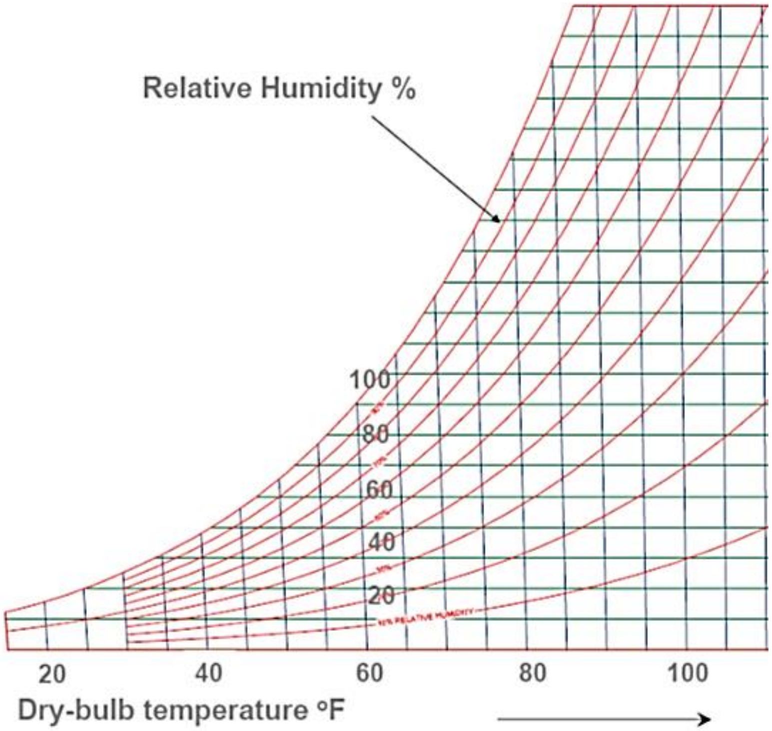


Figure 2.2-1 Psychrometric Chart of Dry-bulb temperature and Relative Humidity

2.2.3 Mechanical Ventilation

During spring, autumn, and cool summer night, natural ventilation can bring fresh air into the indoor spaces. However, when the weather is cold or hot, natural ventilation increases the heating load or the cooling load of the building, which will waste a significant amount of energy to heat or cool the building. When the outdoor air quality is poor, introducing outdoor air directly into the building will increase the contamination concentration in the building.

Through mechanical ventilation, outdoor air is filtered and preheated/precooled and then introduced to the indoor area, which makes it possible to maintain a proper indoor air quality level even when the outdoor air is too cold, hot, or contaminated.

2.2.4 Summary

A typical building usually uses both passive and active approaches to adjust to control the indoor environment. Passive approaches are effective in the solar control, lighting, air ventilation and so on.

Active approaches, on the other hand, are used for same purposes with expenses of energy. However, those active approaches are more efficient, and the active approaches have been continually improving the living conditions and have accelerated the development and expansion of the modern society.

CHAPTER 3. CLIMATE ANALYSIS

3.1 CLIMATE ZONE

The climate is a statistical representation of weather over a period. According to the map showing DOE climate zones, Figure 3.1-1, (ASHRAE, 2010) from ASHRAE 90.1-2010, the United States can be divided into eight parts based on the temperature and into three parts based on the humidity.

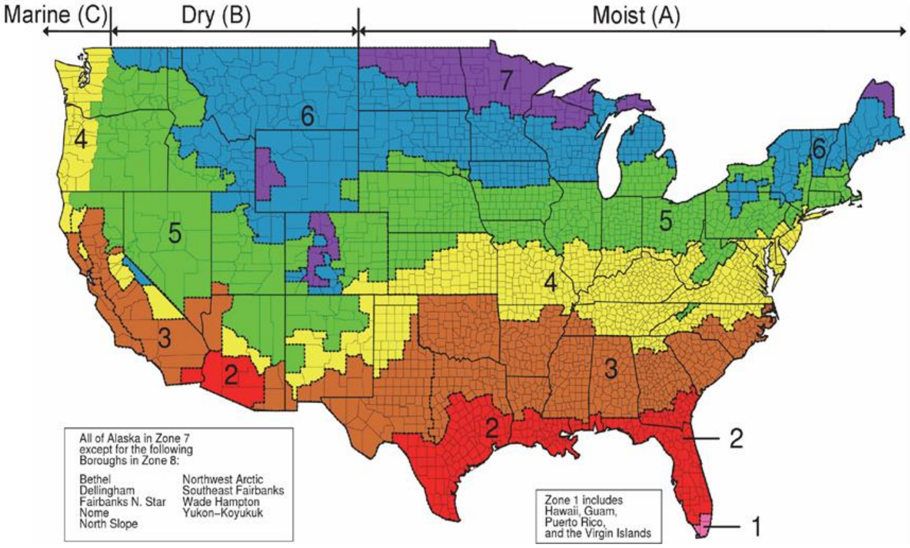


Figure 3.1-1 U.S. Climate Zones (ASHRAE, 2010)

3.2 DATA SOURCES

When performing a climate analysis, there are several major types of data to focus on. These are dry bulb temperature, humidity, sun path, sky cover range, solar radiation, and wind condition. This research concentrates on the features, which directly relate to the HVAC system performance: dry bulb temperature, relative humidity, and solar radiation. In this climate analysis, a psychrometric chart with comfort zone marked for each location indicates what kind of actions should be taken to provide an indoor comfort condition. All climate data is retrieved from Climate Consultant 6.0.

The scope of this research is limited to four cities in different climate zones of the U.S. These 4 zones represent the variations in the U.S. Climate. Each of the city studied in this research has its unique characteristic. The selected cities are Miami, Phoenix, Seattle, and Spokane, as shown in the following Table 3.2-1 and Figure 3.2-1.

Table 3.2-1 Selected Locations

City	Climate Zone	Description
Miami	1A	Hot-Moist
Phoenix	2B	Hot-Dry
Seattle	4C	Mild-Marine
Spokane	5B	Cold-Dry

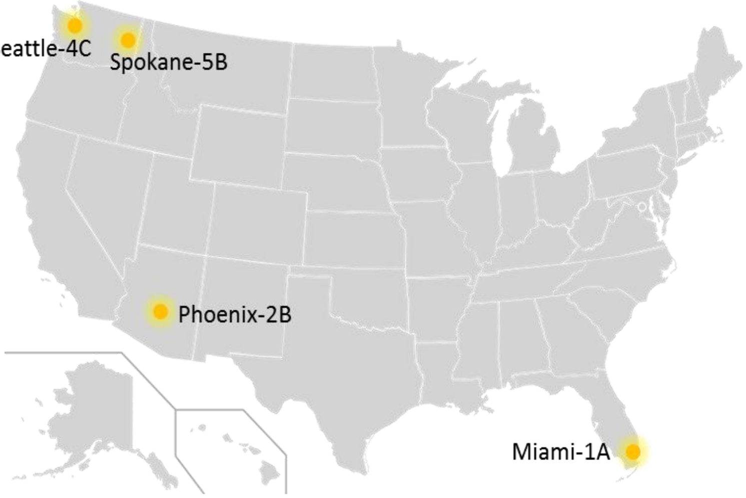


Figure 3.2-1 Selected Locations

3.3 CLIMATE DATA

3.3.1 Miami-1A

Miami locates at 25.82° N, 80.3° W, the southeastern corner of the state of Florida, Figure 3.3-1. Miami experiences a tropical wet climate.

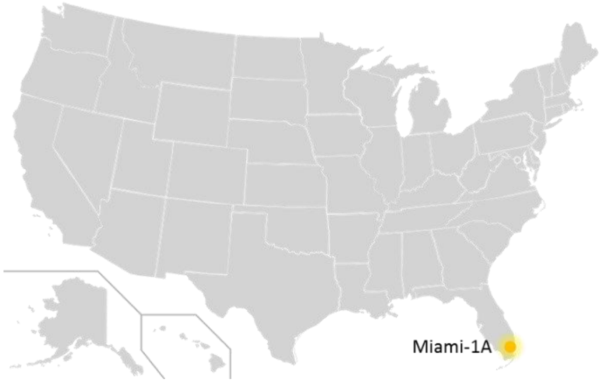


Figure 3.3-1 Miami

According to Figure 3.3-2 Miami hourly dry bulb temperature, the dry bulb temperature during the year has little variation, especially from April to October. The average monthly temperature is around 67°F to 83°F, the annual lowest temperature is 41°F in February, and the highest temperature is 96°F in July.

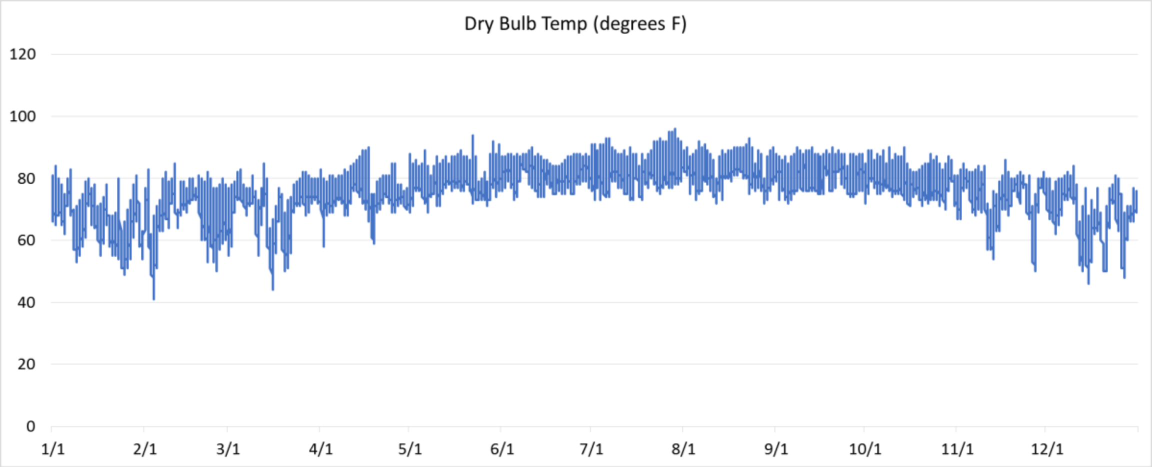


Figure 3.3-2 Miami Hourly Dry Bulb Temperature (Climate Consultant, Version 6.0)

According to Figure 3.3-3 Miami hourly relative humidity, the relative humidity in Miami is very high during the whole year. The average monthly relative humidity is around 67% and 77%. During the summer period, from June to October, the relative humidity is never lower than 40%. This humidity level requires higher energy to dehumidify the indoor air. The condensation issue is also a challenge.

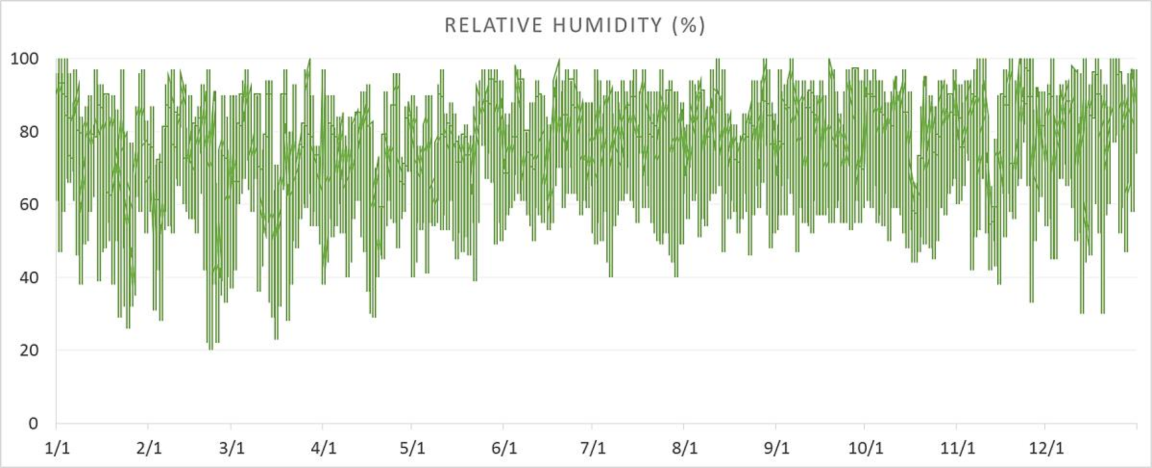


Figure 3.3-3 Miami Hourly Relative Humidity (Climate Consultant, Version 6.0)

Figure 3.3-4 shows the hourly direct solar radiation of Miami. The average monthly direct solar radiation is between 83Btu/SF and 130 Btu/SF, and the annually highest direct solar radiation 320 Btu/SF appears in January.

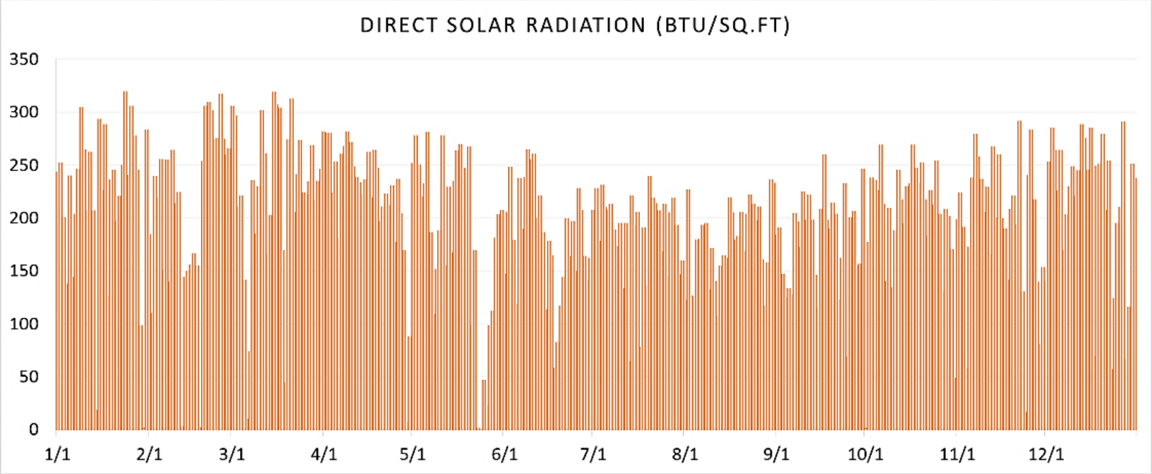


Figure 3.3-4 Miami Annual Direct Solar Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-5, the diffuse solar radiation of Miami is relatively high, which expects to have a good daylight condition. The monthly average diffuse solar radiation is between 41.52 Btu/SF and 71.26 Btu/SF.

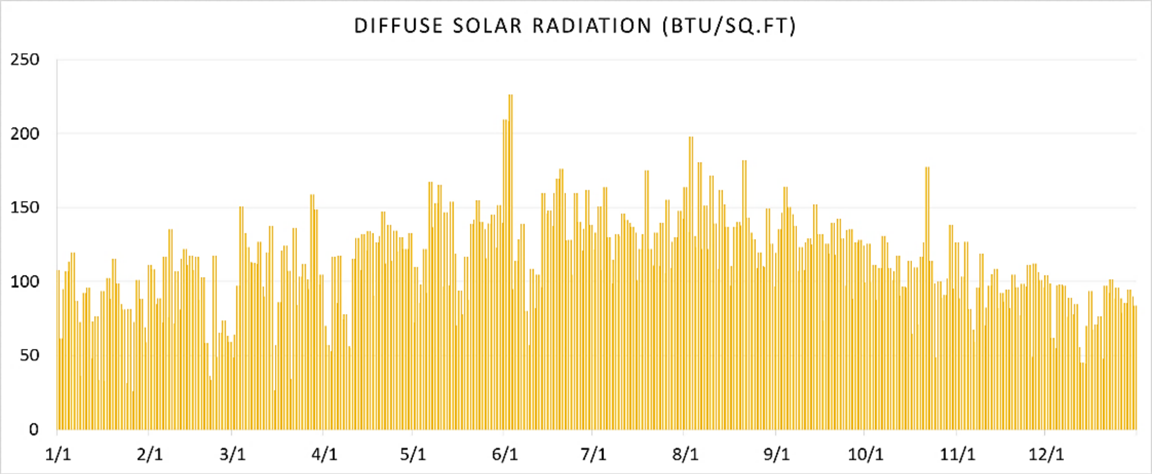
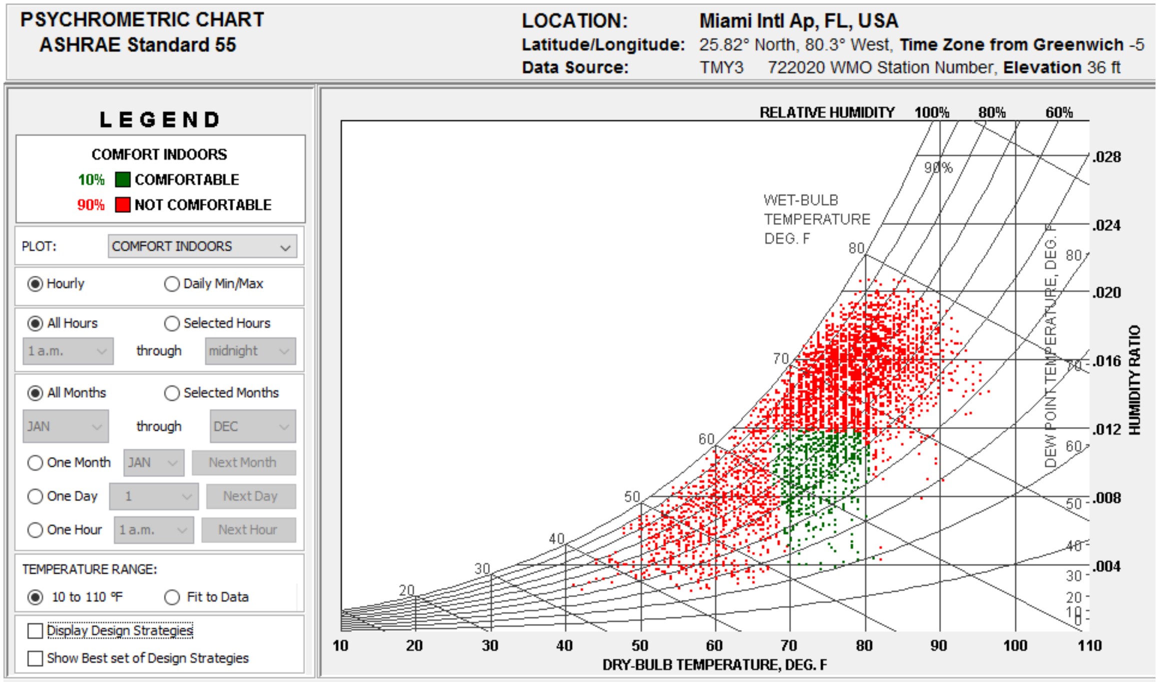


Figure 3.3-5 Miami Annual Diffuse Solar Radiation (Climate Consultant, Version 6.0)

From Figure 3.3-6 psychrometric chart of Miami, there are10% of the hours during a year is considered comfortable, and there are a large number of the points that located above the green comfort zone, which means the air need dehumidification during those hours of the year, when the temperature and humidity are high.



PSYCHROMETRIC CHART
ASHRAE Standard 55

LOCATION: Miami Intl Ap, FL, USA
Latitude/Longitude: 25.82° North, 80.3° West, **Time Zone from Greenwich** -5
Data Source: TMY3 722020 WMO Station Number, **Elevation** 36 ft

LEGEND

COMFORT INDOORS
10% 🟩 COMFORTABLE
90% 🟥 NOT COMFORTABLE

PLOT: COMFORT INDOORS
● Hourly ○ Daily Min/Max
● All Hours ○ Selected Hours 1 a.m. through midnight
● All Months ○ Selected Months JAN through DEC
○ One Month JAN Next Month
○ One Day 1 Next Day
○ One Hour 1 a.m. Next Hour
TEMPERATURE RANGE: ● 10 to 110 °F ○ Fit to Data
☐ Display Design Strategies
☐ Show Best set of Design Strategies

Figure 3.3-6 Psychrometric Chart of Miami (Climate Consultant, Version 6.0)

3.3.2 Phoenix-2B

Phoenix is located at 30.45° N, 111.98° W, in the center of Arizona. Phoenix has a subtropical desert climate.

Figure 3.3-7 Phoenix

According to Figure 3.3-8 Phoenix hourly dry bulb temperature, Phoenix has severe temperature fluctuations both daily and over a year. The average monthly temperature is around 53°F to 96°F, the annual lowest temperature is 36°F in December, and the highest temperature is 112°F in July.

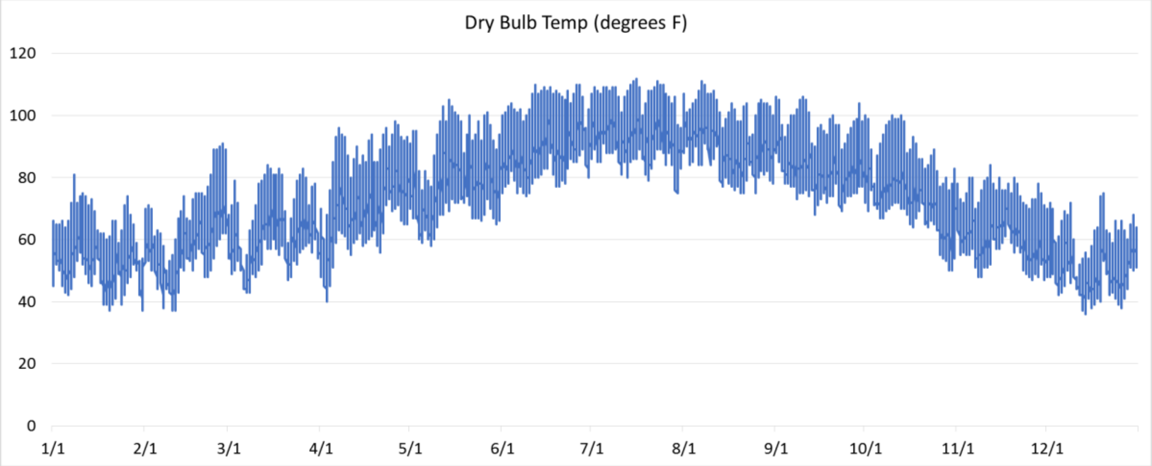


Figure 3.3-8 Phoenix Annual Dry Bulb Temperature (Climate Consultant, Version 6.0)

As the Figure 3.3-9 shows, the relative humidity in Phoenix is low. The average monthly relative humidity is between 19% and 52%, which enables the temperature to fluctuate. As water has a high heat capacity, dryer air with low moisture content is easier to get or lose heat. The lowest relative humidity appears in April and May, which is 3%. During the summer, the humidity of the air is even lower, and condensation is not an issue in Phoenix.

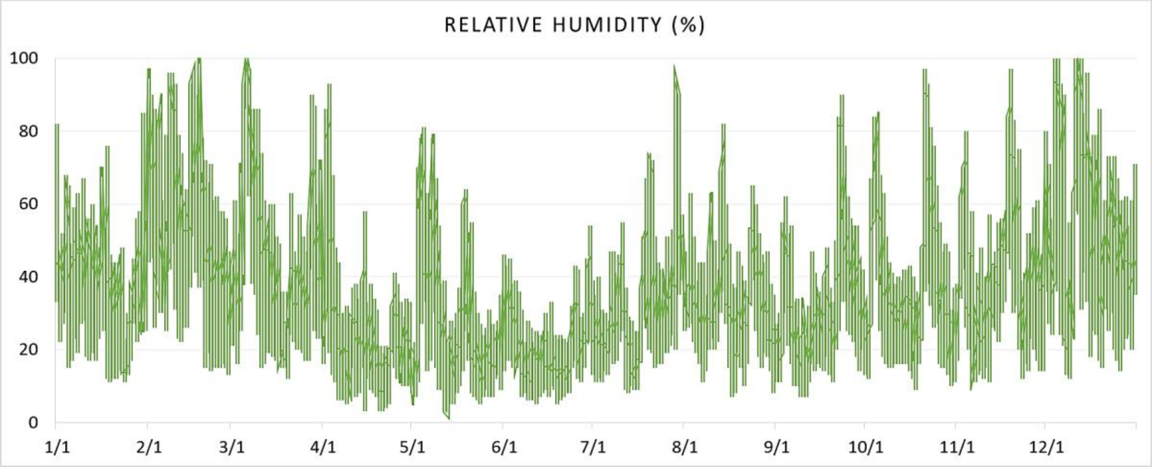


Figure 3.3-9 Phoenix Hourly Relative Humidity (Climate Consultant, Version 6.0)

Figure 3.3-10 shows the hourly direct solar radiation of Phoenix. Phoenix has an extremely high solar radiation. Therefore it is a great location to utilize solar energy. The average monthly direct solar radiation is between 165Btu/SF and 199 Btu/SF, which is almost twice the amount of Miami. The annually highest direct solar radiation 328 Btu/SF appears in April.

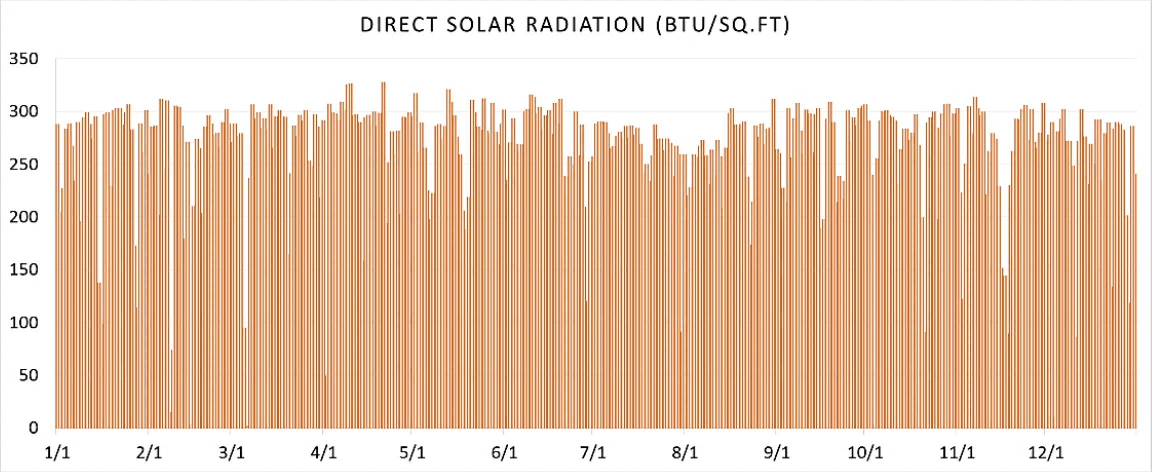


Figure 3.3-10 Phoenix Hourly Direct Solar Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-11, the diffuse solar radiation of Phoenix is relatively low. The monthly average diffuse solar radiation is between 28Btu/SF and 50Btu/SF.

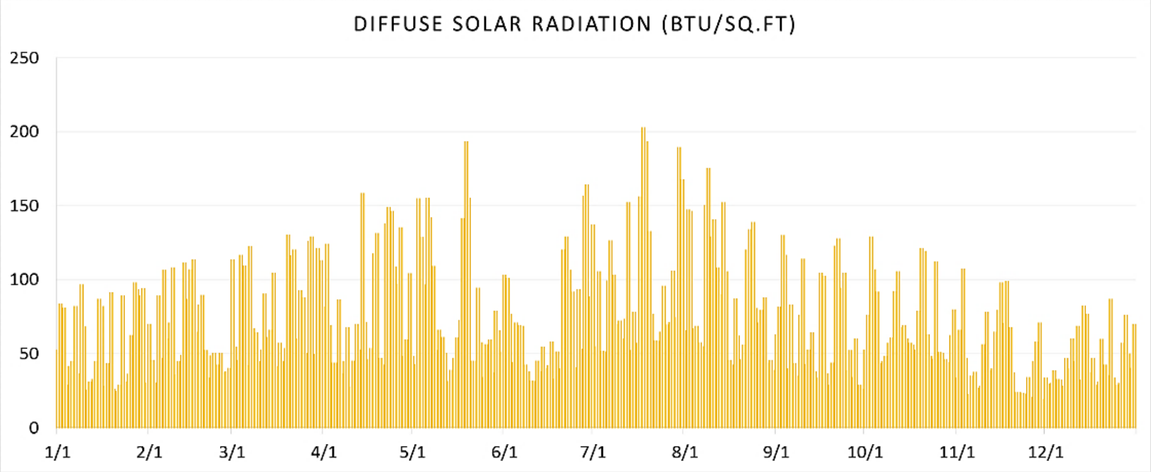
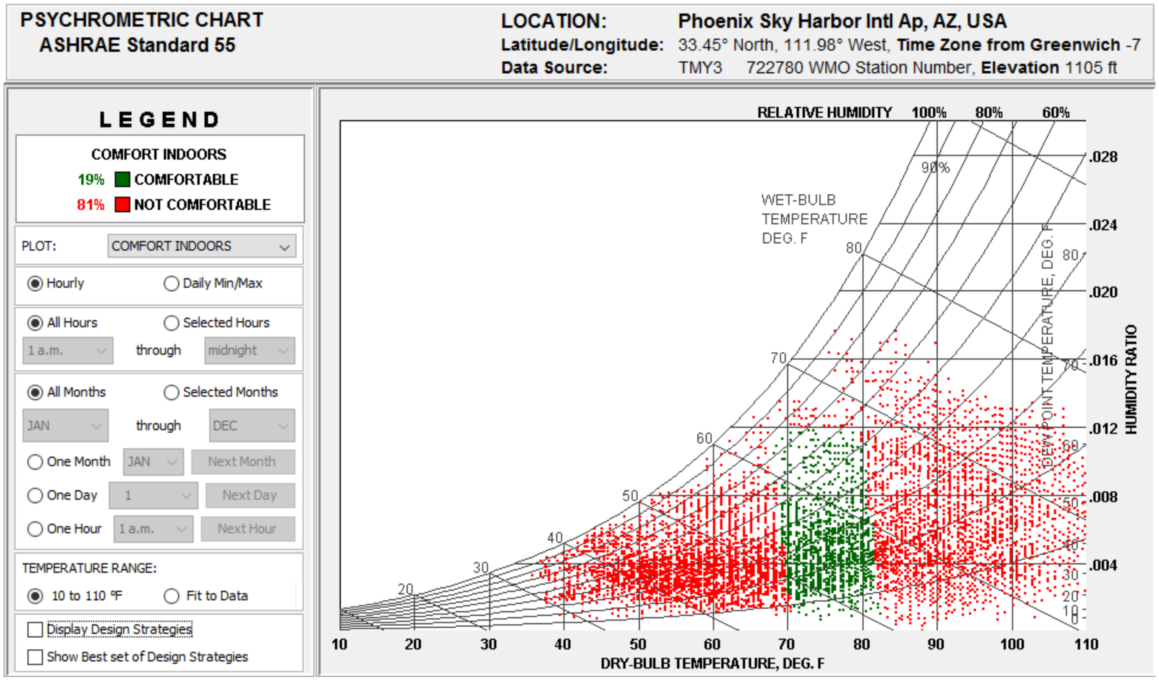


Figure 3.3-11 Phoenix Hourly Diffuse Solar Radiation (Climate Consultant, Version 6.0)

From Figure 3.3-12, the points scattered all over the chart, which indicates significant fluctuations in temperature. The humidity in Phoenix is low. During winter, when the air is heated, it would get even drier, so humidification is desired for Phoenix, especially in the wintertime.



PSYCHROMETRIC CHART
ASHRAE Standard 55

LOCATION: **Phoenix Sky Harbor Intl Ap, AZ, USA**
Latitude/Longitude: 33.45° North, 111.98° West, **Time Zone from Greenwich** -7
Data Source: TMY3 722780 WMO Station Number, **Elevation** 1105 ft

LEGEND

COMFORT INDOORS
19% 🟩 COMFORTABLE
81% 🟥 NOT COMFORTABLE

PLOT: COMFORT INDOORS
● Hourly ○ Daily Min/Max
● All Hours ○ Selected Hours
1 a.m. through midnight
● All Months ○ Selected Months
JAN through DEC
○ One Month JAN Next Month
○ One Day 1 Next Day
○ One Hour 1 a.m. Next Hour
TEMPERATURE RANGE:
● 10 to 110 °F ○ Fit to Data
☐ Display Design Strategies
☐ Show Best set of Design Strategies

Figure 3.3-12 Psychrometric Chart of Phoenix (Climate Consultant, Version 6.0)

3.3.3 Seattle-4C

Seattle is located at 47.47° N, 122.32° W, at the northwest corner of the Washington State.

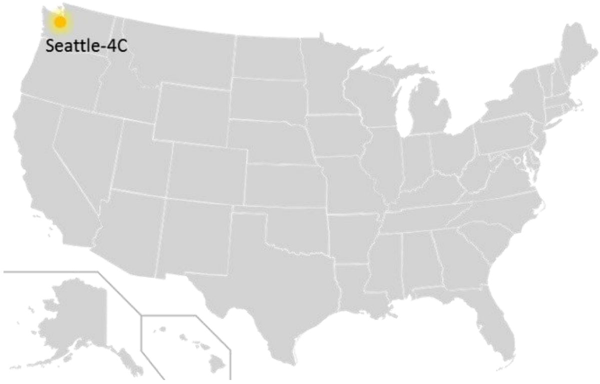


Figure 3.3-13 Seattle

According to Figure 3.3-14 Seattle hourly dry bulb temperature, the dry bulb temperature during the year is relatively stable. The average monthly temperature is around 40°F to 66°F, the annual lowest temperature is 22 °F in February, and the highest temperature is 89°F in August.

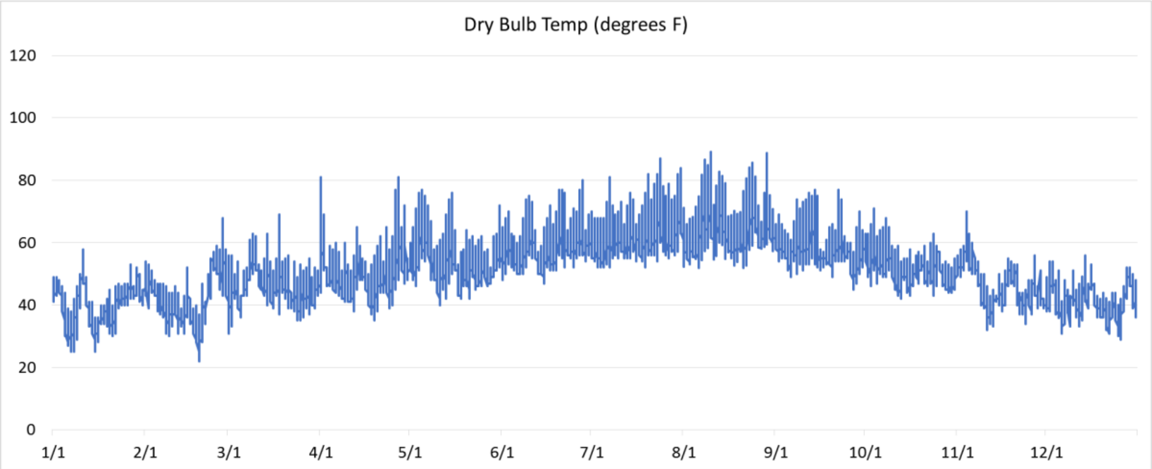


Figure 3.3-14 Settle Hourly Dry Bulb Temperature (Climate Consultant, Version 6.0)

According to Figure 3.3-15 Seattle hourly relative humidity, the relative humidity in Seattle is high during the whole year. The average monthly relative humidity is around 67% and 77%. Due to the high precipitation, the relative humidity is greater in the autumn and winter, between September and February. High humidity during the winter might cause issues with radiation heating systems, especially close to an operable window. This is because it is cold outside and warm inside and there may be condensation on the inside of windows.

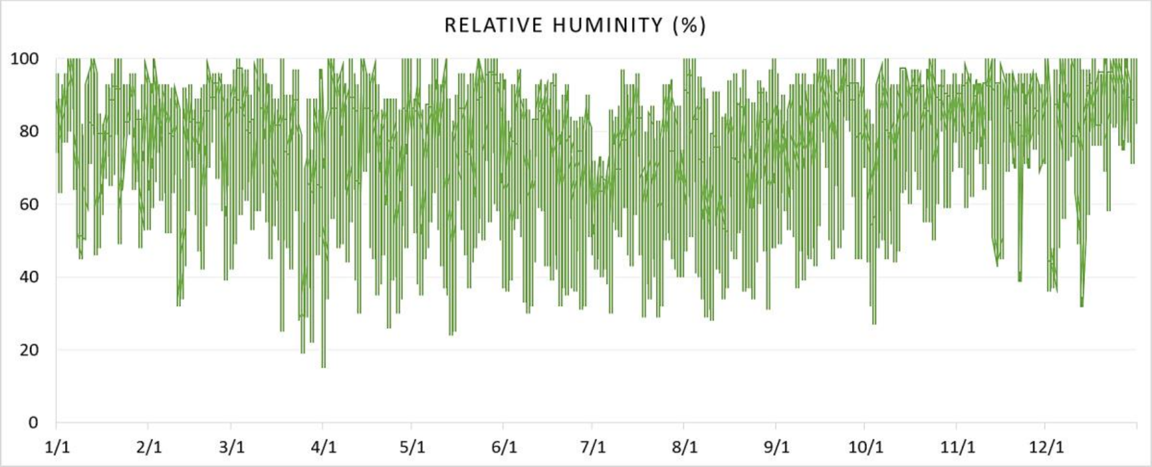


Figure 3.3-15 Seattle Annual Relative Humidity (Climate Consultant, Version 6.0)

Figure 3.3-16 shows the annual direct solar radiation of Seattle. As its high latitude and frequent rainfall during winter, the direct solar radiation is extremely low in Seattle. The average monthly direct solar radiation is between 39Btu/SF and 123Btu/SF. The annually highest direct solar radiation 298 Btu/SF appears in April.

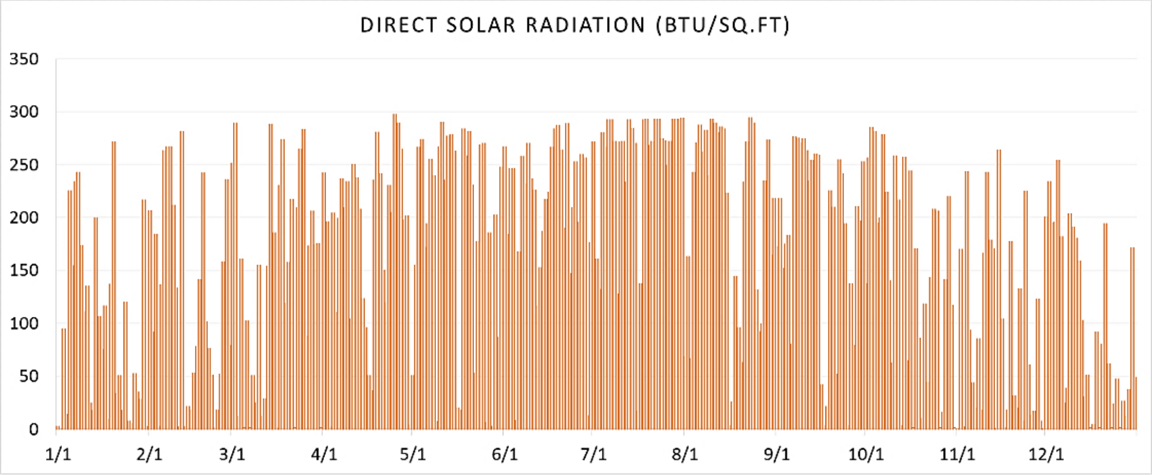


Figure 3.3-16 Seattle Hourly Direct Solar Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-17, the diffuse solar radiation of Seattle is low in the winter, even the highest monthly diffuse radiation between November and February is less and 90Btu/SF. The monthly average diffuse solar radiation is between 22Btu/SF and 59Btu/SF.

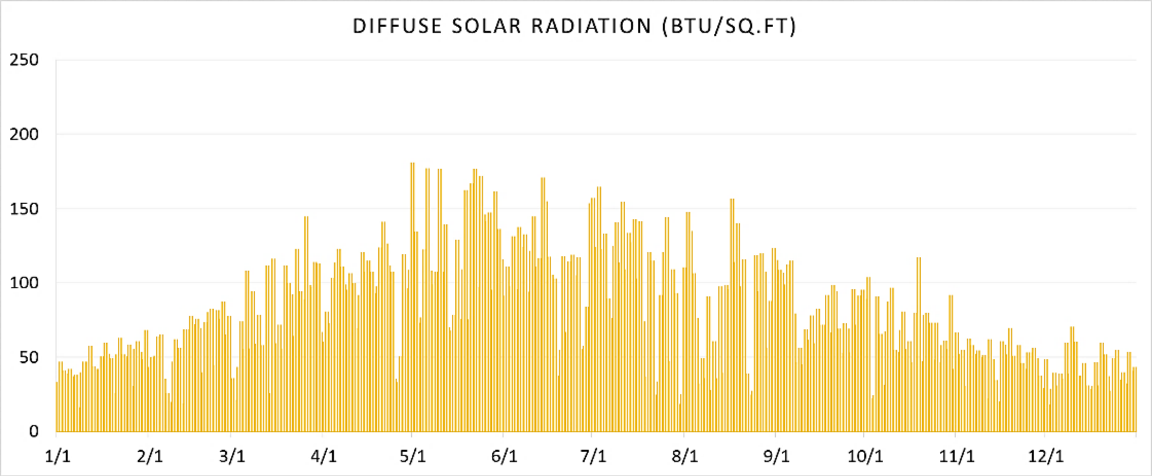
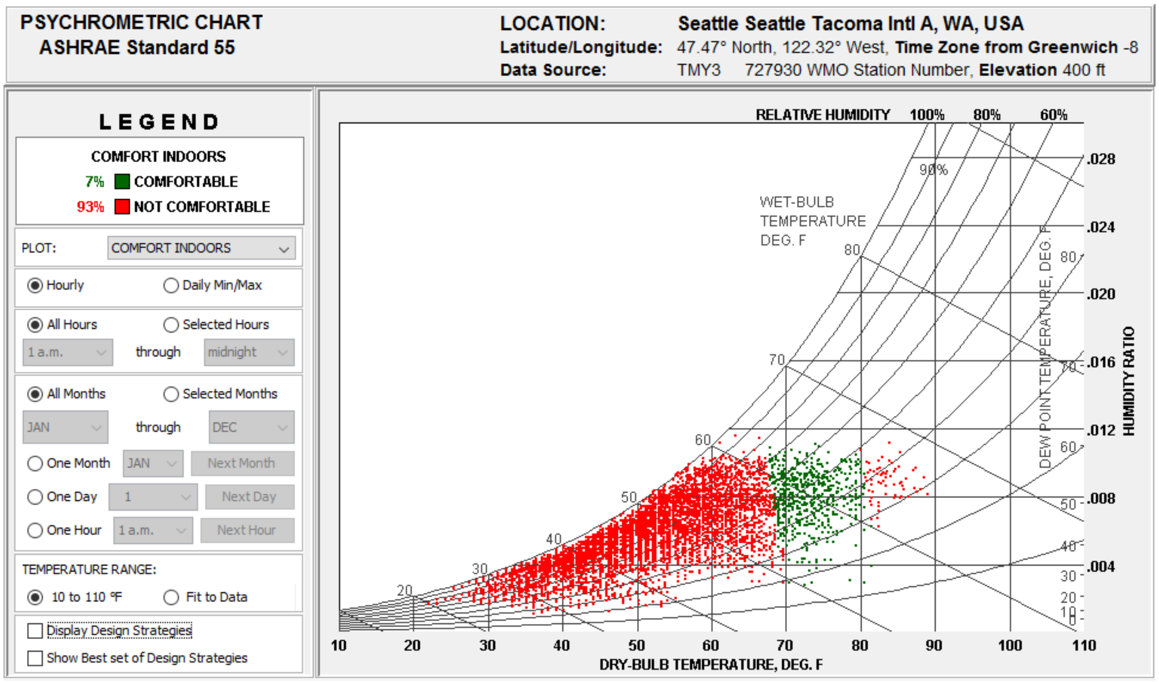


Figure 3.3-17 Seattle Hourly Diffuse Solar Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-18, very few points are on the right side of the green points area, which means there is only a little period during a year needs cooling and there are more hours during a year needs heating instead. During winter, if the air is heated the relative humidity would decrease. However, one should pay attention when applying the floor radiation system in Seattle, because if cold outdoor air penetrates inside, the warm indoor air will be cooled by the cold outdoor air and will hit the dew point, which may cause condensation on the inside of windows.



PSYCHROMETRIC CHART
ASHRAE Standard 55

LOCATION: **Seattle Seattle Tacoma Intl A, WA, USA**
Latitude/Longitude: 47.47° North, 122.32° West, **Time Zone from Greenwich** -8
Data Source: TMY3 727930 WMO Station Number, **Elevation** 400 ft

LEGEND

COMFORT INDOORS
7% 🟩 COMFORTABLE
93% 🟥 NOT COMFORTABLE

PLOT: COMFORT INDOORS

● Hourly ○ Daily Min/Max

● All Hours ○ Selected Hours
1 a.m. through midnight

● All Months ○ Selected Months
JAN through DEC

○ One Month JAN Next Month
○ One Day 1 Next Day
○ One Hour 1 a.m. Next Hour

TEMPERATURE RANGE:
● 10 to 110 °F ○ Fit to Data

☐ Display Design Strategies
☐ Show Best set of Design Strategies

Figure 3.3-18 Psychrometric Chart of Seattle (Climate Consultant, Version 6.0)

3.3.4 Spokane-5B

Spokane is located at 47.49° N, 117.59° W, in the eastern Washington State, of northwestern United States.

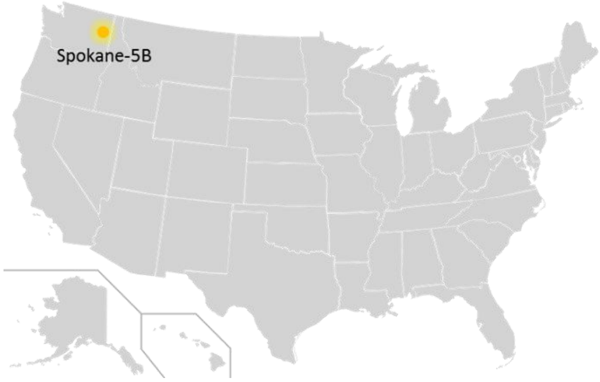


Figure 3.3-19 Spokane

According to Figure 3.3-20, the dry bulb temperature of Spokane is almost similar to Seattle, but it has more severe temperature fluctuations over a year. The temperature is lower during winter and higher during summer than Seattle. The average monthly temperature is around 27°F to 69°F, the annual lowest temperature is 2°F in January, and the highest temperature is 97°F in July. Daily temperature ranges are smaller during winter and larger during summer in Spokane.

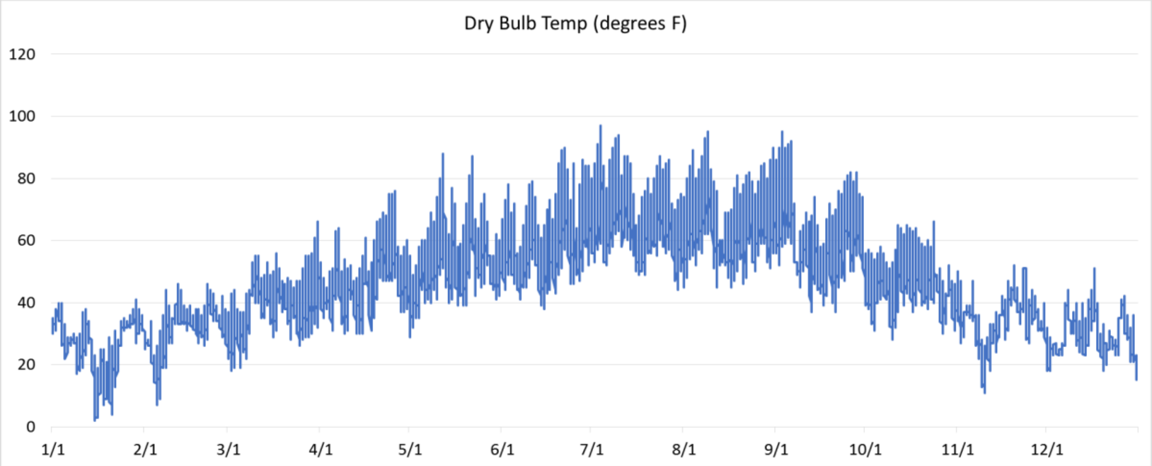


Figure 3.3-20 Spokane Hourly Dry Bulb Temperature (Climate Consultant, Version 6.0)

Spokane has its largest precipitation in December, and the relative humidity is high between October and February. However, the humidity is extremely low between March and September. According to Figure 3.3-21, the average monthly relative humidity is around 46% and 88%.

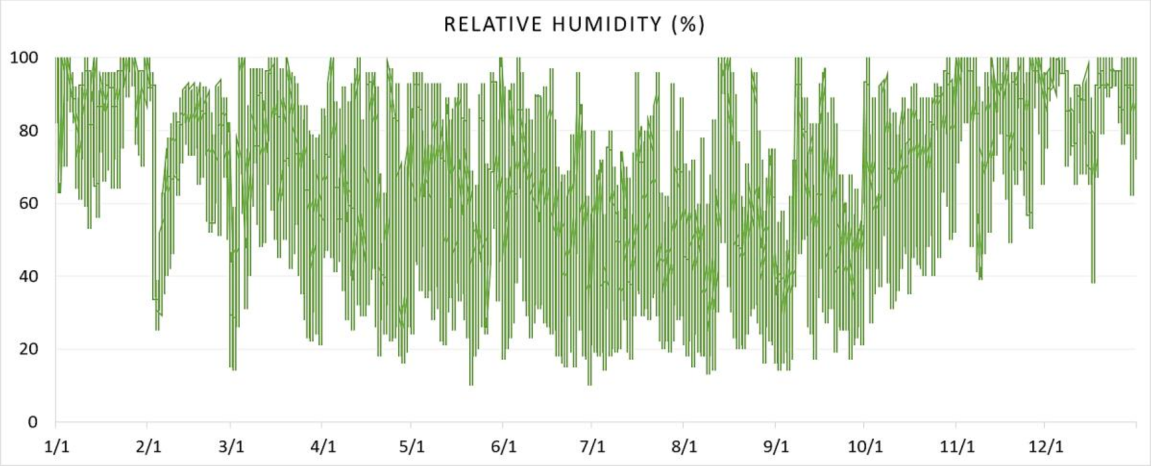


Figure 3.3-21 Spokane Hourly Relative Humidity (Climate Consultant, Version 6.0)

Figure 3.3-22 shows the annual direct solar radiation of Spokane. Similar to Seattle, as the high latitude and sky cover range in December, the direct solar radiation is low during winter in Spokane. The average monthly direct solar radiation is between 50 Btu/SF and 168Btu/SF. The annually highest direct solar radiation in Spokane is 318Btu/SF appears in March.

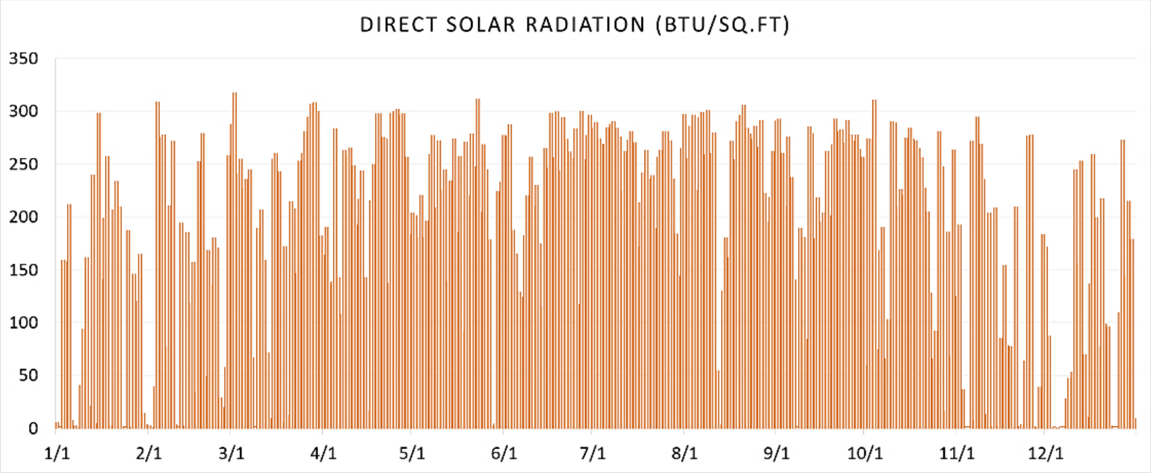


Figure 3.3-22 Spokane Hourly Direct Solar Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-23, the diffuse solar radiation of Spokane is relatively low. The monthly average diffuse solar radiation is between 26Btu/SF and 52Btu/SF.

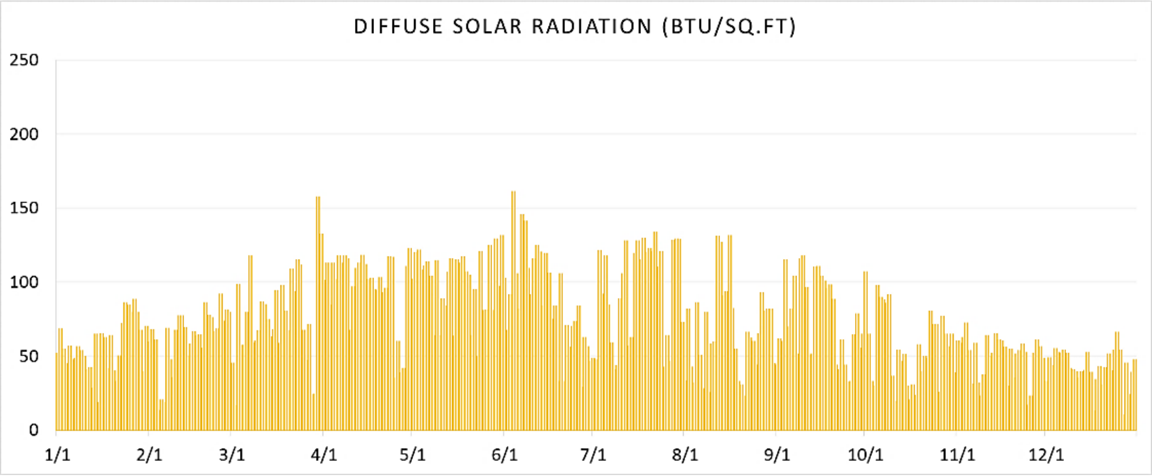
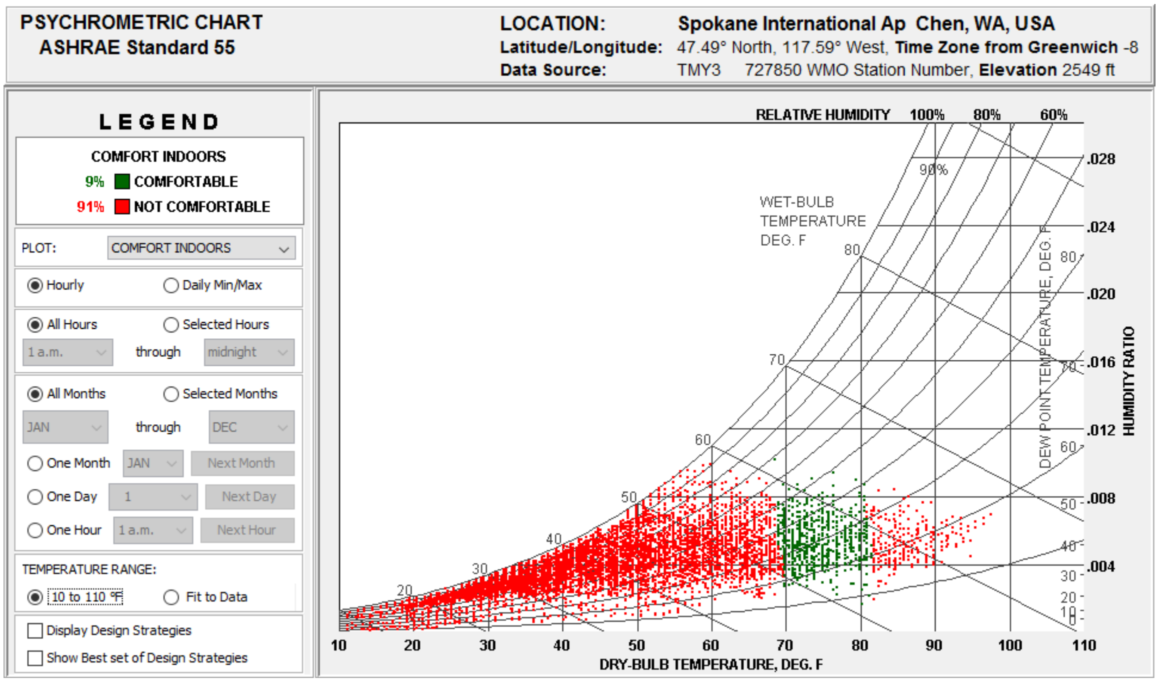


Figure 3.3-23 Spokane Hourly Solar Diffuse Radiation (Climate Consultant, Version 6.0)

According to the Figure 3.3-24, 9% of the hours during a year is located in the comfort zone, and most of the time during a year needs heating instead of cooling, which is similar to Seattle. The humidity during summer is low, so there will not be too much energy required for dehumidification. However, special attentions should be paid when using radiant heating systems in Spokane.


PSYCHROMETRIC CHART
ASHRAE Standard 55

LOCATION: **Spokane International Ap Chen, WA, USA**
Latitude/Longitude: 47.49° North, 117.59° West, **Time Zone from Greenwich** -8
Data Source: TMY3 727850 WMO Station Number, **Elevation** 2549 ft

LEGEND
COMFORT INDOORS
9% 🟩 COMFORTABLE
91% 🟥 NOT COMFORTABLE

PLOT: COMFORT INDOORS
● Hourly ○ Daily Min/Max
● All Hours ○ Selected Hours
1 a.m. through midnight
● All Months ○ Selected Months
JAN through DEC
○ One Month JAN Next Month
○ One Day 1 Next Day
○ One Hour 1 a.m. Next Hour
TEMPERATURE RANGE:
● 10 to 110 °F ○ Fit to Data
☐ Display Design Strategies
☐ Show Best set of Design Strategies

Figure 3.3-24 Psychrometric Chart of Spokane (Climate Consultant, Version 6.0)

CHAPTER 4. HVAC SYSTEMS

4.1 MAJOR ELEMENTS

The major elements of an HVAC system include central plant, distribution system, and delivery system, as Figure 4.1-1 shows.

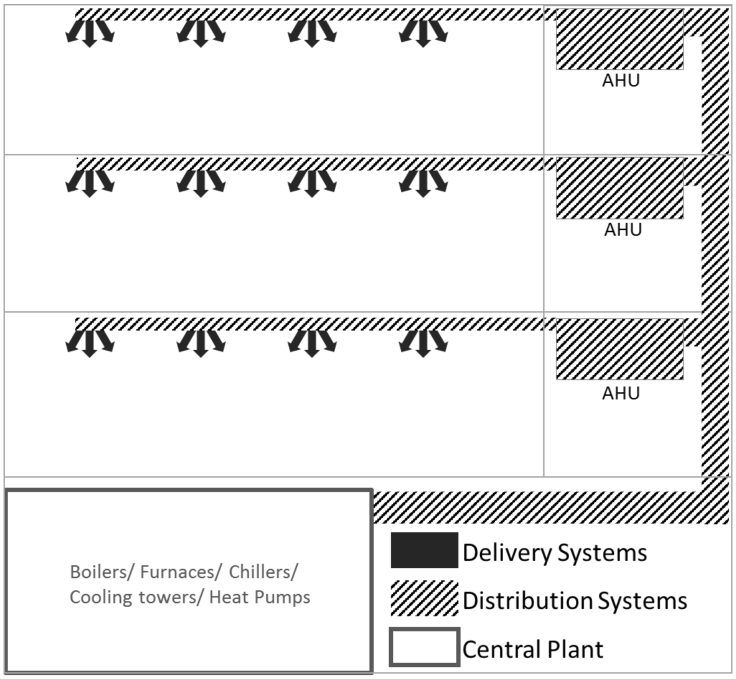


Figure 4.1-1 Major HVAC Elements

The Central plant, also called heat source, is the equipment to generate heating or cooling, such as boilers, furnaces, heat pumps, chillers, cooling towers; this is where most of the electricity or fuel consumption occurs depending on the system. The equipment must be sized to meet the peak cooling load or/ and heating load of the building.

Distribution systems are the equipment that moves the heat conveying medium, water or air, from the central plant to space where heating or cooling is needed. Water pumps and piping systems are used to convey hot water, steam, chilled water, and cooling water. Fans, air-handling units, and ductwork can be utilized for hot and cool air.

The delivery system is devices that deliver heat, cooling, and ventilation to space. There is a variety of different delivery systems available in the market, such as Variable Air Volume (VAV) system, fan coil system, chilled beam system, and floor radiant system.

4.2 VARIABLE AIR VOLUME (VAV) REHEAT SYSTEM

Variable Air Volume (VAV) system is very popular in the commercial sector of the construction industry. VAV system is a type of all air system, which provides complete cooling or heating capacity with air.

Unlike the traditional Constant Air Volume (CAV) System, VAV system satisfies the cooling or heating load of the space by changing its airflow rate while maintaining a constant temperature of the air. (MGEC, 2015)

A reheat system is used when cooling air to a temperature low enough to condense or remove moisture and reheat the air to the desired temperature. The Figure 4.2-1 shows an example of cooling and reheat process in the psychrometric chart.

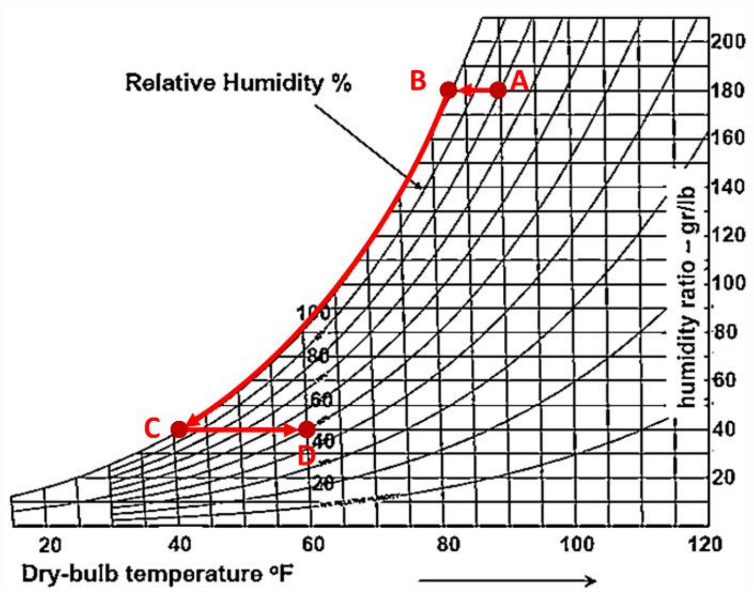


Figure 4.2-1 Psychrometric Chart of Cooling and Reheat Process

According to the figure 4.2-1, the outside air starts at point A, where the dry bulb temperature is 90°F and the relative humidity is 80%. Then the air is cooled to the dew point B (83°F, 100%). When the air is further cooled, the water content condenses out, effectively dehumidifying the air until it reaches point C (40°F, 100%). Finally, the mixture passes the reheat coil that heats the temperature of the air to point D (60°F, 50%), the set point of the thermostat controller.

Figure 4.2-2 is a diagram of a typical VAV reheat system.

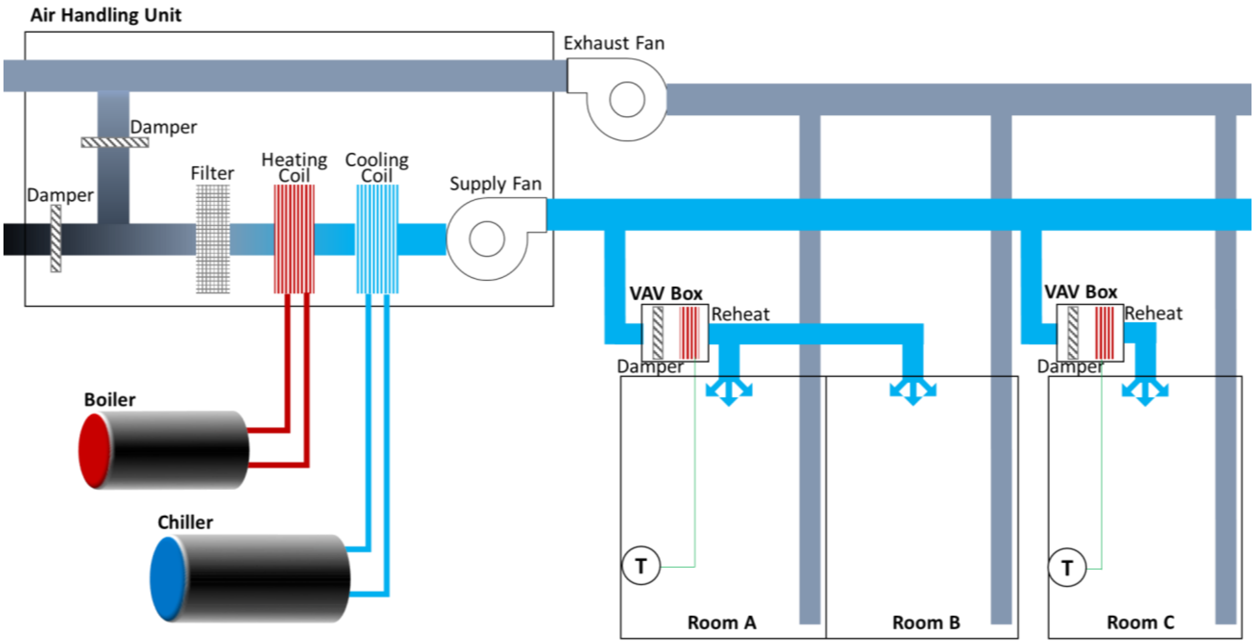


Figure 4.2-2 Typical VAV Reheat System

In the Air Handling Unit (AHU), there is the heating coil and the cooling coil. The heating coil is only used in winter. Before the supply air enters into the air conditioning space, it goes through a VAV box, where the mixture passes the reheating coil that heats the temperature to the set point, which is controlled by the indoor space thermometer.

VAV Reheat system is the most common HVAC system for office buildings in the U.S. However, due to the reheating process, VAV reheat system requires an extra amount of energy than the actual cooling load so that it is not considered as an energy efficient system.

4.3 CHILLED BEAM SYSTEM

Chilled Beam is an air delivery system originated from Europe. (Murphy, 2011) There are two kinds of chilled beam system: one is Active Chilled Beam with Dedicated Outdoor Air System (DOAS); the other is Passive Chilled Beam System with Separate Air Ventilation.

Active Chilled Beam with Dedicated Outdoor Air System (DOAS) is an air-water HVAC system consists of cooling or heating water coil with cold or hot water running through it and a 100% filtered and precooled (or preheated) outside air supply. The water coils remove the 60%-80% of the sensible heat, (Loudermilk, 2016) and the supply outdoor air can handle the other part of space sensible heat and latent heat, and provide ventilation.

As the water coil can take the sensible heat of the space, chilled beam system can operate with smaller ductworks than VAV system, in which both the sensible heat and the latent heat are handled entirely by air.

Also, water is more efficient in conveying heat than air. Water has a 62.4lb/ft3 density versus air 0.0765lb/ft3, and the heat capacity of water is 4.18kJ/K·kg versus air 1.012kJ/K·kg. The quantity of heat **Q** is defined as,

$$\boldsymbol{Q = Cm\Delta t = C\rho V\Delta t} \qquad \text{Equation 4.3.1}$$

, where **Q** stands for the quantity of heat transferred, **C** is the specific heat capacity, **m** is the mass, and ΔT is the resulting temperature change.

According to the equation, water can hold about 3370 times as much heat as the same volume of air. With cooling or heating water coil, chilled beam system can operate with less air volume than the all-air systems.

Figure 4.3-1 shows how an active chilled beam works.

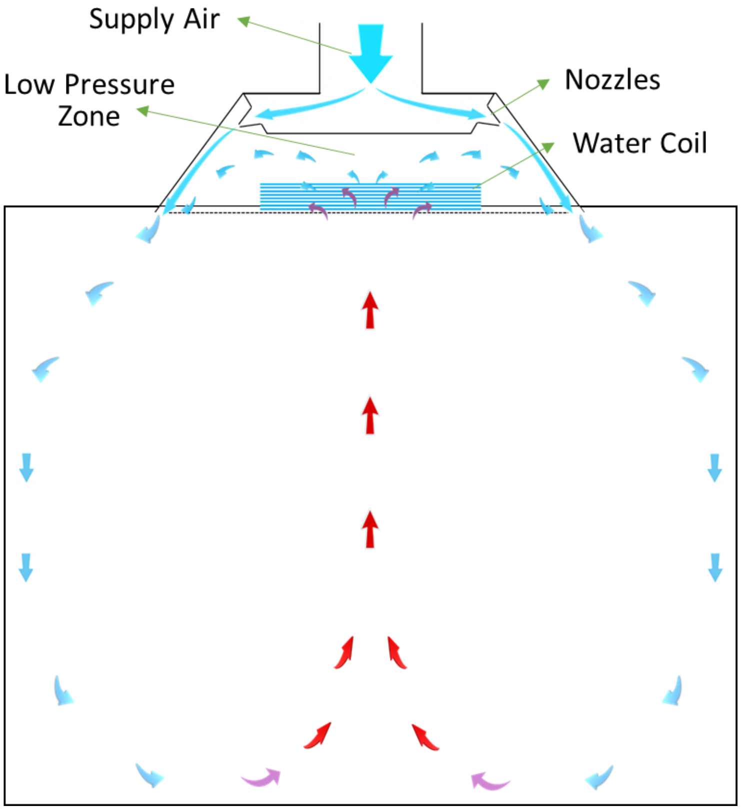


Figure 4.3-1 Active Chilled Beam and the Air Flow

The supply air comes into the top and goes into the nozzles and out to the space, which creates a low-pressure zone around the water coils. (Murphy, 2011) This low-pressure zone allows the chilled beam to operate even under a heating condition. The low-pressure zone induces the room air up into the chilled beam, goes over the water coil, and is either cooled or heated. Then the cooled air or the heated air will mix with the outdoor supply air and enter into room space.

The induction ratio is the ratio of the induced air volume to the outdoor supply air volume, which is critical to a chilled beam. The induction ratio of a typical chilled beam is at least 1:3 to

1:4, (Setty, 2011, p.3) which means that space gets about 3 to 4 times more air volume than the primary outdoor supply air volume in total to handle the cooling load with chilled beam system. It can reduce the size of the supply air ducts. (Loudermilk, 2009)

A typical chilled beam system requires 60%-80% less primary airflow. (Loudermilk, 2016) The less supply air requirement results in smaller ductworks and smaller air handling units. The reduction of the air ducts allows chilled beam system to operate with less energy (fan power) and at the same time reduces the ceiling capacity for the ductworks, which saves the structure cost of the building.

Figure 4.3-2 is a diagram of a typical active chilled beam system.

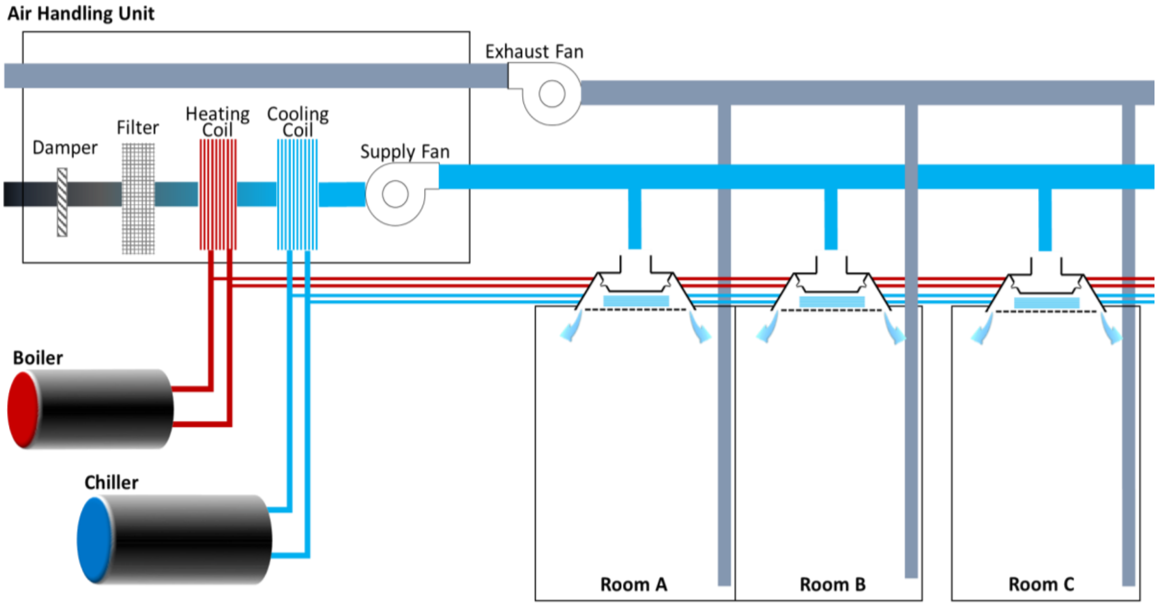


Figure 4.3-2 Typical Active Chilled Beam System

The air handling units of chilled beam system are similar to the VAV reheat system, except the returning air is not mixed with the outdoor air. In a chilled beam system, dampers can only be found in the air handling unit, but there is not any inside the chilled beam, so chilled beam system requires less maintenance than the VAV reheat system does. The ductworks and the supply and exhaust fans are much smaller in the chilled beam system.

The chilled water and hot water from the chiller and boiler go to both the air handling units and the chilled beams, and the outdoor air would be precooled or preheated by the cooling or heating coils before entering into those chilled beams. The chilled water for chilled beam systems requires being warmer than the chilled water in the VAV system. If the chilled water temperature is lower than the dew point temperature when it is running through the chilled beam cooling coil, it might cause condensation issues.

Figure 4.3-3 shows a diagram of a passive chilled beam system.

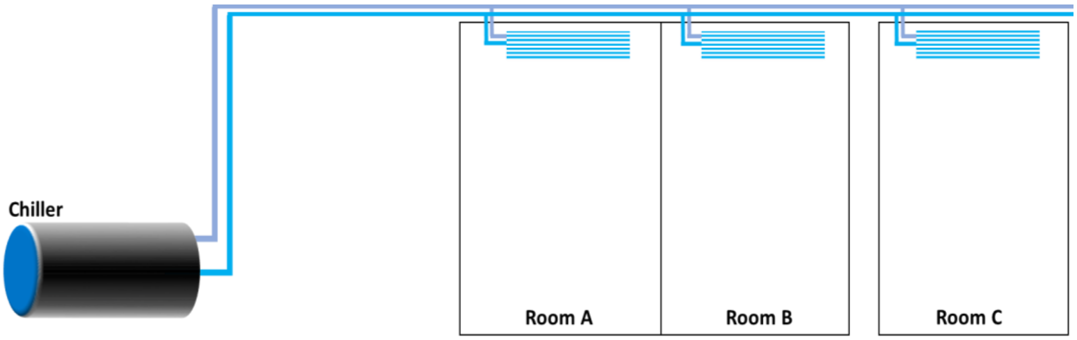


Figure 4.3-3 Passive Chilled Beam System

Passive chilled beam system is an all-water radiation system consist of water coils with chilled water running inside. As there is no air supply in a passive chilled beam system, the occupant space should have another separate ventilation system or operable windows. When the room temperature is high, hot air rises and pass over the cooling water coil. Then the air will be cooled and sink to the lower level of the space.

As passive chilled beam system utilizes the concept of hot air rises and cold air sinks, it is not effective when dealing with the heating load. Passive chilled beam systems are mostly used in projects when there is only cooling needed.

In the energy modeling process, simulating a chilled beam system will be a challenge, (Rumsey et al, 2009) since eQUEST does not have the chilled beam component. (Vaughn, 2012)

4.4 AIR-SOURCE HEAT PUMP (ASHP) SYSTEM

Pumps are used to move fluids with mechanical action, mostly from a low elevation to a higher elevation. Similarly, heat pump is a device that moves heat from a low-grade heat source (like air, water, soil, solar, and industrial waste heat) to a high-grade heat medium.

There are four major components in a heat pump system: evaporator, compressor, condenser, and expansion valve. A refrigerant circulates through these four components and experiences the evaporation, compression, condensation and expansion process, which is the way heat pumps move the heat from a lower temperature to a higher temperature.

A refrigerant used in an air-source heat pump is a substance with a low saturation temperature so that it can evaporate at a room temperature when the pressure is low and can condense in an outdoor warmer temperature when the pressure is high. Figure 4.4-1 (ASHRAE, 2005) shows the refrigeration cycle (heat pump cooling cycle) on the pressure-enthalpy diagram for R-134a.

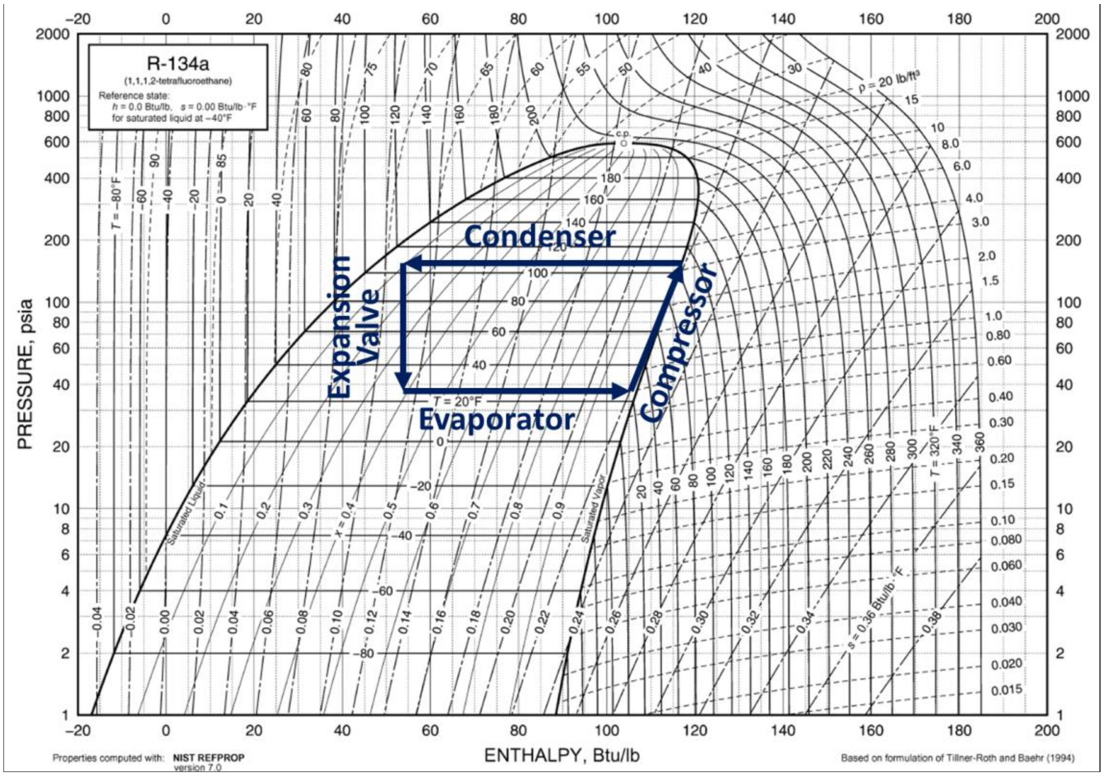


Figure 4.4-1 Cooling Cycle on the R-134a Pressure-Enthalpy Diagram (ASHRAE, 2005)

The most common type of air-source heat pumps is air to air heat pump, which extracts heat from the indoor air and then transfers heat to outdoor in a summer cooling cycle. In a winter heating cycle, the air-source heat pump extracts heat from the outdoor air and then moves the heat to the indoor space.

Figure 4.4-2 shows a typical cooling cycle of the air-source heat pump.

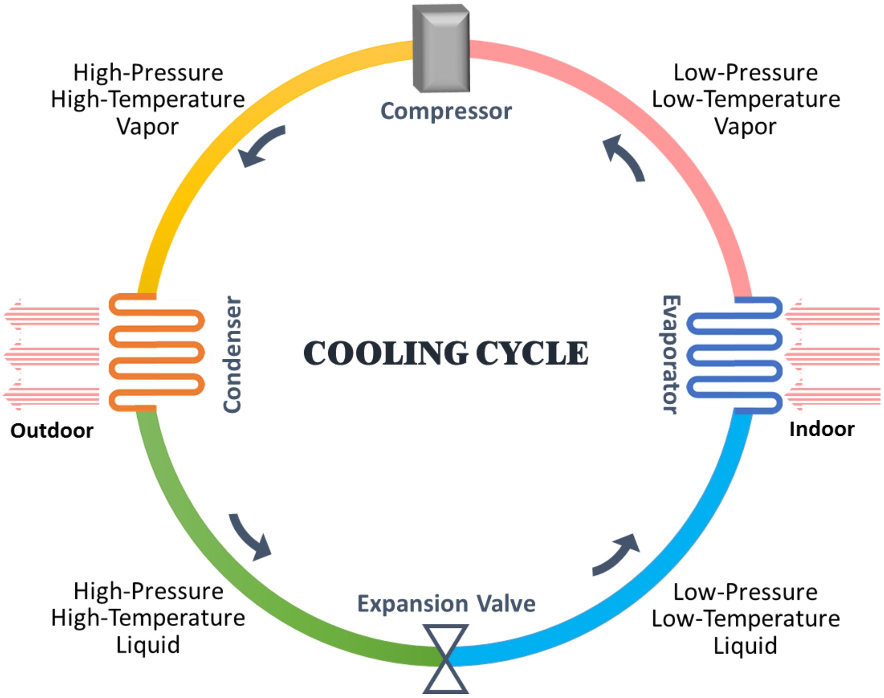


Figure 4.4-2 Basic Heat Pump Cycle (Cooling Cycle)

In the cooling cycle, the low-pressure low-temperature liquid refrigerant absorbs heat from the evaporator (indoor air conditioning unit) vaporized into low-pressure low-temperature Vapor. Then the refrigerant is compressed into high-pressure and high-temperature vapor through the compressor. After the compression process, the refrigerant releases the heat it absorbed earlier in the evaporation process through a condenser (outdoor air conditioning unit) and condensates into a liquid with high-pressure and high-temperature. This liquid will go through an expansion valve and become low-pressure low-temperature liquid again. (Natural Resources Canada, 2004)

In winter heating season, the heat pump cycle can be reversed. In the heating cycle, the refrigerant flows in the opposite direction, the indoor coil works as the condenser, and the outdoor coil works as the evaporator, moving the outdoor heat to the indoor space during the winter season, (Natural Resources Canada, 2004) as Figure 4.4-2 shows.

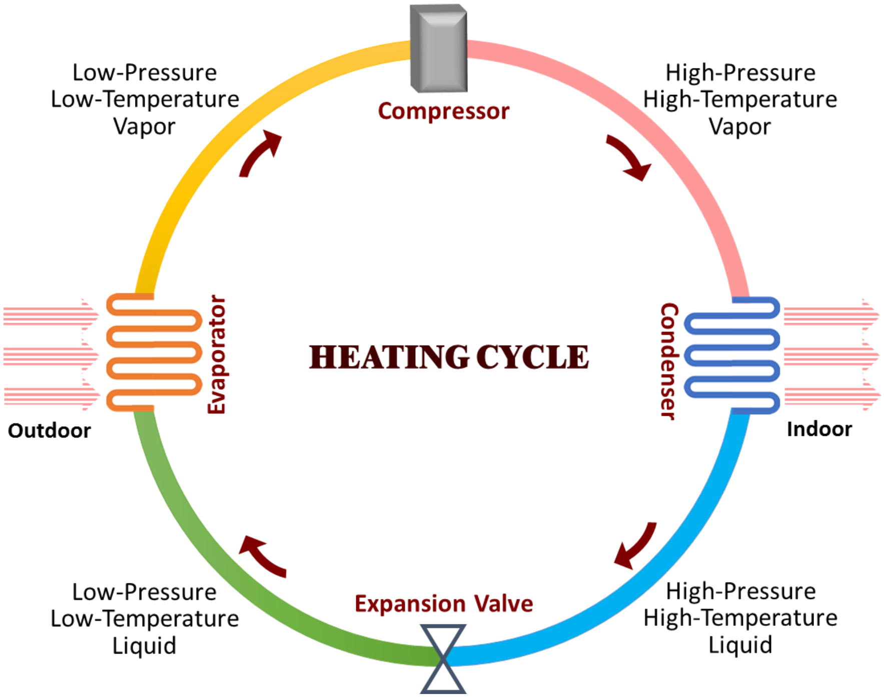


Figure 4.4-3 Heating Cycle of a Typical Heat Pump

In the heating cycle, the high-pressure high-temperature liquid refrigerant runs through an expansion valve and becomes a low-pressure low-temperature liquid. Then it absorbs heat from the outdoor air through the evaporator (outdoor air conditioning unit) and boils to vapor. After that, the refrigerant vapor is compressed into a smaller volume, with a higher pressure and higher temperature in the compressor device. And then, the refrigerant releases the heat, which it absorbed earlier in the evaporation process, through a condenser (indoor air conditioning unit) to the indoor air and condensates into liquid again. (Natural Resources Canada, 2004)

Figure 4.4-4 (ASHRAE, 2005) shows the heat pump heating cycle on the pressure-enthalpy diagram of R-134a.

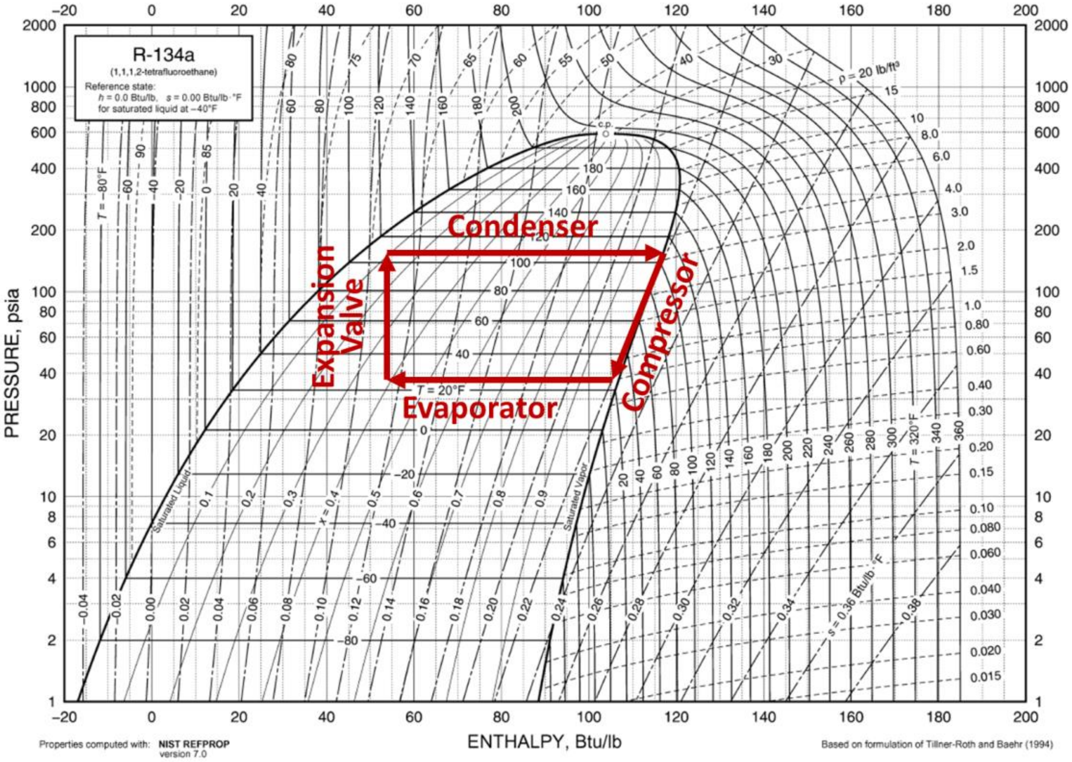


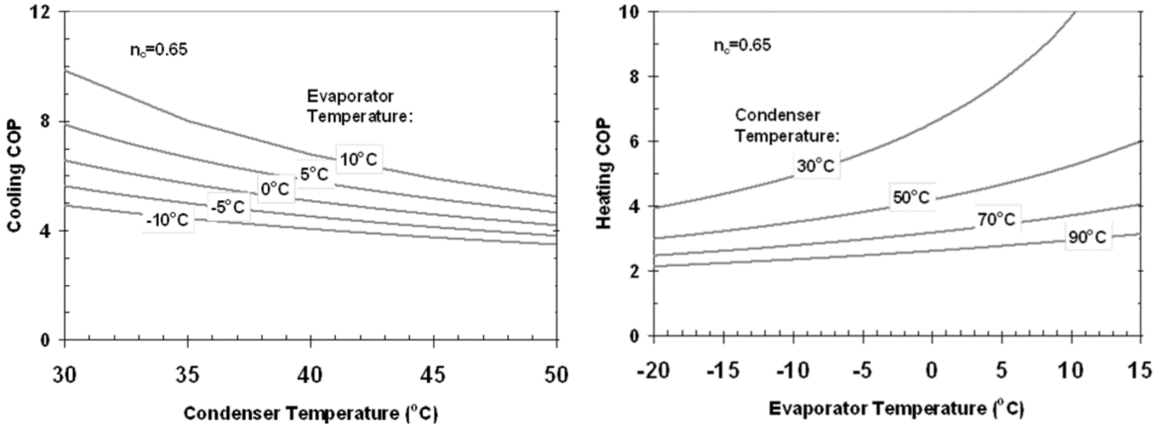
Figure 4.4-4 Heating Cycle on the R-134a Pressure-Enthalpy Diagram (ASHRAE, 2005)

In the evaporation process of a heating cycle, the temperature of the refrigerant inside the coil should be lower than the outdoor air temperature so that the refrigerant can absorb the heat from the outdoor air. When the cold and humid outdoor air passes over the outside coil, the refrigerant inside the coil absorbs the heat from the outdoor air and cools the outdoor air. If the air temperature gets cooler than its due point temperature, the air will condense. When the temperature is below freezing point, the condensed water will freeze on the outside coil.

If there are too much frost on the coil, the refrigerant can not absorb heat from the outdoor air, which reduces the efficiency of the evaporation process. At this time, the heat pump should be switched into the cooling cycle with the outdoor fan shut off, which is the defrost mode. (Natural Resources Canada, 2004) The amount of frost on the coil depends on the temperature and the humidity of the outdoor air, so the frost issue happens more often in the cold and humid climates.

The Coefficient of Performance (COP) is used to measure the efficiency of a heat pump in steady state ratings obtained at one set of temperature conditions, which is defined as the ratio of the energy output to the electrical energy consumption of the heat pump. The COP depends on the outdoor temperature and the indoor setpoint temperature.

The COP of an air-source heat pump in the severe conditions, where the climate is too hot or too cold, is affected and decreased. Either to move heat to a higher temperature or extract heat from the cooler air is difficult. Figure 4.4-5 (Harvey, 2009) shows an example of the temperature-COP curves of a heat pump cooling cycle and heating.



a) b)

Figure 4.4-5 Variation in the COP of a Heat Pump (Harvey, 2009)

Figure 4.4-5 a) is a temperature-COP diagram of a heat pump cooling cycle, where the x-axis is the temperature of the condensation process (outdoor temperature), the y-axis is the cooling COP, and the curves shows the COP-T relationship with different temperatures of the indoor coil refrigerant. According to the Figure 4.4-5 a), when the outdoor temperature is higher than 40℃ (104°F), the cooling COP will get below 6.

Figure 4.4-5 b) is a temperature-COP diagram of a heat pump heating cycle, where the x-axis indicates the evaporator temperature (outdoor temperature), the y-axis is the heating COP, and the curves shows the COP-T relationship with different temperatures of the indoor coil refrigerant.

According to the Figure 4.4-5 b), when the outdoor temperature is lower than 0℃ (32°F), the cooling COP will get below 4.

4.5 GROUND-SOURCE HEAT PUMP (GRHP) SYSTEM

Ground temperature is relatively constant compared to the air temperature. If the outside coils are buried underground, they are not affected by the severe outdoor air temperatures. (Karr, 2011)

Figure 4.5-1 shows the monthly average ground temperature of Miami, Phoenix, Seattle and Spokane by different depth from the top of the soil. The annual highest and lowest dry-bulb temperatures of the four cities are marked on the figure.

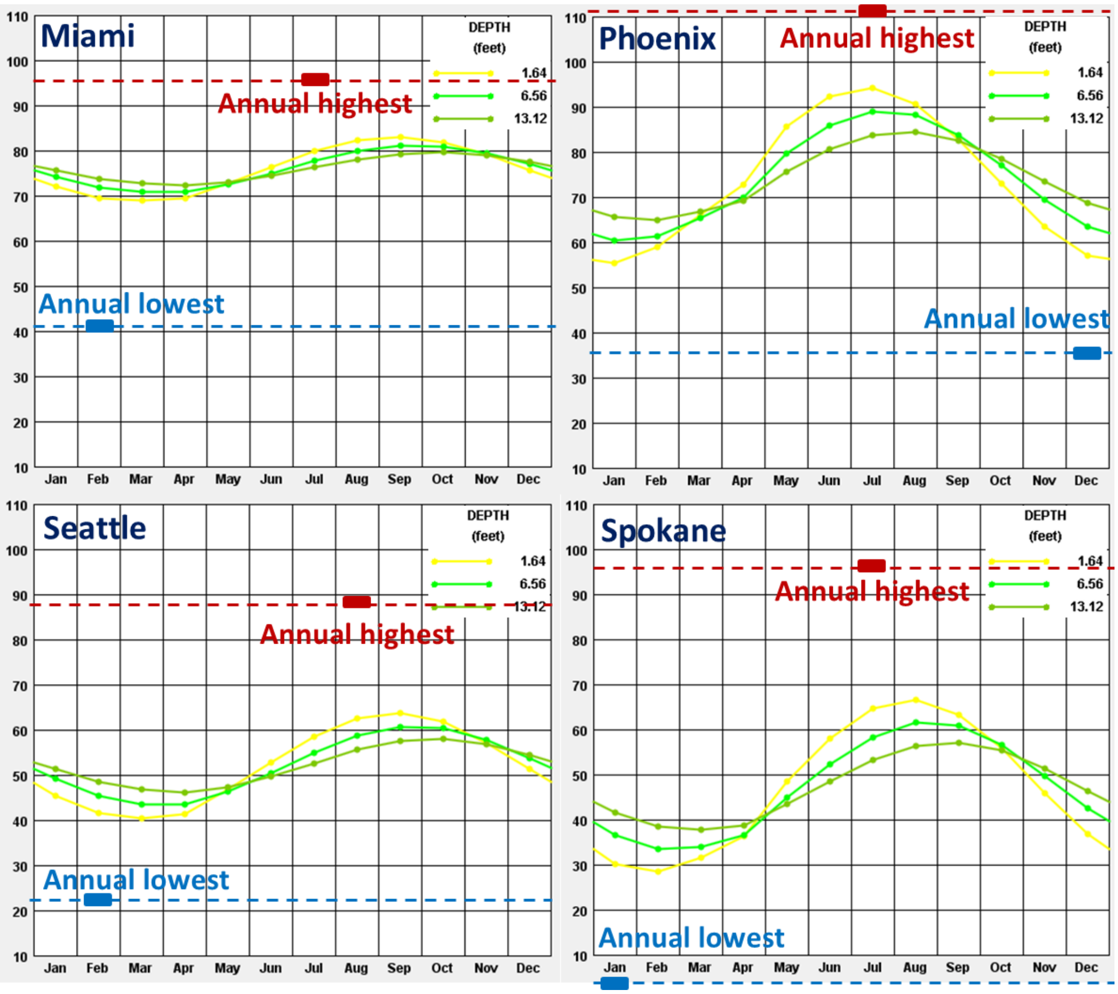


Figure 4.5-1 Monthly Average Ground Temperature (Climate Consultant, Version 6.0)

The maximize and the minimum dry-bulb temperatures and the maximize and the minimum 13.12-foot-deep ground temperatures are compared in the Table 4.5-1.

Table 4.5-1 Air and Ground Temperatures (Climate Consultant, Version 6.0)

	Miami	Phoenix	Seattle	Spokane
Max Dry-Bulb	96°F	112°F	89°F	97°F
Min Dry-Bulb	41°F	36°F	22°F	2°F
Max Ground (13.12ft)	80°F	84°F	58°F	58°F
Min Ground (13.12ft)	72°F	64°F	47°F	39°F

The ground temperature has a less fluctuation around a year than the dry-bulb temperature. Ground-Source Heat Pump System can effectively transfer the heat stored in the soil to the indoor space in the winter, and extract heat from the indoor space and release the heat underground in the summer.

Another factor, which will also influence the performance of GSHP system, is the ground-loop heat exchanger’s loading history. If the system is rejecting too much heat than it is extracting from the soil, there will be an imbalance, which will decrease the capacity of the ground heat exchanger in GSHP system. (Wang, 2015) However, in this research, the imbalance issue is not addressed in the eQUEST energy modeling process.

Figure 4.5-2 shows the components of a ground-source heat pump system (cooling cycle).

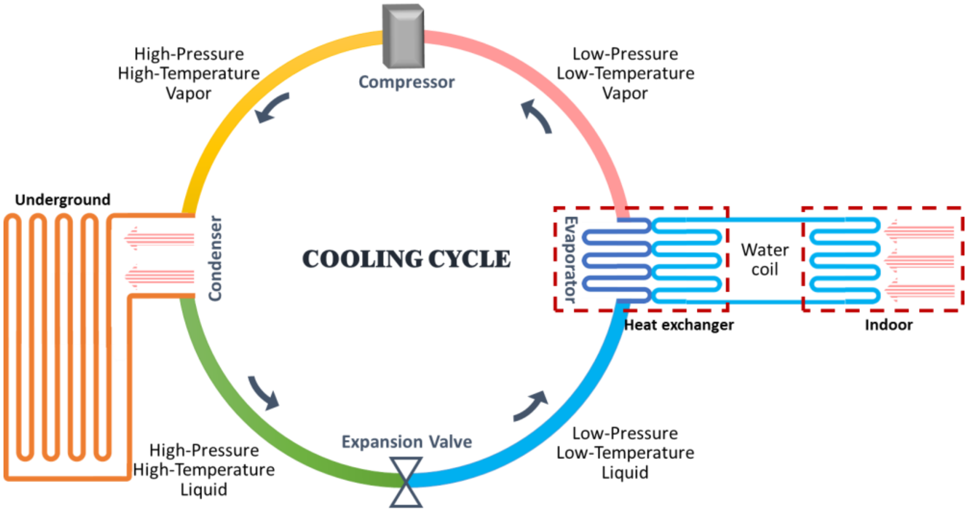


Figure 4.5-2 Cooling Cycle of a Ground-Source Heat Pump

One major difference between a ground-source heat pump system and an air-source heat pump is the outside coil. In a ground-source heat pump system, the outside coil is buried underground so that the refrigerant can extract heat from and release heat to the underground soil.

As ground-source heat pump system has a higher initial cost for the additional underground pipes, this system is most widely used for medium to large size of projects. There is indoor water coil running through each space within the building and absorbing heat from the spaces. With ground-source heat pump system, it is efficient in Moving heat from the underground to the desired rooms. In the evaporation process of the cooling cycle, the refrigerant extracts heat from the indoor cooling water coil, rather than the indoor air in an air-source heat pump system. (Natural Resources Canada, 2004)

A ground-source heat pump can also work in a heating mode. The refrigerant medium flows conversely and moves heat from underground soil to the indoor space. Figure 4.5-3 shows the heating cycle of a ground-source heat pump.

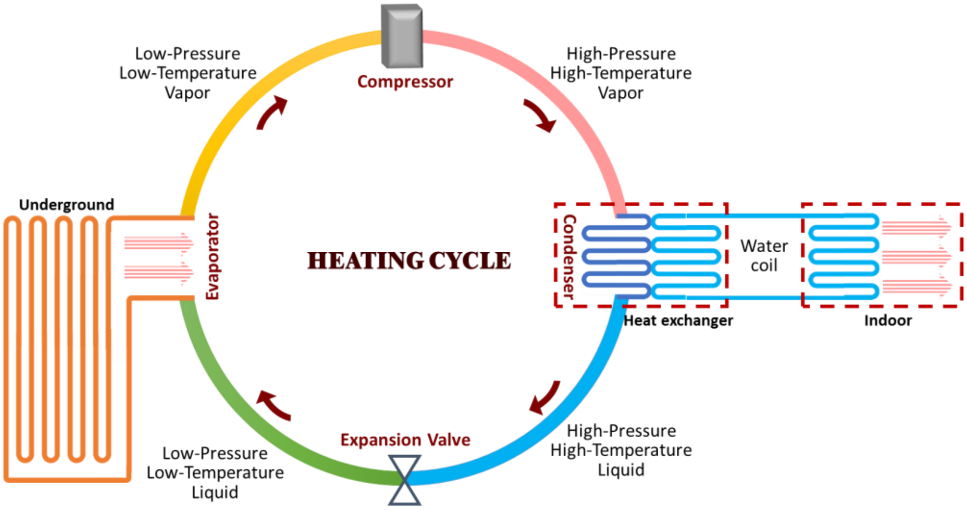


Figure 4.5-3 Heating Cycle of a Ground-Source Heat Pump

CHAPTER 5. ENERGY MODEL

5.1 PHYSICAL MODEL

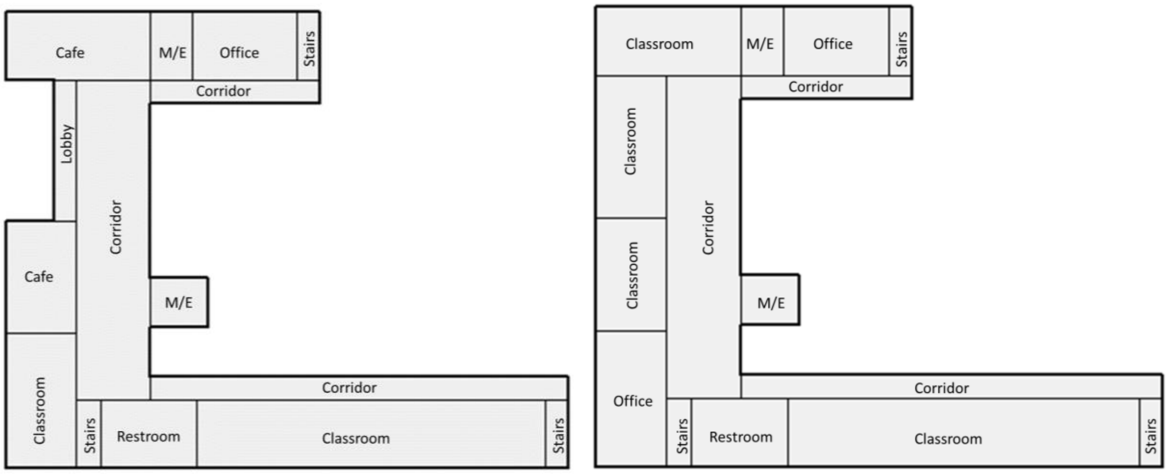
The sixteen energy models in this thesis are developed using eQUEST, a graphical-user interface that runs on DOE-2.2 (eQUEST, 2008). The building selected for the analysis is a three-story educational office building simplified based on an embedded sample project from the Revit software. Table 5.1-1 shows the floor areas of different functions of space within the building on each floor.

Table 5.1-1 Floor areas

	Level 1	Level 2	Level 3	Total	% of total
Classroom	4981 ft^2	7648 ft^2	7648 ft^2	20276 ft^2	37.8%
Office	1047 ft^2	2467 ft^2	2467 ft^2	5980 ft^2	11.1%
Café	2618 ft^2	-	-	2618 ft^2	4.9%
Lobby	448 ft^2	-	-	448 ft^2	0.8%
Corridor & Stairs	6265 ft^2	6265 ft^2	6265 ft^2	18795 ft^2	35.0%
Mechanical and Electrical	873 ft^2	873 ft^2	873 ft^2	2618 ft^2	4.9%
Bathroom	983 ft^2	983 ft^2	983 ft^2	2949 ft^2	5.5%
Total	17215 ft^2	18235 ft^2	18235 ft^2	53684 ft^2	-

The total floor area of the sample building is 53,684 square feet. The first floor is 17,215 square feet and includes classrooms, an office, cafés, mechanical and electrical rooms, and a lobby facing west. The second floor is 18,235 square feet and includes classrooms, offices, mechanical and electrical rooms. The floor area of the third floor is 18,235 square feet and shares the same floor plan with the second floor in the energy modeling process. The major room type of this building is classroom with a floor area of 20,276 square feet, which accounts for 37.8% of the building total floor area.

The Figure 5.1-1 (a) is the detailed floor plan of level 1. The Figure 5.1-1 (b) shows the floor plan for level 2 and level 3.



(a) (b)

Figure 5.1-1 Floor Plan

The energy model is build based on the floor plan with eQUEST energy modeling software. The Figure 5.1-2 and the Figure 5.1-3 show the 3D Views of the eQUEST energy model.

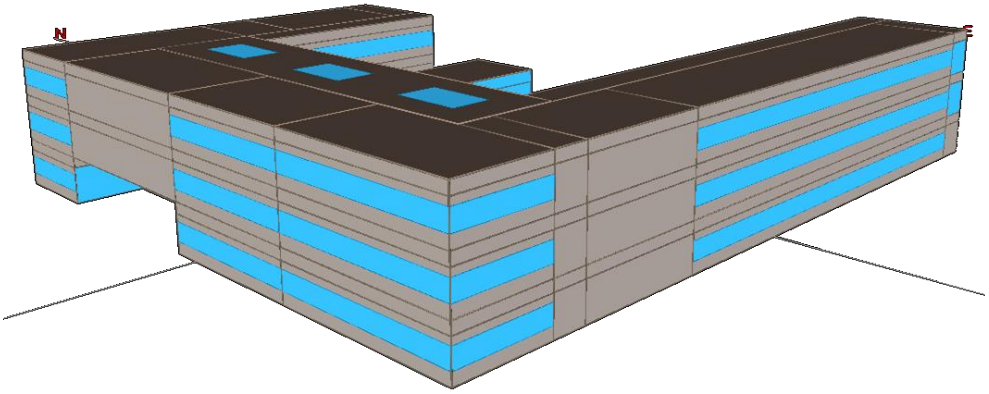


Figure 5.1-2 eQUEST Model 3D View (South and West Facade)

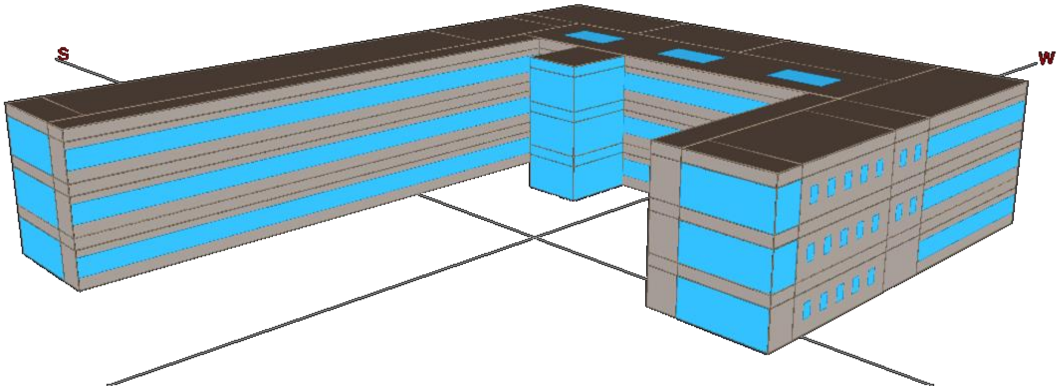


Figure 5.1-3 eQUEST Model 3D View (East and West Facade)

It is a metal frame building with the frame intervals larger than 24 inches. The insulation level of the roof and the wall varies in different climate zones. The ground floor concrete is 12-inch thick with a full under slab insulation of R=10, as Table 5.1-2.

Table 5.1-2 Envelope Material

	Structure	Exterior Finish	Insulation
Roof	Metal frame >24 in. o. c.	Built-up roof	Exterior insulation depends on the climate zone
Walls	Metal frame >24 in. o. c.	Stucco/gunite	Exterior insulation or/ and additional insulation depends on the climate zone
Ground Floor	12 in. Concrete	Earth contact	Full under slab insulation, R=10

5.2 ENVELOPE PERFORMANCE

As described in Chapter 2, ASHRAE 90.1-2010 gives a baseline of each envelope element in different climate zones, which can be used as a guideline in material selection so that the envelope can provide a sound insulation performance.

The Table 5.2-1 shows the baseline requirements from ASHRAE 90.1-2010 for the climate zone 1A (Miami), climate zone 2B (Phoenix), climate zone 4C (Seattle), and climate zone 5B (Spokane).

Table 5.2-1 ASHRAE Building Envelope Requirements (ASHRAE, 2010)

		1A (Miami)	2B (Phoenix)	4C (Seattle)	5B (Spokane)
Opaque Envelope	Roof	U-0.063 R-15.0 ci.	U-0.048 R-20.0 ci.	U-0.048 R-20.0 ci.	U-0.048 R-20.0 ci.
	Wall	U-0.124 R-13.0	U-0.124 R-13.0	U-0.064 R-13.0+R-7.5 ci.	U-0.064 R-13.0+R-7.5 ci.
	Floor	U-0.322 R (Not Required)	U-0.107 R-6.3 ci.	U-0.087 R-8.3 ci.	U-0.074 R-10.4 ci.
Transparent Envelope	Window	U-1.2 SHGC-0.25	U-0.75 SHGC-0.25	U-0.4 SHGC-0.40	U-0.35 SHGC-0.40
	Skylight	U-1.36 SHGC-0.19	U-1.36 SHGC-0.19	U-0.69 SHGC-0.39	U-0.69 SHGC-0.39

The R-value and the U-value measure the insulation performance of building envelope materials. R-value is a measure of a material's resistance to heat flow from the warmer side to the cooler side, which only considers the resistance to heat flow via conduction. A material with higher R-value has a higher capacity to reduce the flow of heat. (Brenden, 2010) R-value describes the insulation performance of materials with a certain thickness. A high R-value there is little impact of heat flow via radiation, such as exterior walls, roofs, and ground floors.

U-value is a measure of a material's heat transfer rate, which accounts for the heat transfer through both conduction and radiation. A material with lower U-value has a better ability to resist heat conduction. (Brenden, 2010) U-value is more often used to describe transparent envelope materials, such as windows, skylights, and glass curtain walls as it accounts for the heat flow from both the temperature difference and the solar radiation.

SHGC is another value to for transparent envelope materials. SHGC is the percentage of solar radiation incident upon a given window or skylight assembly that ends up in a building as heat. (Stein, B., Reynolds, Grondzik, and Kwok, 2006) As the temperature of each climate zone increases, it requires a lower SHGC for the windows and skylights. However, in a cold climate zone, the SHGC does not quite matter, while it needs all envelope materials to have enough insulation capability so that it can protect the building from losing heat through the envelope in cold temperatures.

Table 5.2-2 shows the R-values of the roof, wall, and ground floor. Each parameter in the table meets the ASHRAE 90.1 requirements.

Table 5.2-2 Opaque Building Envelope Parameters

Opaque Envelope	1A (Miami)	2B (Phoenix)	4C (Seattle)	5B (Spokane)
Roof	R-18.0	R-20.0	R-20.0	R-24.0
Wall	R-13.0	R-13.0	R-19.0 + R-2.0	R-21.0 + R-4.0
Floor	R-10.0	R-10.0	R-10.0	R-10.0

Table 5.2-3 shows the U-values and SHGC of windows and skylights, which are used as input parameters in the energy models.

Table 5.2-3 Transparent Building Envelope Parameters

Transparent Envelope	1A (Miami)	2B (Phoenix)	4C (Seattle)	5B (Spokane)
Window	U-0.54 SHGC-0.23	U-0.47 SHGC-0.23	U-0.40 SHGC-0.38	U-0.34 SHGC-0.39
Skylight	U-0.69 SHGC-0.39	U-0.69 SHGC-0.39	U-0.69 SHGC-0.39	U-0.69 SHGC-0.39

5.3 INTERNAL HEAT GAIN ASSUMPTIONS

In a typical building, the major internal heat gain sources are from the building occupants, lighting, and other equipment. The building occupants contribute to the internal heat gain through the temperature differences and the respiration process.

Table 5.3-1 is the input occupancy densities of different spaces in the energy model, which shows the occupancy density during the hour when 100% of the occupants in the specific space. The assuming total heat generation rate by the building occupants is 450BTU/h/person, which is used to calculate the occupant heat gain per square foot.

Table 5.3-1 also shows the minimum ventilation rate for each space. According to the ASHRAE 62.1-2010 requirements, different functions of spaces require a different rate of ventilation for the occupants.

Table 5.3-1 Occupancy Density

	Design Max Occupancy			Design Min Ventilation	
	ft^2/person	person/ft^2	BTU/ft^2	CFM/person	CFM/ft^2
Classroom	75	0.0133	6.000	15	0.200
Office	200	0.0050	2.250	20	0.100
Cafe	100	0.0100	4.500	20	0.200
Lobby	100	0.0100	4.500	20	0.200
Corridor & Stairs	1000	0.0010	0.450	50	0.050
Mechanical/Electrical	2000	0.0005	0.225	100	0.050
Restroom	300	0.0033	1.500	50	0.167

Table 5.3-2 shows the input parameters of lighting power density and the equipment power density for different spaces in the energy model. These parameters are set based on ASHRAE 90.1-2010 baseline standard.

Table 5.3-2 Lighting and Equipment Power Density

W/ft2	Lighting Power Density	Equipment Power Density
Classroom	1.24	0.75
Office	1.11	0.75
Cafe	0.89	0.50
Lobby	0.90	0.25
Corridor & Stairs	0.69	-
Mechanical/Electrical	0.95	0.10
Restroom	0.98	0.10

5.4 OPERATING SCHEDULES

Typical educational office buildings always have interims between the semesters or quarters. However, to have a consistent energy consumption data from the energy simulation, the analysis must include the coldest and hottest days during a whole year. Thus, in the energy model, the operation schedules follow a typical office building. The building will operate on weekdays during the year without considering the summer and winter breaks between semesters.

Figure 5.4-1 to Figure 5.4-4 show the occupancy, lighting, equipment, and how water schedules of a typical operating workday. The 100% in schedules means the highest density.

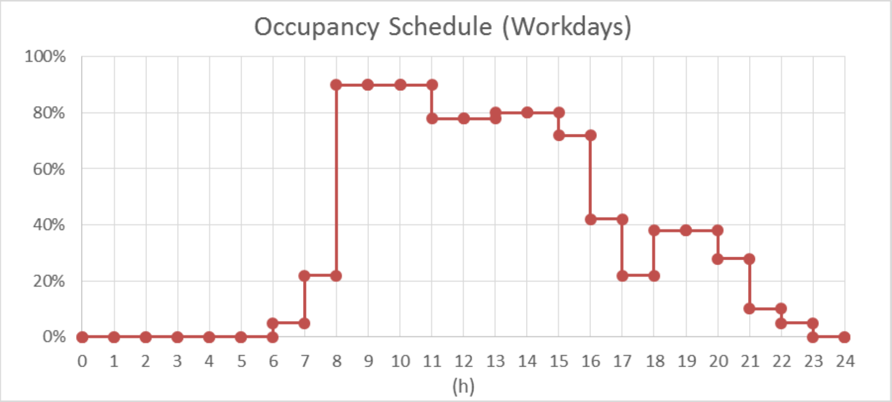


Figure 5.4-1 Occupancy Schedule (eQUEST, Version 3-65)

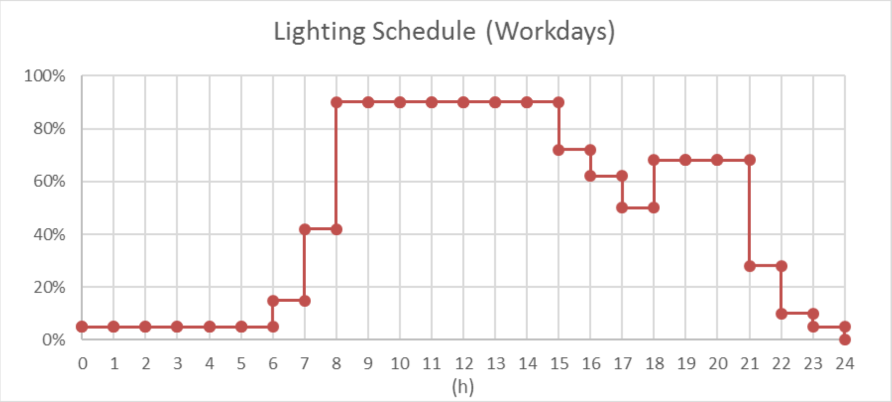


Figure 5.4-2 Lighting Schedule (eQUEST, Version 3-65)

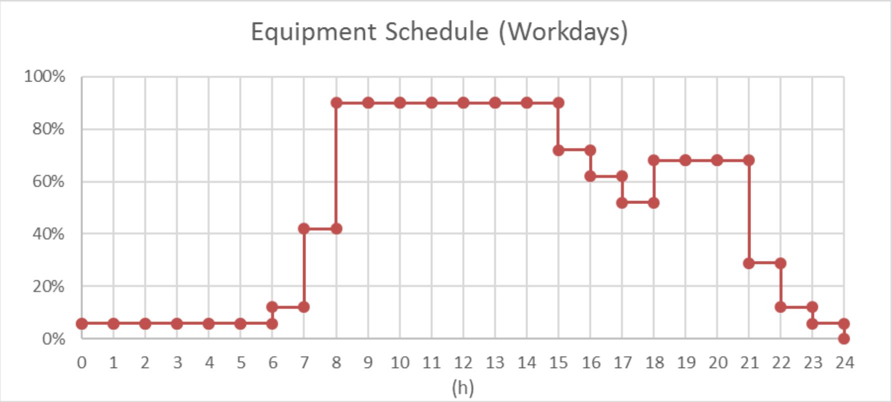


Figure 5.4-3 Equipment Schedule (eQUEST, Version 3-65)

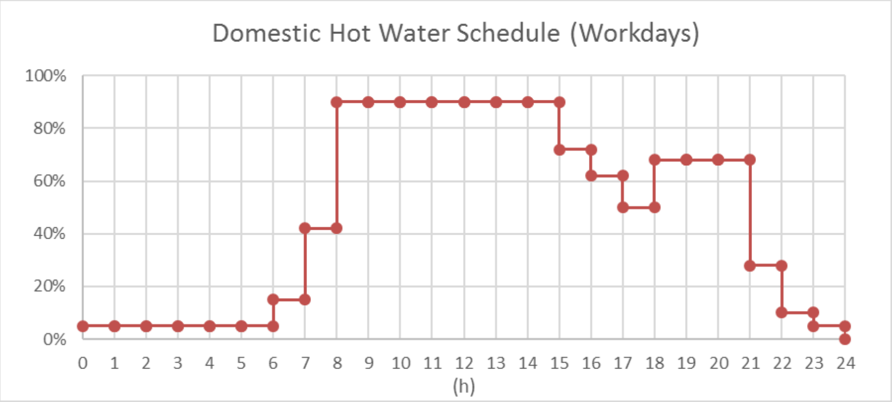


Figure 5.4-4 Domestic Hot Water Schedule (eQUEST, Version 3-65)

CHAPTER 6. SIMULATION RESULT

6.1 BUILDING LOAD

This research focuses on the energy consumption by different HVAC systems, which depends on the heating and cooling demand of the building model. The primary building load is related to the climate, insulation, layout, and the indoor heat generation rate of the building. After setting the building model, envelope materials, room functions, indoor heat densities and schedules, eQUEST can calculate the heating and cooling load for the building model in a particular climate condition.

6.1.1 Miami, FL

Table 6.1-1 and Figures 6.1-1 show the monthly heating and cooling load of the Miami Model.

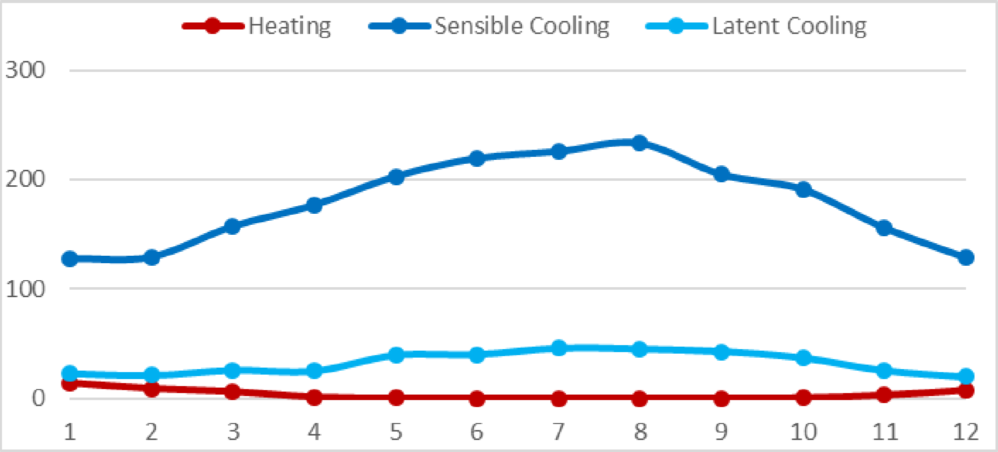


Figure 6.1-1 Heating and Cooling Load of Miami Model (MBTU)

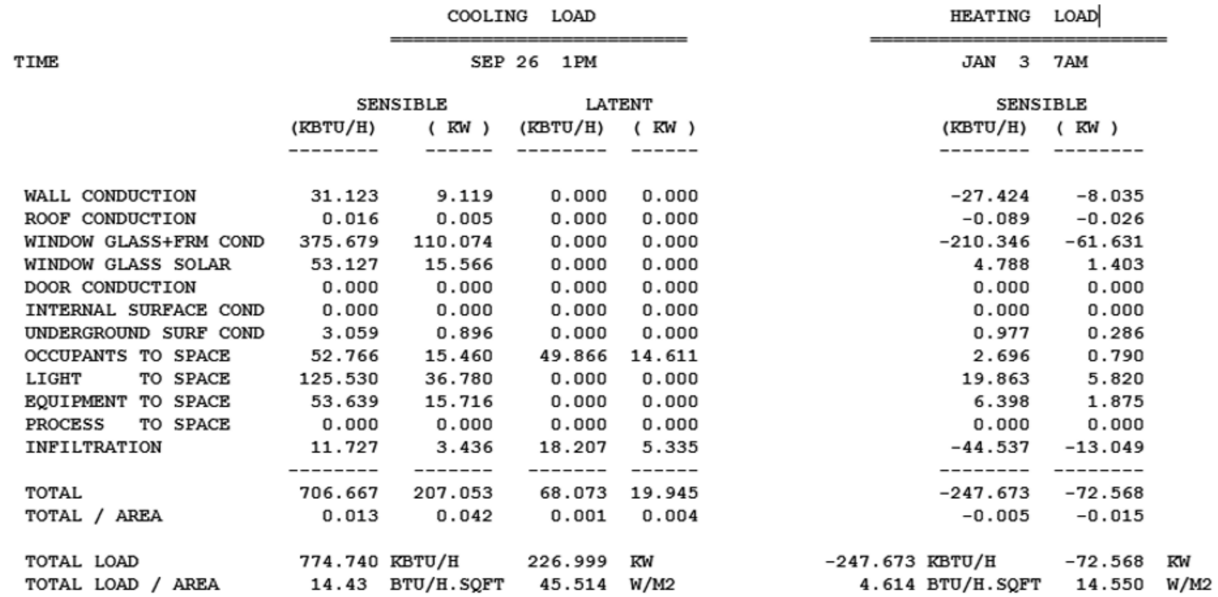
Table 6.1-1 Heating and Cooling Load of Miami Model

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	MBTU
Heating	14.00	8.81	6.11	1.10	0.07	-	-	-	-	0.36	2.64	7.30	**40.40**
Cooling	150.22	149.51	182.20	201.07	241.48	258.71	270.77	277.79	246.91	227.54	180.88	147.83	**2534.91**
Sensitive	127.46	128.58	156.65	176.26	202.43	219.05	225.36	232.95	204.43	190.71	155.57	128.19	2147.61
Latent	22.77	20.93	25.56	24.81	39.06	39.66	45.41	44.84	42.48	36.84	25.31	19.64	387.30

Miami is a cooling dominant climate year round. The annual total cooling load of the Miami model is 2535MBTU, while the annual heating load of the building is 40MBTU. The humidity in

Miami is relatively high during a year, which leads to the highest latent cooling load among the selected four cities in this study.

Figure 6.1-2 shows the peak cooling load and the peak heating load of the Miami model. The peak heating load of the Miami model is 248 BTU/h, which appears at 7 am on January 3; the peak cooling load is 775 BTU/h, which appears at 1 pm on September 26.



```
                                      COOLING   LOAD                              HEATING   LOAD
                                 ==========================                ==========================
TIME                                    SEP 26  1PM                                JAN  3  7AM

                              SENSIBLE            LATENT                            SENSIBLE
                        (KBTU/H)    ( KW )  (KBTU/H)  ( KW )                   (KBTU/H)  ( KW )
                        ---------    ------ ---------  ------                  --------  --------

 WALL CONDUCTION           31.123     9.119     0.000   0.000                      -27.424    -8.035
 ROOF CONDUCTION            0.016     0.005     0.000   0.000                       -0.089    -0.026
 WINDOW GLASS+FRM COND    375.679   110.074     0.000   0.000                     -210.346   -61.631
 WINDOW GLASS SOLAR        53.127    15.566     0.000   0.000                        4.788     1.403
 DOOR CONDUCTION            0.000     0.000     0.000   0.000                        0.000     0.000
 INTERNAL SURFACE COND      0.000     0.000     0.000   0.000                        0.000     0.000
 UNDERGROUND SURF COND      3.059     0.896     0.000   0.000                        0.977     0.286
 OCCUPANTS TO SPACE        52.766    15.460    49.866  14.611                        2.696     0.790
 LIGHT     TO SPACE       125.530    36.780     0.000   0.000                       19.863     5.820
 EQUIPMENT TO SPACE        53.639    15.716     0.000   0.000                        6.398     1.875
 PROCESS   TO SPACE         0.000     0.000     0.000   0.000                        0.000     0.000
 INFILTRATION              11.727     3.436    18.207   5.335                      -44.537   -13.049
                        ---------   -------  --------  ------                     --------  --------
 TOTAL                    706.667   207.053    68.073  19.945                     -247.673   -72.568
 TOTAL / AREA               0.013     0.042     0.001   0.004                       -0.005    -0.015

 TOTAL LOAD               774.740 KBTU/H      226.999  KW             -247.673 KBTU/H        -72.568  KW
 TOTAL LOAD / AREA         14.43  BTU/H.SQFT   45.514  W/M2              4.614 BTU/H.SQFT    14.550  W/M2
```

Figure 6.1-2 Peak Load of Miami Model (eQUEST, Version 3-65)

6.1.2 Phoenix, AZ

Figure 6.1-3 and Table 6.1-2 show the monthly load of the Phoenix model.

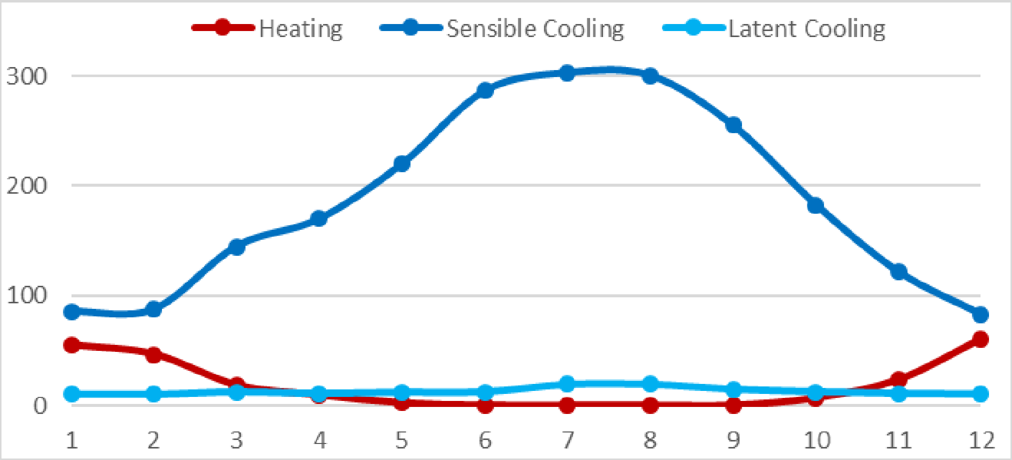


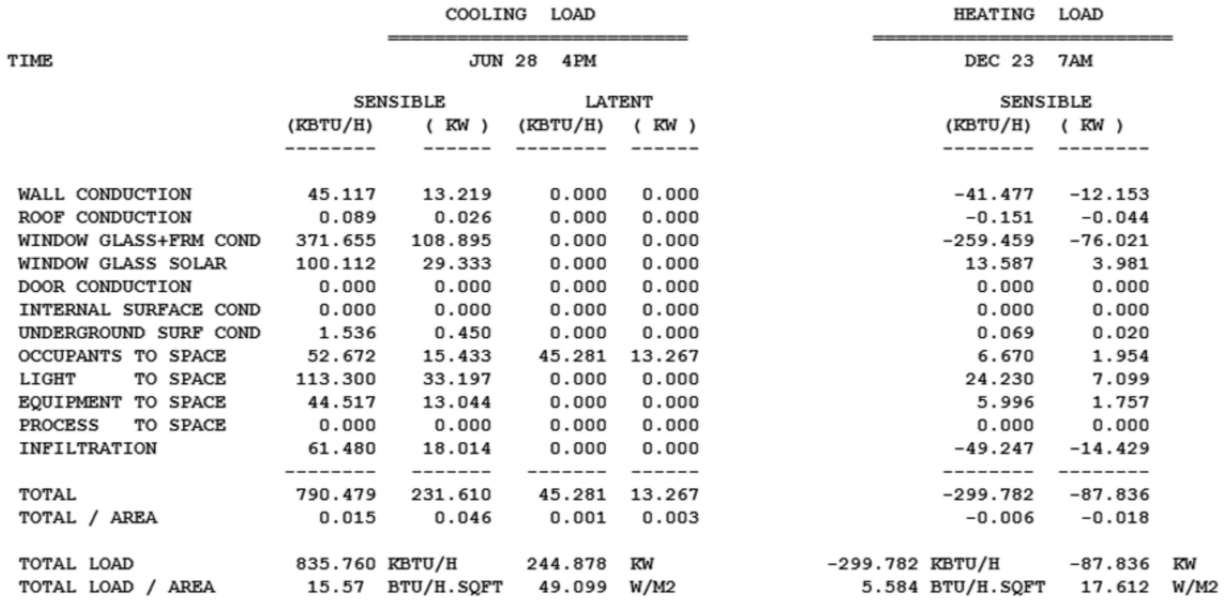
Figure 6.1-3 Heating and Cooling Load of Phoenix Model (MBTU)

Table 6.1-2 Heating and Cooling Load of Phoenix Model

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	MBTU
Heating	54.93	46.56	18.45	9.29	2.16	0.05	-	-	0.07	6.71	23.30	60.49	**222.00**
Cooling	95.56	97.41	157.38	180.77	232.14	298.98	322.50	319.88	269.99	195.31	132.67	93.05	**2395.64**
Sensitive	85.18	87.39	144.85	169.75	219.96	286.42	303.04	300.37	254.95	182.75	121.51	82.66	2238.82
Latent	10.38	10.02	12.54	11.02	12.19	12.56	19.46	19.50	15.04	12.57	11.16	10.39	156.82

Phoenix experiences a dry and hot climate during the year, but it does require 222 MBTU of heating. The annual cooling load of the Phoenix model is 2396 MBTU. Phoenix has the highest sensible cooling load (July - 303 MBTU) and the total cooling load (July - 322.5 MBTU) among the selected four cities.

Figure 6.1-4 shows the peak cooling load and the peak heating load of the Phoenix model. The peak heating load of the Phoenix model is 300 BTU/h, which appears at 7 am on December 23; the peak cooling load is 836 BTU/h, which appears at 4 pm on June 28.



```
                                        COOLING  LOAD                                  HEATING  LOAD
                                   ===========================                  ===========================
 TIME                                   JUN 28  4PM                                   DEC 23  7AM

                                SENSIBLE              LATENT                              SENSIBLE
                           (KBTU/H)     ( KW )  (KBTU/H)   ( KW )                   (KBTU/H)   ( KW )
                           --------     ------  --------   ------                   --------   --------

  WALL CONDUCTION           45.117     13.219     0.000    0.000                    -41.477    -12.153
  ROOF CONDUCTION            0.089      0.026     0.000    0.000                     -0.151     -0.044
  WINDOW GLASS+FRM COND    371.655    108.895     0.000    0.000                   -259.459    -76.021
  WINDOW GLASS SOLAR       100.112     29.333     0.000    0.000                     13.587      3.981
  DOOR CONDUCTION            0.000      0.000     0.000    0.000                      0.000      0.000
  INTERNAL SURFACE COND      0.000      0.000     0.000    0.000                      0.000      0.000
  UNDERGROUND SURF COND      1.536      0.450     0.000    0.000                      0.069      0.020
  OCCUPANTS TO SPACE        52.672     15.433    45.281   13.267                      6.670      1.954
  LIGHT     TO SPACE       113.300     33.197     0.000    0.000                     24.230      7.099
  EQUIPMENT TO SPACE        44.517     13.044     0.000    0.000                      5.996      1.757
  PROCESS   TO SPACE         0.000      0.000     0.000    0.000                      0.000      0.000
  INFILTRATION              61.480     18.014     0.000    0.000                    -49.247    -14.429
                           --------    -------   -------   ------                   --------   --------
  TOTAL                    790.479    231.610    45.281   13.267                   -299.782    -87.836
  TOTAL / AREA               0.015      0.046     0.001    0.003                     -0.006     -0.018

  TOTAL LOAD               835.760 KBTU/H       244.878  KW               -299.782 KBTU/H       -87.836  KW
  TOTAL LOAD / AREA         15.57  BTU/H.SQFT    49.099  W/M2                5.584 BTU/H.SQFT   17.612  W/M2
```

Figure 6.1-4 Peak Load of Phoenix Model (eQUEST, Version 3-65)

6.1.3 Seattle, WA

Figure 6.1-5 and Table 6.1-3 show the heating and cooling load of the model of Seattle.

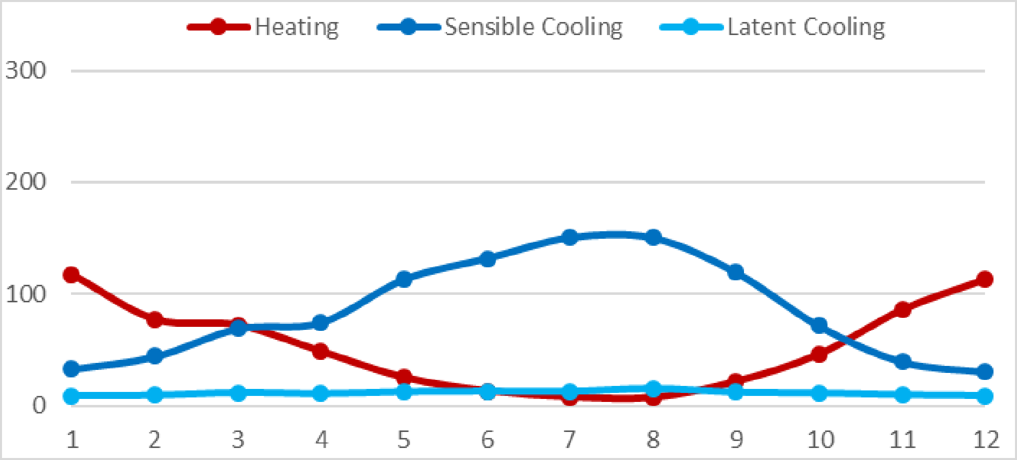
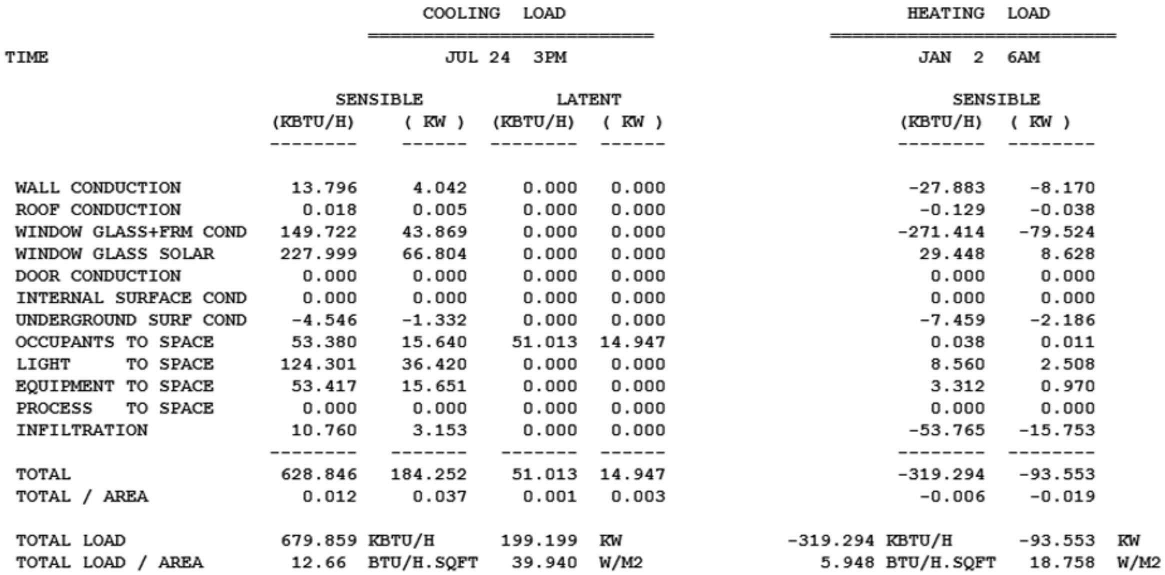


Figure 6.1-5 Heating and Cooling Load of Seattle Model (MBTU)

Table 6.1-3 Heating and Cooling Load of Seattle Model

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	MBTU
Heating	117.58	77.17	72.01	48.33	24.89	12.93	7.44	6.97	21.31	45.98	85.82	113.11	**633.53**
Cooling	41.79	53.24	80.58	85.14	125.44	144.68	162.97	165.17	130.89	82.50	48.92	39.27	**1160.60**
Sensitive	32.81	44.09	69.08	74.60	113.22	131.96	150.51	150.10	119.14	71.33	39.31	30.38	1026.54
Latent	8.98	9.15	11.50	10.54	12.22	12.72	12.45	15.07	11.75	11.18	9.61	8.90	134.06

Seattle has a heating dominant climate. The temperature in Seattle is not too hot or too cold. This model requires 634 MBTU of heating load and 1161 MBTU of cooling load annually. The relative humidity in Seattle is very high during winter, but heating can automatically reduce the moisture level of the air.



```
                                      COOLING   LOAD                              HEATING   LOAD
                                 ==========================                ==========================
TIME                                    JUL 24   3PM                               JAN  2  6AM

                              SENSIBLE            LATENT                              SENSIBLE
                        (KBTU/H)    ( KW )  (KBTU/H)  ( KW )                     (KBTU/H)  ( KW )
                        --------    ------  --------  ------                     --------  --------

 WALL CONDUCTION          13.796     4.042     0.000   0.000                      -27.883    -8.170
 ROOF CONDUCTION           0.018     0.005     0.000   0.000                       -0.129    -0.038
 WINDOW GLASS+FRM COND   149.722    43.869     0.000   0.000                     -271.414   -79.524
 WINDOW GLASS SOLAR      227.999    66.804     0.000   0.000                       29.448     8.628
 DOOR CONDUCTION           0.000     0.000     0.000   0.000                        0.000     0.000
 INTERNAL SURFACE COND     0.000     0.000     0.000   0.000                        0.000     0.000
 UNDERGROUND SURF COND    -4.546    -1.332     0.000   0.000                       -7.459    -2.186
 OCCUPANTS TO SPACE       53.380    15.640    51.013  14.947                        0.038     0.011
 LIGHT     TO SPACE      124.301    36.420     0.000   0.000                        8.560     2.508
 EQUIPMENT TO SPACE       53.417    15.651     0.000   0.000                        3.312     0.970
 PROCESS   TO SPACE        0.000     0.000     0.000   0.000                        0.000     0.000
 INFILTRATION             10.760     3.153     0.000   0.000                      -53.765   -15.753
                        --------   -------   -------  ------                     --------  --------
 TOTAL                   628.846   184.252    51.013  14.947                     -319.294   -93.553
 TOTAL / AREA              0.012     0.037     0.001   0.003                       -0.006    -0.019

 TOTAL LOAD              679.859 KBTU/H      199.199   KW              -319.294 KBTU/H      -93.553   KW
 TOTAL LOAD / AREA        12.66  BTU/H.SQFT   39.940   W/M2               5.948 BTU/H.SQFT   18.758   W/M2
```

Figure 6.1-6 Peak Load of Seattle Model (eQUEST, Version 3-65)

Figure 6.1-6 shows the peak cooling load and the peak heating load of the Seattle model. The peak heating load of the Seattle model is 319 BTU/h, which appears at 6 am on January 2; the peak cooling load is 680 BTU/h, which appears at 3 pm on July 24.

6.1.4 Spokane, WA

Figure 6.1-7 and Table 6.1-4 exhibit the heating and cooling load of the building model in Spokane.

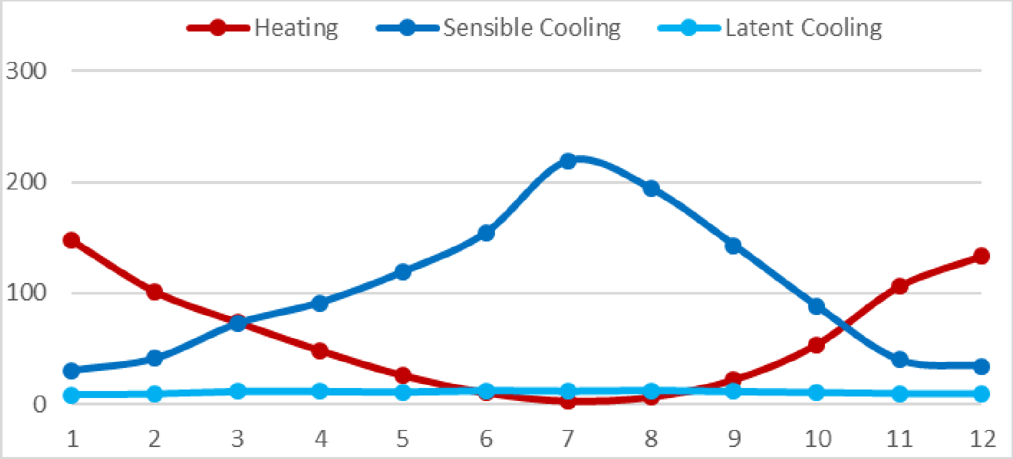


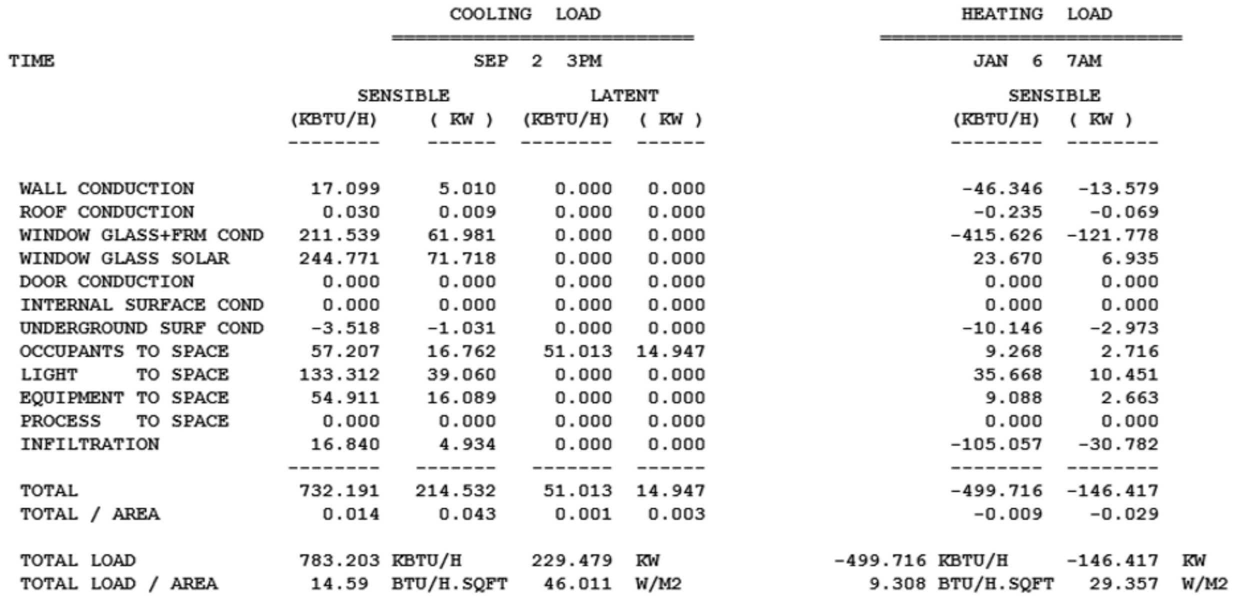
Figure 6.1-7 Heating and Cooling Load of Spokane Model (MBTU)

Table 6.1-4 Heating and Cooling Load of Spokane Model

	Jan	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**MBTU**
Heating	147.59	101.67	74.07	48.41	25.90	10.34	2.85	6.47	22.09	53.95	106.42	133.86	**733.62**
Cooling	38.46	50.01	84.03	102.95	129.96	166.59	230.67	206.44	154.37	98.76	49.24	43.57	**1355.05**
Sensitive	30.32	41.08	72.52	91.35	119.06	154.14	218.85	193.88	142.93	88.30	39.90	34.40	1226.73
Latent	8.14	8.92	11.51	11.61	10.90	12.45	11.82	12.56	11.43	10.46	9.34	9.17	128.32

Spokane experiences a large range of temperature. The summer is warm and dry, while the winter is cold and humid. The Spokane model requires for more heating and cooling than the Seattle model. The annual heating load of Spokane model is 734 MBTU, and the annual cooling load is 1355 MBTU.

Figure 6.1-8 shows the peak cooling load and the peak heating load of the Spokane model. The peak heating load of the Spokane model is 500 BTU/h, which appears at 7 am on January 6; the peak cooling load is 783 BTU/h, which appears at 3 pm on Sep 2.



```
                                       COOLING   LOAD                                  HEATING   LOAD
                                 ==========================                     ==========================
TIME                                   SEP  2   3PM                                    JAN  6  7AM

                                 SENSIBLE          LATENT                                SENSIBLE
                            (KBTU/H)    ( KW )  (KBTU/H)   ( KW )                     (KBTU/H)   ( KW )
                            --------    ------  --------   ------                     --------   --------

 WALL CONDUCTION             17.099      5.010     0.000    0.000                      -46.346    -13.579
 ROOF CONDUCTION              0.030      0.009     0.000    0.000                       -0.235     -0.069
 WINDOW GLASS+FRM COND      211.539     61.981     0.000    0.000                     -415.626   -121.778
 WINDOW GLASS SOLAR         244.771     71.718     0.000    0.000                       23.670      6.935
 DOOR CONDUCTION              0.000      0.000     0.000    0.000                        0.000      0.000
 INTERNAL SURFACE COND        0.000      0.000     0.000    0.000                        0.000      0.000
 UNDERGROUND SURF COND       -3.518     -1.031     0.000    0.000                      -10.146     -2.973
 OCCUPANTS TO SPACE          57.207     16.762    51.013   14.947                        9.268      2.716
 LIGHT     TO SPACE         133.312     39.060     0.000    0.000                       35.668     10.451
 EQUIPMENT TO SPACE          54.911     16.089     0.000    0.000                        9.088      2.663
 PROCESS   TO SPACE           0.000      0.000     0.000    0.000                        0.000      0.000
 INFILTRATION                16.840      4.934     0.000    0.000                     -105.057    -30.782
                            --------   -------   -------   ------                     --------   --------

 TOTAL                      732.191    214.532    51.013   14.947                     -499.716   -146.417
 TOTAL / AREA                 0.014      0.043     0.001    0.003                       -0.009     -0.029

 TOTAL LOAD                 783.203 KBTU/H      229.479  KW               -499.716 KBTU/H     -146.417  KW
 TOTAL LOAD / AREA           14.59  BTU/H.SQFT   46.011  W/M2                9.308 BTU/H.SQFT   29.357  W/M2
```

Figure 6.1-8 Peak Load of Spokane Model (eQUEST, Version 3-65)

6.1.5 Summary

Figure 6.1-9 summarizes the heating load, sensible cooling load and latent cooling load of different locations.

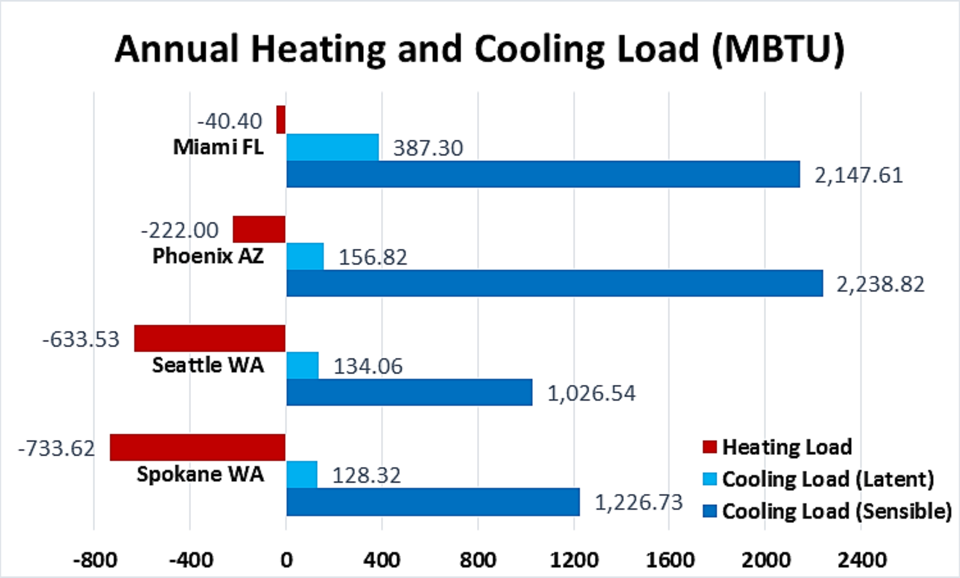


Figure 6.1-9 Annual Heating and Cooling Load Comparison

Figure 6.1-10 and Table 6.1-5 summarize the annual peak heating and cooling load and their appearance time.

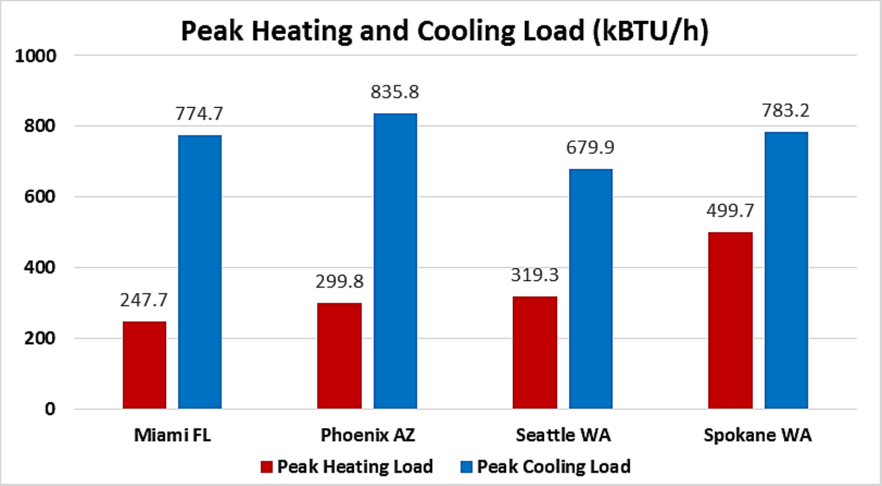


Figure 6.1-10 Peak Heating and Cooling Load

Table 6.1-5 Peak Heating and Cooling Load

Peak Load	Heating (kBTU/h)	Time	Cooling (kBTU/h)	Time
Miami FL	247.7	7am Jan 3	774.7	1pm Sep 26
Phoenix AZ	299.8	7am Dec 23	835.8	4pm Jun 28
Seattle WA	319.3	6am Jan 2	679.9	3pm Jul 24
Spokane WA	499.7	7am Jan 6	783.2	3pm Sep 2

6.2 ENERGY CONSUMPTION

All the eQUEST energy models are set to use only electricity as their heating and cooling source; this allows the energy consumption by different HVAC systems to be compared in a straightforward approach.

The following content in this chapter analyzes sixteen energy models by the climate zones and then by systems.

6.2.1 Miami, FL

Figure 6.2-1 shows the energy consumption result by the energy models of VAV reheat system, chilled beam system, ASHP system and GSHP system in Miami.

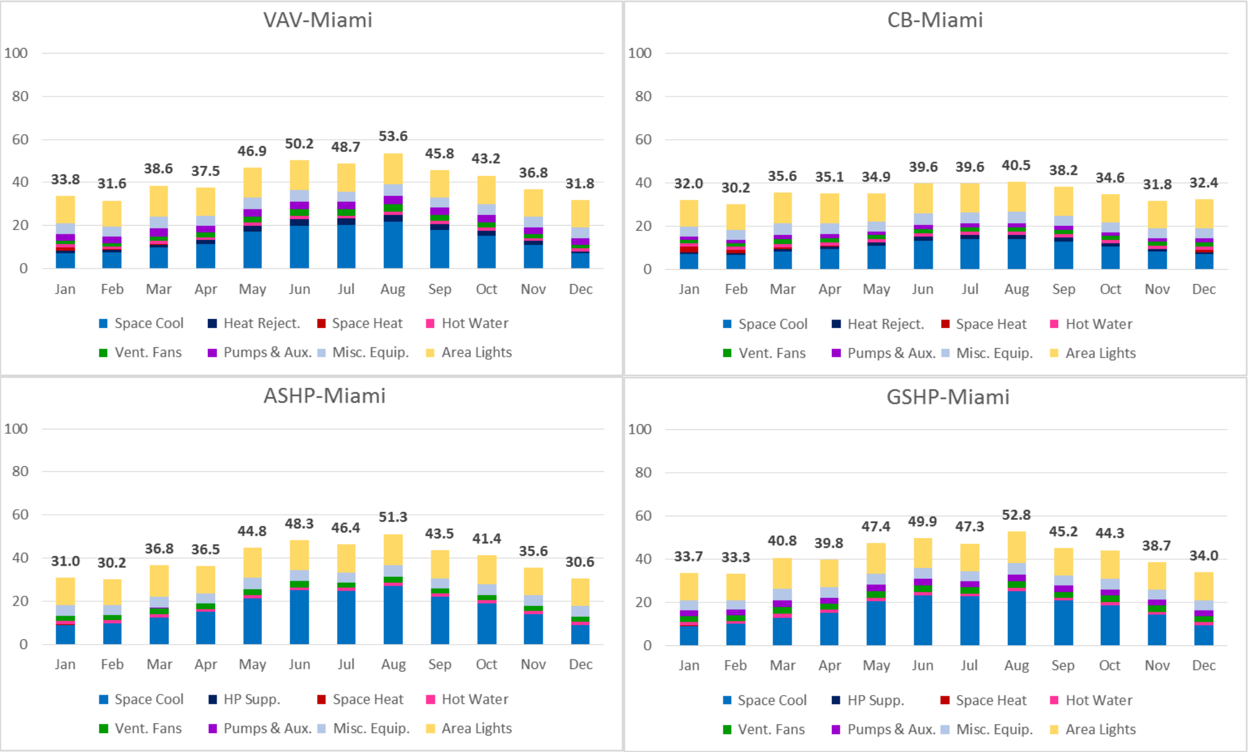


Figure 6.2-1 Monthly Electric Consumption in Miami (10^3 kWh)

Table 6.2-1 Monthly Electric Consumption Data of Miami Models (10^3 kWh)

VAV Reheat	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	7.37	7.74	9.85	11.64	17.42	20.05	20.26	21.92	18.16	15.36	11.18	7.05	**167.99**
Heat Reject.	1.03	1.06	1.34	1.60	2.59	2.95	2.95	3.19	2.68	2.25	1.63	0.90	**24.17**
Space Heat	1.59	0.23	0.30	-	-	-	-	-	-	-	-	0.10	**2.22**
Hot Water	1.48	1.41	1.69	1.48	1.56	1.50	1.35	1.49	1.32	1.40	1.38	1.44	**17.49**
Vent. Fans	1.44	1.46	1.81	1.94	2.59	3.07	3.18	3.40	2.77	2.39	1.76	1.44	**27.27**
Pumps. Aux.	3.33	3.16	3.82	3.33	3.66	3.66	3.33	3.82	3.33	3.49	3.33	3.33	**41.57**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**33.81**	**31.63**	**38.60**	**37.47**	**46.87**	**50.20**	**48.65**	**53.63**	**45.75**	**43.22**	**36.76**	**31.82**	**498.43**
Chilled Beam	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	6.89	6.58	8.30	9.31	10.78	13.24	13.99	13.97	12.88	10.54	8.18	7.01	**121.67**
Heat Reject.	0.91	0.82	1.06	1.27	1.57	1.90	2.04	2.03	1.89	1.53	1.16	0.82	**17.00**
Space Heat	2.58	1.44	0.67	0.08	-	-	-	-	-	-	-	1.07	**5.83**
Hot Water	1.42	1.41	1.69	1.60	1.44	1.50	1.40	1.43	1.37	1.35	1.38	1.50	**17.48**
Vent. Fans	1.78	1.78	2.16	2.06	1.87	2.06	1.96	2.06	1.96	1.87	1.87	1.97	**23.40**
Pumps. Aux.	1.58	1.58	1.92	1.84	1.70	1.91	1.84	1.92	1.82	1.70	1.66	1.75	**21.23**
Misc. Equip.	4.52	4.45	5.32	5.10	4.72	5.10	4.92	5.12	4.90	4.72	4.70	4.92	**58.49**
Area Lights	12.32	12.12	14.48	13.87	12.86	13.87	13.40	13.94	13.33	12.86	12.79	13.40	**159.23**
Total	**32.00**	**30.18**	**35.59**	**35.13**	**34.94**	**39.58**	**39.55**	**40.47**	**38.15**	**34.56**	**31.75**	**32.43**	**424.32**

ASHP	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	8.87	9.81	12.34	15.04	21.47	25.06	24.94	27.11	22.26	19.03	14.21	9.13	**209.25**
HP Supp.	0.01	-	-	-	-	-	-	-	-	-	-	-	**0.01**
Space Heat	0.51	0.05	0.07	-	-	-	-	-	-	-	-	0.01	**0.64**
Hot Water	1.48	1.42	1.69	1.48	1.56	1.50	1.35	1.49	1.32	1.40	1.38	1.44	**17.50**
Vent. Fans	2.48	2.36	2.85	2.48	2.73	2.73	2.48	2.85	2.48	2.60	2.48	2.48	**31.01**
Pumps. Aux.	0.08	0.02	0.02	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	**0.12**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**31.00**	**30.22**	**36.77**	**36.49**	**44.82**	**48.26**	**46.35**	**51.25**	**43.54**	**41.36**	**35.56**	**30.64**	**476.25**

GSHP	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	9.06	10.02	12.92	15.28	20.69	23.41	22.83	25.13	20.88	18.77	14.33	9.44	**202.76**
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	**0.00**
Space Heat	0.20	0.02	0.02	-	-	-	-	-	-	-	-	-	**0.24**
Hot Water	1.48	1.42	1.69	1.48	1.56	1.50	1.35	1.49	1.32	1.40	1.38	1.44	**17.50**
Vent. Fans	2.78	2.64	3.19	2.78	3.05	3.05	2.78	3.19	2.78	2.91	2.78	2.78	**34.70**
Pumps. Aux.	2.65	2.59	3.14	2.74	3.01	3.01	2.74	3.15	2.74	2.87	2.74	2.72	**34.08**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**33.74**	**33.25**	**40.76**	**39.76**	**47.37**	**49.94**	**47.27**	**52.76**	**45.20**	**44.28**	**38.72**	**33.96**	**507.00**

According to the energy consumption result from the Miami models, the total electricity consumed by the VAV reheat system model is $498.43\times 10^3 kWh$; the chilled beam system model uses $424.32\times 10^3 kWh$ of electricity, the model with ASHP system consumes $476.25\times 10^3 kWh$, and GSHP system consumes $507.00\times 10^3 kWh$.

The chilled beam model in Miami consumes the lowest energy and provides an even monthly energy consumption over a year. The GSHP system model consumes the highest amount of energy in Miami.

As discussed in Chapter 4.3, chilled beam system requires less cooling or heating energy than the VAV reheat system. From the simulation result, chilled beam system consumes less energy for cooling, but uses more energy for heating, which is not consistent with the assumptions that chilled beam system is more efficient to heat and cool a space. However, a chilled beam system does consume less overall energy on the total.

According to the result, the GSHP system consumes the highest amount of energy among the four systems in Miami. The GSHP system needs to move the refrigerant medium through the long underground pipes. What’s more, when comparing the energy consumption of the Miami GSHP model with the GSHP system in the other three cities, the Miami model is still the highest. As noted before in Figure 4.5-1 from the chapter 4.5, the temperature difference between the ground layer and the 10 feet deep underground layer in Miami is less than 5°F. Thus, it is tough to transfer heat through the underground pipes, which is a reason why GSHP system is not efficient in a climate zone like Miami.

6.2.2 Phoenix, AZ

Figure 6.2-2 and Table 6.2-2 exhibit the monthly energy consumption data by the four energy models of different HVAC systems in Phoenix.

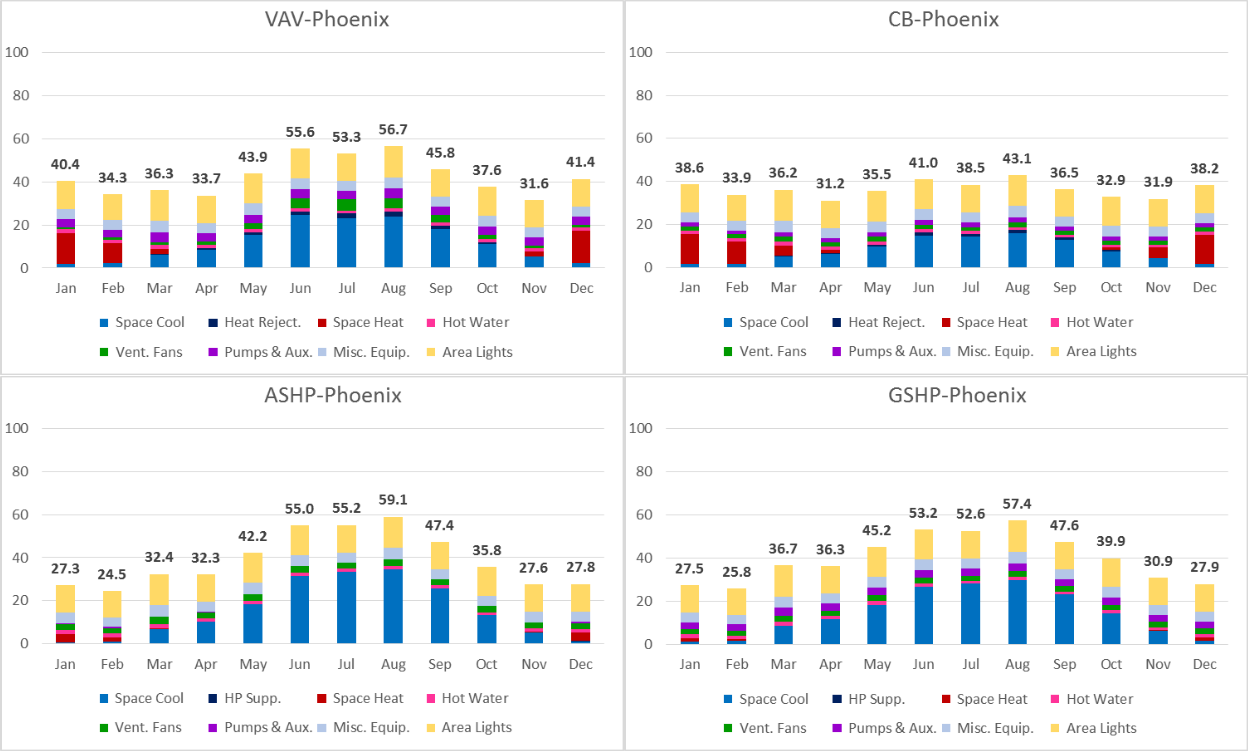


Figure 6.2-2 Monthly Electric Consumption in Phoenix (10^3 kWh)

Table 6.2-2 Monthly Electric Consumption Data of Phoenix Models (10^3 kWh)

VAV Reheat	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	2.04	2.28	6.31	8.64	15.36	24.47	23.21	24.06	18.14	11.21	5.28	2.32	**143.32**
Heat Reject.	0.03	0.05	0.34	0.55	1.07	1.88	2.17	2.24	1.64	0.83	0.28	0.05	**11.11**
Space Heat	14.23	9.21	2.22	0.04	-	-	-	-	-	0.05	2.24	15.02	**43.01**
Hot Water	1.66	1.63	1.94	1.68	1.69	1.54	1.31	1.40	1.24	1.36	1.41	1.55	**18.41**
Vent. Fans	1.00	0.94	1.32	1.48	2.51	4.56	5.11	4.84	3.50	1.85	1.07	0.99	**29.18**
Pumps. Aux.	3.85	3.63	4.33	3.83	4.17	4.15	3.85	4.33	3.83	4.01	3.83	3.85	**47.71**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**40.39**	**34.30**	**36.26**	**33.72**	**43.86**	**55.56**	**53.25**	**56.67**	**45.83**	**37.63**	**31.60**	**41.37**	**510.44**

Chilled Beam	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	1.66	1.68	5.24	6.51	9.87	15.05	14.30	15.98	12.80	7.62	4.35	1.84	**96.91**
Heat Reject.	0.03	0.04	0.30	0.41	0.70	1.18	1.36	1.46	1.12	0.56	0.25	0.05	**7.47**
Space Heat	13.92	10.43	4.71	1.33	0.05	-	0.05	-	-	1.11	4.70	13.49	**49.79**
Hot Water	1.66	1.63	1.94	1.68	1.69	1.53	1.31	1.40	1.23	1.36	1.41	1.55	**18.38**
Vent. Fans	1.88	1.78	2.16	1.88	2.07	2.07	1.88	2.16	1.88	1.97	1.88	1.88	**23.48**
Pumps. Aux.	1.84	1.73	2.08	1.87	2.08	2.18	2.03	2.28	1.99	1.97	1.85	1.84	**23.73**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**38.56**	**33.86**	**36.22**	**31.16**	**35.52**	**40.98**	**38.50**	**43.07**	**36.51**	**32.91**	**31.92**	**38.23**	**437.46**

ASHP	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	0.69	0.92	7.02	10.28	18.40	31.45	33.56	34.69	25.89	13.25	5.48	1.06	**182.69**
HP Supp.	0.09	0.07	-	-	-	-	-	-	-	-	-	0.25	**0.41**
Space Heat	3.87	2.14	0.34	0.02	0.00	0.00	0.00	0.00	0.00	0.01	0.29	3.92	**10.60**
Hot Water	1.67	1.63	1.94	1.68	1.69	1.54	1.31	1.40	1.23	1.36	1.41	1.56	**18.42**
Vent. Fans	2.75	2.61	3.16	2.75	3.02	3.02	2.75	3.16	2.75	2.89	2.75	2.75	**34.36**
Pumps. Aux.	0.60	0.55	0.13	0.04	0.00	0.00	0.00	0.00	0.00	0.00	0.12	0.64	**2.09**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**27.25**	**24.50**	**32.39**	**32.26**	**42.17**	**54.98**	**55.20**	**59.05**	**47.36**	**35.83**	**27.55**	**27.76**	**466.29**

GSHP	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	1.22	1.48	8.40	11.52	18.22	26.52	28.07	29.71	23.22	14.34	6.40	1.63	**170.72**
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	**0.00**
Space Heat	1.60	0.84	0.12	-	-	-	-	-	-	-	0.10	1.59	**4.27**
Hot Water	1.67	1.63	1.94	1.68	1.69	1.54	1.31	1.40	1.23	1.36	1.41	1.56	**18.42**
Vent. Fans	2.44	2.32	2.81	2.44	2.69	2.69	2.44	2.81	2.44	2.56	2.44	2.44	**30.53**
Pumps. Aux.	2.98	2.90	3.60	3.17	3.49	3.49	3.17	3.65	3.17	3.33	3.09	3.11	**39.14**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**27.49**	**25.75**	**36.67**	**36.31**	**45.15**	**53.20**	**52.57**	**57.37**	**47.56**	**39.91**	**30.93**	**27.91**	**480.80**

Similar to Miami, the chilled beam system in Phoenix consumes the lowest energy, 437.46× $10^3 kWh$. While, the VAV reheat system consumes 510.44× $10^3 kWh$ of energy, which is the highest. The model with ASHP system consumes 466.29× $10^3 kWh$. The GSHP model in Phoenix is better than that in Miami, which consumes 480.80× $10^3 kWh$, but is still very high. Although the temperature difference between the ground layer and the 10 feet deep underground layer in Phoenix is higher (> 10℉), but due to the large cooling demand, GSHP may not provide enough cooling for Phoenix.

Figure 6.2-3 compares the heating and cooling load between Miami and Phoenix.

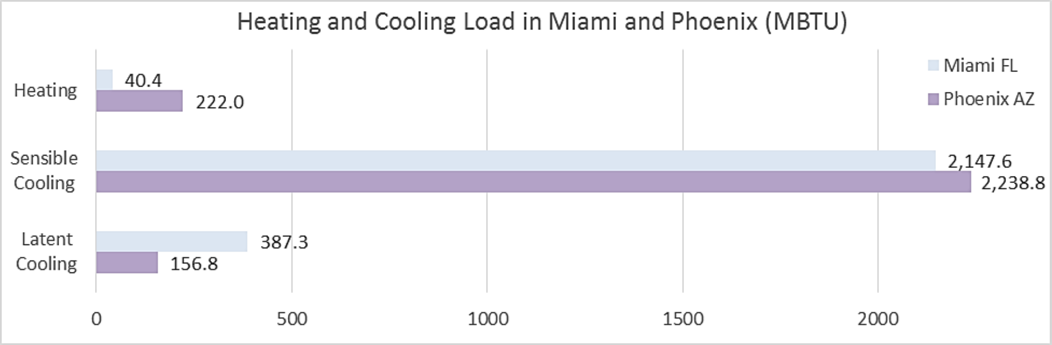


Figure 6.2-3 Load Comparisons between Miami and Phoenix

Miami and Phoenix are both cooling dominant climate, but Phoenix requires more heating in winter. The sensible cooling load in Phoenix is slightly (4%) larger than the sensible cooling load in Miami, but the latent cooling load in Phoenix is only 40% of the latent cooling load in Miami.

Figure 6.2-4 compares the energy used for space heating and cooling in Miami and Phoenix.

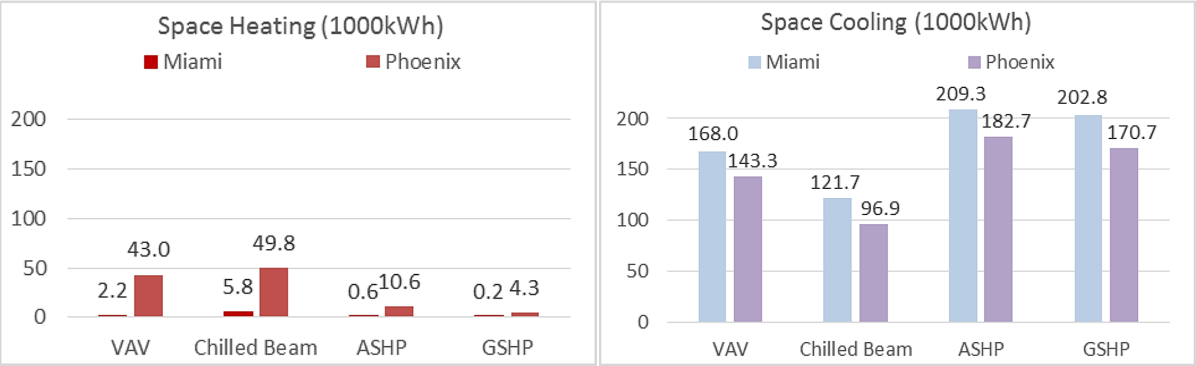


Figure 6.2-4 Energy Usage for Space Heating Cooling in Miami and Phoenix

Although Phoenix has a larger sensible cooling load, all the systems in Phoenix use less energy for space cooling than the systems in Miami, which indicates that it is more challenging to provide cool air in the humid climate.

The VAV reheat system model also includes an open cooling tower on the condenser side (water head– 51.6 feet), so it needs to use more electricity to pump up the water on the condenser side to the cooling tower. Again, the chilled beam system consumes the least electricity in Phoenix.

6.2.3 Seattle, WA

Figure 6.2-5 shows the monthly energy consumption result by eQUEST models of VAV reheat system, Chilled Beam system, ASHP system and GSHP system. Table 6.2-3 shows the specific energy usage data of each component by different energy models.

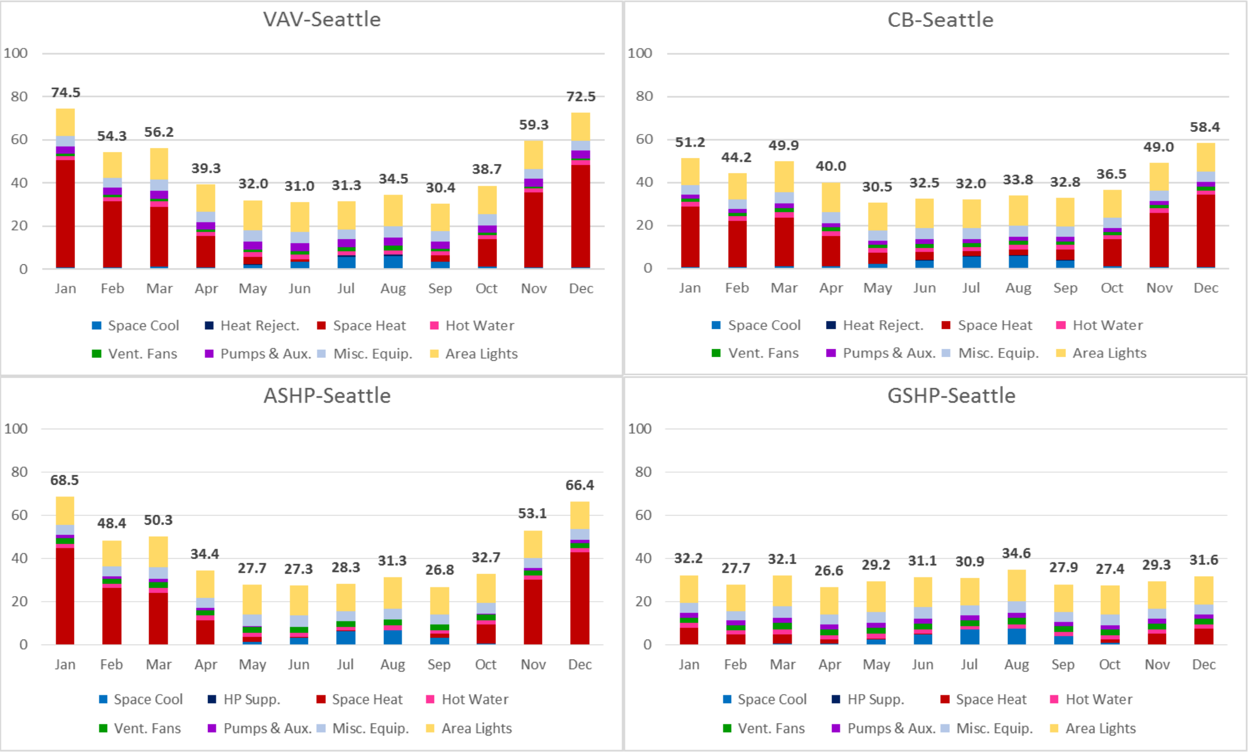


Figure 6.2-5 Monthly Electric Consumption in Seattle (10^3 kWh)

Table 6.2-3 Monthly Electric Consumption Data of Seattle Models (10^3 kWh)

VAV Reheat	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	0.92	0.89	1.16	0.98	2.06	3.26	5.72	6.06	3.30	1.28	0.91	0.91	**27.45**
Heat Reject.	-	-	0.01	0.00	0.11	0.22	0.54	0.56	0.26	0.02	-	-	**1.74**
Space Heat	49.48	30.63	27.83	14.33	3.53	1.08	0.12	-	2.76	12.47	34.56	47.49	**224.28**
Hot Water	2.07	1.98	2.37	2.07	2.17	2.07	1.85	2.03	1.80	1.93	1.91	2.00	**24.25**
Vent. Fans	1.00	0.94	1.16	1.01	1.32	1.71	2.00	2.12	1.35	1.10	0.99	1.00	**15.69**
Pumps. Aux.	3.48	3.27	3.88	3.45	3.75	3.72	3.48	3.88	3.45	3.61	3.45	3.48	**42.91**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**74.53**	**54.28**	**56.21**	**39.33**	**31.99**	**31.04**	**31.28**	**34.46**	**30.41**	**38.72**	**59.32**	**72.45**	**554.03**

Chilled Beam	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	0.58	0.60	0.85	1.10	2.05	3.70	5.29	5.68	3.53	1.18	0.80	0.63	**25.99**
Heat Reject.	-	-	-	0.03	0.12	0.26	0.48	0.52	0.30	0.03	0.01	-	**1.74**
Space Heat	28.36	21.70	22.89	13.79	5.23	3.66	2.27	2.72	5.15	12.33	25.18	33.64	**176.93**
Hot Water	1.98	1.98	2.37	2.25	2.00	2.07	1.93	1.96	1.87	1.85	1.91	2.08	**24.25**
Vent. Fans	1.62	1.60	1.94	1.85	1.68	1.85	1.77	1.85	1.77	1.68	1.73	1.79	**21.14**
Pumps. Aux.	1.81	1.76	2.09	2.00	1.87	2.01	1.98	2.05	1.95	1.87	1.87	1.96	**23.22**
Misc. Equip.	4.52	4.45	5.32	5.10	4.72	5.10	4.92	5.12	4.90	4.72	4.70	4.92	**58.49**
Area Lights	12.32	12.12	14.48	13.87	12.86	13.87	13.40	13.94	13.33	12.86	12.79	13.40	**159.23**
Total	**51.19**	**44.21**	**49.94**	**39.99**	**30.53**	**32.53**	**32.03**	**33.84**	**32.80**	**36.51**	**48.99**	**58.43**	**490.99**

ASHP	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	-	-	0.12	0.04	1.37	2.99	6.40	6.66	3.34	0.39	-	-	**21.31**
HP Supp.	0.01	-	-	-	-	-	-	-	-	-	-	0.01	**0.04**
Space Heat	44.84	26.21	23.91	11.41	2.02	0.47	0.08	0.07	1.63	8.99	30.01	42.83	**192.47**
Hot Water	2.07	1.99	2.38	2.08	2.18	2.08	1.86	2.04	1.80	1.93	1.92	2.00	**24.33**
Vent. Fans	2.38	2.26	2.74	2.38	2.62	2.62	2.38	2.74	2.38	2.50	2.38	2.38	**29.75**
Pumps. Aux.	1.63	1.35	1.35	0.97	0.49	0.16	0.00	0.00	0.20	0.60	1.26	1.60	**9.60**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**68.51**	**48.39**	**50.30**	**34.37**	**27.73**	**27.28**	**28.30**	**31.30**	**26.84**	**32.73**	**53.06**	**66.41**	**495.21**

GSHP	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	-	0.01	0.33	0.42	2.50	4.81	6.82	7.34	3.74	0.71	-	-	**26.67**
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	**0.00**
Space Heat	7.84	4.62	4.22	1.96	0.28	0.05	-	-	0.25	1.56	5.24	7.29	**33.31**
Hot Water	2.08	1.99	2.38	2.08	2.18	2.08	1.86	2.04	1.80	1.93	1.92	2.01	**24.35**
Vent. Fans	2.59	2.46	2.98	2.59	2.85	2.85	2.59	2.98	2.59	2.72	2.59	2.59	**32.42**
Pumps. Aux.	2.10	1.99	2.41	2.09	2.30	2.31	2.10	2.41	2.07	2.17	2.08	2.10	**26.12**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**32.19**	**27.65**	**32.12**	**26.63**	**29.16**	**31.06**	**30.94**	**34.58**	**27.94**	**27.42**	**29.32**	**31.57**	**360.58**

According to the simulation result of the Seattle energy models, the total electricity consumed by the VAV reheat system model is 554.03$\times 10^3 kWh$, highest; the chilled beam system model uses 490.99$\times 10^3 kWh$ of electricity, the model with ASHP system consumes 495.21$\times 10^3 kWh$, and GSHP system consumes 360.58$\times 10^3 kWh$.

The VAV reheat system in Seattle uses more energy for space heating than the other three systems. According to the result, the VAV reheat system consumes 224.28 $\times 10^3$ kWh for space heating, which accounts for 40.5% of the total energy consumption.

Unlike Miami and Phoenix, the chilled beam model in Seattle consumes less energy than VAV reheat system for both cooling and heating from the simulation result, which is consistent with the assumptions for chilled beam systems.

The GSHP system is incredibly efficient in Seattle. The total energy consumption by GSHP model is only 65% of the total energy consumption by VAV reheat system. The temperature in Seattle mild during the year, the heating and cooling demand in Seattle is not too high. Also, from Figure 4.5-1 of chapter 4.5, the temperature difference between the ground layer and the 10 feet deep underground layer in Seattle is about 10℉. The ground source is enough to provide heating and cooling for the Seattle model, as both cooling and heating demand is not high.

6.2.4 Spokane, WA

Figure 6.2-6 shows the monthly energy consumption result by eQUEST models of VAV reheat system, Chilled Beam system, ASHP system and GSHP system. Table 6.2-4 is the specific energy usage data of each component by different energy models.

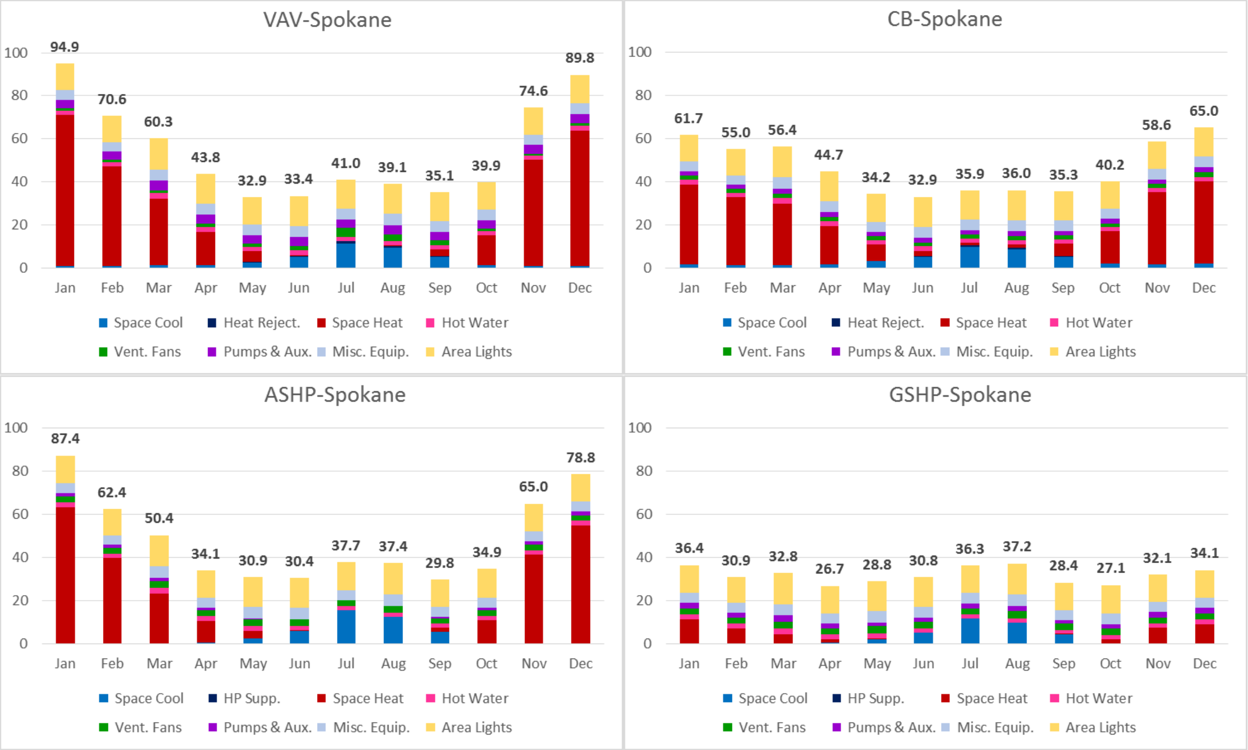


Figure 6.2-6 Monthly Electric Consumption in Spokane (10^3 kWh)

Table 6.2-4 Monthly Electric Consumption Data of Spokane Models (10^3 kWh)

VAV Reheat	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	0.97	0.96	1.17	1.44	2.57	5.10	11.44	9.50	5.05	1.33	1.01	1.06	**41.61**
Heat Reject.	-	-	-	0.02	0.14	0.38	1.06	0.84	0.38	0.02	-	-	**2.84**
Space Heat	69.99	46.06	31.03	15.33	5.05	0.58	0.16	0.20	3.35	13.95	49.10	62.87	**297.69**
Hot Water	2.15	2.18	2.61	2.46	2.13	2.14	1.93	1.93	1.85	1.85	1.97	2.21	**25.41**
Vent. Fans	1.05	1.04	1.26	1.25	1.45	2.00	3.90	3.22	2.10	1.14	1.09	1.15	**20.65**
Pumps. Aux.	3.85	3.76	4.45	4.27	4.00	4.27	4.15	4.30	4.12	4.00	3.97	4.15	**49.32**
Misc. Equip.	4.52	4.45	5.32	5.10	4.72	5.10	4.92	5.12	4.90	4.72	4.70	4.92	**58.49**
Area Lights	12.32	12.12	14.48	13.87	12.86	13.87	13.40	13.94	13.33	12.86	12.79	13.40	**159.23**
Total	**94.85**	**70.57**	**60.32**	**43.75**	**32.93**	**33.44**	**40.96**	**39.05**	**35.07**	**39.88**	**74.64**	**89.76**	**655.23**
Chilled Beam	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**	**Total**
Space Cool	1.60	1.03	1.15	1.73	2.96	5.02	9.62	8.44	5.08	1.82	1.42	1.98	**41.87**
Heat Reject.	0.02	0.01	-	0.04	0.17	0.38	0.85	0.73	0.36	0.05	0.01	0.03	**2.64**
Space Heat	37.03	31.59	28.66	17.47	7.62	2.37	1.12	1.71	5.93	15.21	33.70	37.95	**220.36**
Hot Water	2.15	2.18	2.61	2.46	2.13	2.14	1.93	1.93	1.85	1.85	1.97	2.21	**25.41**
Vent. Fans	2.05	1.70	1.94	1.86	1.71	1.86	1.77	1.85	1.77	1.69	1.92	2.32	**22.43**
Pumps. Aux.	2.03	1.91	2.23	2.14	2.01	2.17	2.24	2.27	2.11	1.99	2.05	2.21	**25.36**
Misc. Equip.	4.52	4.45	5.32	5.10	4.72	5.10	4.92	5.12	4.90	4.72	4.70	4.92	**58.49**
Area Lights	12.32	12.12	14.48	13.87	12.86	13.87	13.40	13.94	13.33	12.86	12.79	13.40	**159.23**
Total	**61.72**	**54.98**	**56.41**	**44.65**	**34.19**	**32.92**	**35.85**	**35.99**	**35.32**	**40.19**	**58.55**	**65.01**	**555.77**

ASHP	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	0.31	2.23	5.97	15.55	12.41	5.36	0.16	-	-	**41.99**
HP Supp.	0.25	0.02	-	-	-	-	-	-	-	-	0.01	0.02	**0.31**
Space Heat	63.02	39.66	23.39	10.18	3.66	0.17	0.04	0.07	2.07	10.56	41.32	54.81	**248.95**
Hot Water	2.25	2.19	2.62	2.28	2.32	2.15	1.86	2.01	1.77	1.94	1.97	2.12	**25.48**
Vent. Fans	2.62	2.49	3.01	2.62	2.88	2.88	2.62	3.01	2.62	2.75	2.62	2.62	**32.73**
Pumps. Aux.	1.66	1.49	1.52	1.22	0.72	0.21	0.08	0.06	0.50	1.12	1.59	1.66	**11.82**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**87.37**	**62.42**	**50.35**	**34.10**	**30.86**	**30.35**	**37.72**	**37.36**	**29.81**	**34.85**	**65.01**	**78.81**	**579.00**

GSHP	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	0.25	1.85	4.87	11.58	9.63	4.23	0.13	-	-	**32.54**
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	**0.00**
Space Heat	11.23	7.04	4.34	1.72	0.64	0.01	-	-	0.33	1.83	7.32	9.04	**43.50**
Hot Water	2.26	2.19	2.62	2.28	2.32	2.15	1.86	2.01	1.77	1.94	1.98	2.13	**25.51**
Vent. Fans	2.86	2.71	3.28	2.86	3.14	3.14	2.86	3.28	2.86	3.00	2.86	2.86	**35.70**
Pumps. Aux.	2.48	2.35	2.78	2.15	1.83	1.66	2.43	2.45	1.69	1.92	2.48	2.48	**26.71**
Misc. Equip.	4.72	4.45	5.32	4.70	5.12	5.10	4.72	5.32	4.70	4.92	4.70	4.72	**58.49**
Area Lights	12.86	12.12	14.48	12.79	13.94	13.87	12.86	14.48	12.79	13.40	12.79	12.86	**159.23**
Total	**36.41**	**30.87**	**32.82**	**26.74**	**28.84**	**30.79**	**36.31**	**37.18**	**28.37**	**27.13**	**32.13**	**34.08**	**381.68**

According to the Spokane models, the total electricity consumed by the VAV reheat system is 655.23× $10^3 kWh$; the chilled beam system model uses 555.77× $10^3 kWh$ of electricity, the model with ASHP system consumes 579.00× $10^3 kWh$, and GSHP system consumes 381.68× $10^3 kWh$.

Similar to the Seattle models, the VAV reheat system in Spokane consumes the largest amount of total energy and the greatest amount of energy for space heating. According to the result, the VAV reheat system consumes 297.69 × 10^3 kWh for space heating, which accounts for 45.4% of the total energy consumption, even more than the Seattle model.

The chilled beam model in Spokane consumes less energy than VAV reheat system for both cooling and heating from the simulation result, which is consistent with a typical chilled beam system.

Although the cooling and heating load in Spokane is larger than that in Seattle, the GSHP system in Spokane is even more efficient than that in Seattle. From Figure 4.5-1 of chapter 4.5, the temperature difference between the ground layer and the 10 feet deep underground layer in Seattle is about 15℉, the largest among the four cities.

6.2.5 Summary

Figure 6.2-7 and Table 6.2-5 summarize the total electricity consumptions of the sixteen energy models.

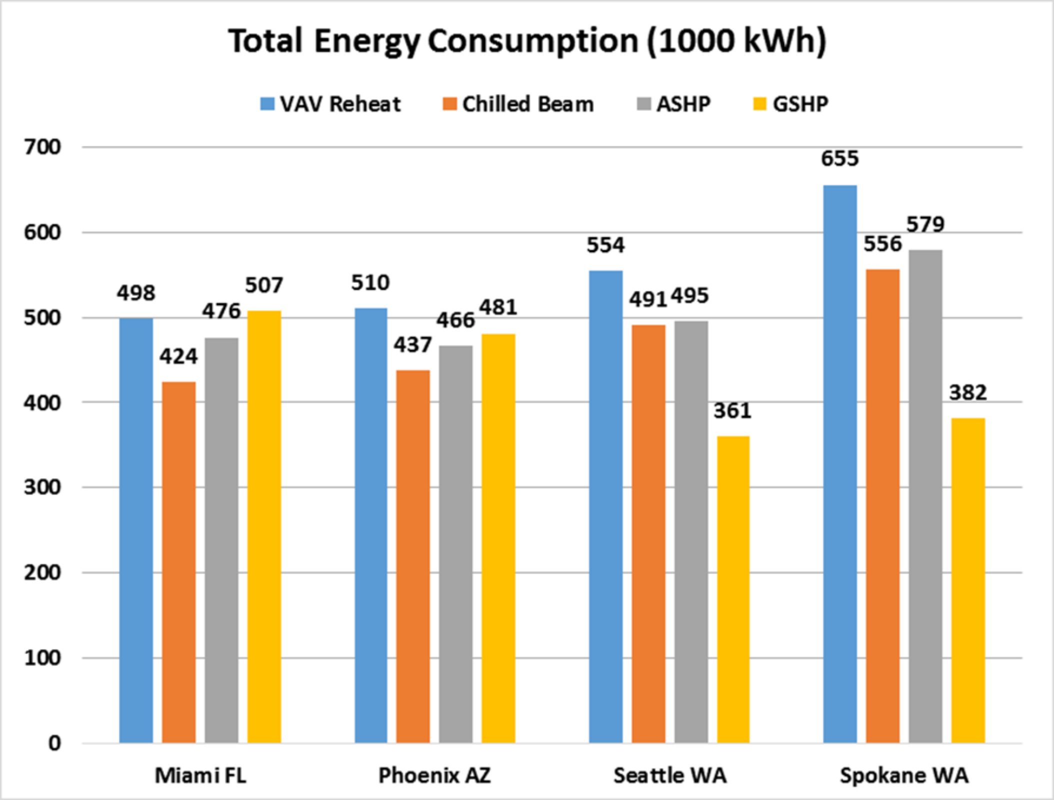


Figure 6.2-7 Comparison of Total Electricity Consumption (10^3 kWh)

Table 6.2-5 Total Electricity Consumption (10^3 kWh)

	Miami	Phoenix	Seattle	Spokane
VAV Reheat	498.4	510.4	554.0	655.2
Chilled Beam	424.3	437.5	491.0	555.8
ASHP	476.3	466.3	495.2	579.0
GSHP	507.0	480.8	360.6	381.7

Previous research shows that chilled beams can be simulated in eQUEST is with an Induction Units (IU) system. (Betz, F., McNeill, J., Talbert, B., Thimmanna, H., and Repka, N., 2012) The eQUEST program does not have an exactly chilled beam component, (Vaughn, 2012) simulation of chilled beam models in this research adopt the IU system. Induction units are the most similar in concept to chilled beams.

In Miami, GSHP system has the highest energy consumption, follows by the VAV reheat system, the Chilled Beam system consumes the lowest energy, and the ASHP is the second lowest system.

In Phoenix, even though it is a hot climate zone, due to the constancy of the underground temperature, the GSHP system is more efficient than in Miami. The VAV reheat system consumes the highest energy, follows the GSHP system. Similar to Miami, the Chilled Beam system is the most energy-efficient one; the ASHP is the second most efficient system.

The systems in Seattle and Spokane have exact the same sequence. The systems in Spokane consume more energy than those in Seattle, due to the higher heating and cooling load demand. The VAV reheat system consumes the highest energy in both Seattle and Spokane, follows by the ASHP system and then the Chilled Beam system. The GSHP system is the most energy efficient system, which uses 193,450 kWh less energy than the VAV reheat system in Seattle and 273,550 kWh less energy than the VAV reheat system in Spokane.

CHAPTER 7. LIFE-CYCLE COST ANALYSIS

This study is to compare the Life-Cycle Cost of VAV system, chilled beam system, air source heat pump system, ground source heat pump system in four different climate zones through a Life-Cycle Cost Analysis. The Life-Cycle Cost includes the system capital cost, energy cost, system maintenance and replacement cost over a 20-year of life span.

7.1 CAPITAL COST

The capital cost of HVAC system varies with system components, locations, building type, building size, and year. To get the potential ranges of the capital costs for the systems, the following examples have been collected. Table 7.1-1 is the square foot costs of different HVAC system from the references.

Table 7.1-1 Capital Costs Data

System Description	Location	Square Foot Cost	Year	Data Source
VAV	Palo Alto, CA	$23.03/ ft^2	2005	Davis et al, 2005
VAV	National	$23.23/ ft^2	2007	Smeed, 2007
VAV	Cincinnati, OH	$25.00/ ft^2	2009	Feldkamp, 2009
VAV + DX cooling	National	$14.70/ ft^2	2012	Ihnen et al, 2012
VAV + boiler + chiller	National	$18.58/ ft^2	2012	Ihnen et al, 2012
VAV Reheat	National	$16.25/ ft^2	2013	Park, 2013
VAV Reheat	Sacramento, CA	$25.00/ ft^2	2013	Stein, J. & Taylor, 2013
VAV Reheat	Sacramento, CA	$28.00/ ft^2	2014	Taylor Engineering, 2014
Packaged VAV	New York, NY	$11.20/ ft^2	2015	New York City MOS, 2015
Chilled Beam	National	$39.95/ ft^2	2007	Smeed, 2007
Chilled Beam	Cincinnati, OH	$19.50/ ft^2	2009	Feldkamp, 2009
Chilled Beam	Sacramento, CA	$62.00/ ft^2	2013	Stein, J. & Taylor, 2013
ASHP	Sioux City, Iowa	$14.58/ ft^2	2006	Chiasson, 2006
ASHP	National	$16.81/ ft^2	2012	Ihnen et al, 2012
ASHP	National	$24.50/ ft^2	2013	Apprill, 2013
GSHP	Centralia, IL	$26.10/ ft^2	2010	Kavanaugh et al, 2012
GSHP	National	$20.75/ ft^2	2011	Kavanaugh et al, 2012
GSHP	National	$25.73/ ft^2	2012	Ihnen et al, 2012
GSHP	New York, NY	$30.85/ ft^2	2015	New York City MOS, 2015

Life-Cycle Cost analysis is to compare the difference between the alternatives, based on the data above, calculated averages of those costs are applied to the Life-Cycle Cost Analysis. The data for specific locations can be converted to a national value through Equation 7.1-1 (R.S. Means Company, 2016), based on the City Cost Index (total) for HVAC system (R.S. Means Company, 2016), shows in Table 7.1-2.

Table 7.1-2 City Cost Index - Division 23: HVAC (R.S. Means Company, 2016)

	Material	Installation	Total
Palo Alto, CA	96.7	144.1	116.8
Cincinnati, OH	100.0	83.1	92.8
Sacramento, CA	99.9	120.1	108.4
New York, NY	100.1	168.3	128.9
Sioux City, IA	100.1	75.0	89.5
Centralia, IL	96.5	94.2	95.6
Miami, FL	100.0	60.0	83.1
Phoenix, AZ	100.0	79.2	91.2
Seattle, WA	100.1	114.2	106.0
Spokane, WA	100.6	84.0	93.6

$$\boldsymbol{National\ Average\ Costs\ (total)} = \frac{\mathbf{100}}{\boldsymbol{City\ Cost\ Index\ (total)}} \times \boldsymbol{City\ Cost} \qquad \text{Equation 7.1.1}$$

The capital costs in Table 7.1-2 are collected from 2005 to 2015, which need to be converted to the current estimated price through the historical cost Indexes. Table 7.1-3 lists the RSMeans historical cost Indexes. (R.S. Means Company, 2016)

Table 7.1-3 RS Means Historical Cost Index Jan.1, 1993=100 (R.S. Means Company, 2016)

Year	2016	2015	2014	2013	2012	2011	2010	2009	2008	2007	2006	2005
Estimated Cost Index	207.2											
Actual Cost Index		206.2	204.9	201.2	194.6	191.2	183.5	180.1	180.4	169.4	162.0	151.2

$$\boldsymbol{Cost\ of\ Year\ 2016} = \frac{\boldsymbol{Index\ for\ Year\ 2016}}{\boldsymbol{Index\ for\ Year\ N}} \times \boldsymbol{Cost\ of\ Year\ N} \qquad \text{Equation 7.1.2}$$

With the Equation 7.1.2 (R.S. Means Company, 2016), the estimated cost in the year of 2016 is available based on the indexes and the actual cost of the specific year derived from the references.

Table 7.1-4 shows the converted national costs in the year of 2016, and Table 7.1-5 is the calculated average data.

Table 7.1-4 Converted National Cost

System Description	Location	Square Foot Cost	City Index	National Cost	Data Year	Cost Index	National Cost in 2016
VAV	Palo Alto, CA	\$23.03/ ft^2	116.8	\$19.72/ ft^2	2005	151.2	\$27.02/ ft^2
VAV	National	\$23.23/ ft^2	\	\$23.23/ ft^2	2007	169.4	\$28.41/ ft^2
VAV	Cincinnati, OH	\$25.00/ ft^2	92.8	\$26.94/ ft^2	2009	180.1	\$30.99/ ft^2
VAV DX cooling	National	\$14.70/ ft^2	\	\$14.70/ ft^2	2012	194.6	\$15.65/ ft^2
VAV boiler, chiller	National	\$18.58/ ft^2	\	\$18.58/ ft^2	2012	194.6	\$19.78/ ft^2
VAV Reheat	National	\$16.25/ ft^2	\	\$16.25/ ft^2	2013	201.2	\$16.73/ ft^2
VAV Reheat	Sacramento, CA	\$25.00/ ft^2	108.4	\$23.06/ ft^2	2013	201.2	\$23.75/ ft^2
VAV Reheat	Sacramento, CA	\$28.00/ ft^2	108.4	\$25.83/ ft^2	2014	204.9	\$26.12/ ft^2
Packaged VAV	New York, NY	\$11.20/ ft^2	128.9	\$ 8.69/ ft^2	2015	206.2	\$ 8.73/ ft^2
Chilled Beam	National	\$39.95/ ft^2	\	\$39.95/ ft^2	2007	169.4	\$48.86/ ft^2
Chilled Beam	Cincinnati, OH	\$19.50/ ft^2	92.8	\$21.01/ ft^2	2009	180.1	\$24.17/ ft^2
Chilled Beam	Sacramento, CA	\$62.00/ ft^2	108.4	\$57.20/ ft^2	2013	201.2	\$58.90/ ft^2
`ASHP	Sioux City, Iowa	\$14.58/ ft^2	89.5	\$16.29/ ft^2	2006	162.0	\$20.84/ ft^2
ASHP	National	\$16.81/ ft^2	\	\$16.81/ ft^2	2012	194.6	\$17.90/ ft^2
ASHP	National	\$24.50/ ft^2	\	\$24.50/ ft^2	2013	201.2	\$25.23/ ft^2
GSHP	Centralia, IL	\$26.10/ ft^2	95.6	\$27.30/ ft^2	2010	183.5	\$30.83/ ft^2
GSHP	National	\$20.75/ ft^2	\	\$20.75/ ft^2	2011	191.2	\$22.49/ ft^2
GSHP	National	\$25.73/ ft^2	\	\$25.73/ ft^2	2012	194.6	\$27.40/ ft^2
GSHP	New York, NY	\$30.85/ ft^2	128.9	\$23.93/ ft^2	2015	206.2	\$24.05/ ft^2

Table 7.1-5 Calculated National Average Cost

System	National Average Cost
VAV Reheat	\$21.91/ ft^2
Chilled Beam	\$43.98/ ft^2
ASHP	\$21.32/ ft^2
GSHP	\$26.19/ ft^2

Using the estimated national average cost of the systems (Table 7.1-5) and the city cost index for HVAC system (Table 7.1-2), we can calculate the system capital cost of each city through Equation 7.1.3 (R.S. Means Company, 2016), shows in Table 7.1-6.

$$\boldsymbol{City\ Cost} = \frac{\boldsymbol{City\ Cost\ Index}}{\mathbf{100}} \times \boldsymbol{National\ Average\ Costs} \qquad \text{Equation 7.1.3}$$

Table 7.1-6 System Capital Cost by City

System		Miami, FL	Phoenix, AZ	Seattle, WA	Spokane, WA
VAV Reheat	Unit cost	$18.21/ ft^2	$19.98/ ft^2	$23.23/ ft^2	$20.51/ ft^2
	Total	$977,476.75	$1,072,754.26	$1,246,841.58	$1,100,984.64
Chilled Beam	Unit cost	$36.55/ ft^2	$40.11/ ft^2	$46.62/ ft^2	$41.17/ ft^2
	Total	$1,962,016.33	$2,153,259.80	$2,502,692.31	$2,209,924.53
ASHP	Unit cost	$17.72/ ft^2	$19.45/ ft^2	$22.60/ ft^2	$19.96/ ft^2
	Total	$951,186.33	$1,043,901.24	$1,213,306.27	$1,071,372.33
GSHP	Unit cost	$21.76/ ft^2	$23.89/ ft^2	$27.76/ ft^2	$24.51/ ft^2
	Total	$1,168,362.41	$1,282,246.11	$1,490,329.91	$1,315,989.43

7.2 ENERGY COST

Table 7.2-1 shows electricity price of each city. (Electricity Local, 2017) Table 7.2-2 shows the annual electricity costs based on the annual electricity consumption of each system.

Table 7.2-1 Electricity Price

	Miami, FL	Phoenix, AZ	Seattle, WA	Spokane, WA
Commercial	8.72 ¢/kWh	10.22 ¢/kWh	6.80 ¢/kWh	5.62 ¢/kWh

Table 7.2-2 Annual Electricity Cost

Systems		Miami, FL	Phoenix, AZ	Seattle, WA	Spokane, WA
VAV Reheat	Energy Consume	498,400 kWh/ yr	510,400 kWh/ yr	554,000 kWh/ yr	655,200 kWh/ yr
	Total Energy Cost	$43,460.48/ yr	$52,162.88/ yr	$37,672.00/ yr	$36,822.24/ yr
Chilled Beam	Energy Consume	424300 kWh/ yr	437,500 kWh/ yr	491,000 kWh/ yr	555,800 kWh/ yr
	Total Energy Cost	$36,998.96/ yr	$44,712.50/ yr	$33,388.00/ yr	$31,235.96/ yr
ASHP	Energy Consume	476300 kWh/ yr	466,300 kWh/ yr	495,200 kWh/ yr	579,000 kWh/ yr
	Total Energy Cost	$41,533.36/ yr	$47,655.86/ yr	$33,673.60/ yr	$32,539.80/ yr
GSHP	Energy Consume	507000 kWh/ yr	480,800 kWh/ yr	360,600 kWh/ yr	381,700 kWh/ yr
	Total Energy Cost	$44,210.40/ yr	$49,137.76/ yr	$24,520.80/ yr	$21,451.54/ yr

7.3 MAINTENANCE COST

Maintenance costs contain the planned equipment maintenance, such as cleaning and repair.

The annual maintenance costs of HVAC systems used in the analysis were from a paper by Bloomquist, 2001, as Figure 7.3-1 shows.

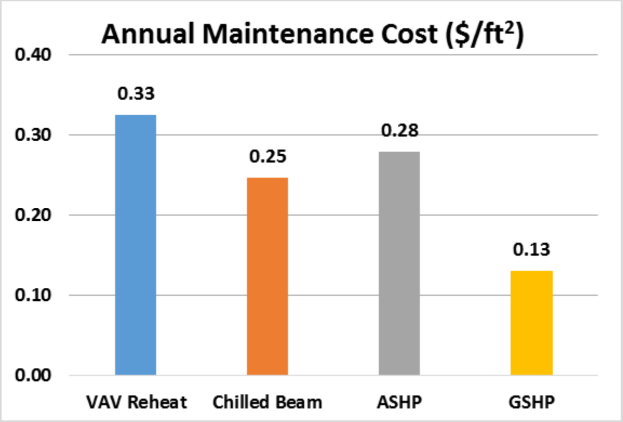


Figure 7.3-1 HVAC Annual Maintenance Cost by Systems (Bloomquist, 2001)

Using those data, we can calculate the annual maintenance costs for specific locations (Miami, Phoenix, Seattle and Spokane) in 2016 through the city indexes and the historical index as Equation 7.1.2 and Equation 7.1.3, shown in Table 7.3-1. The historical Index for the year 2001 is 125.1 based on Jan.1, 1993=100 (R.S. Means Company, 2016)

Table 7.3-1 Annual Maintenance Cost

Maintenance Cost	Miami, FL	Phoenix, AZ	Seattle, WA	Spokane, WA
VAV Reheat	$24,025.70	$26,367.56	$30,646.51	$27,061.44
Chilled Beam	$18,190.89	$19,964.01	$23,203.78	$20,489.38
ASHP	$20,593.46	$22,600.77	$26,268.43	$23,195.52
GSHP	$9,610.28	$10,547.02	$12,258.60	$10,824.58

7.4 LIFE-CYCLE COST ANALYSIS

The HVAC systems capital costs range from $951,186.33 to $2,502,692.31 based on the system types and the locations, with an expected life of 20 years.

The annual energy consumptions are derived from the energy models through the eQUEST simulation, then multiply by the regional electricity prices to get the and the annual energy costs.

The annual electric costs are calculated with the inflation rate of 3% per year with an additional commercial fuel price index according to Figure 7.4-1. (Lavappa and Kneifel, 2016)

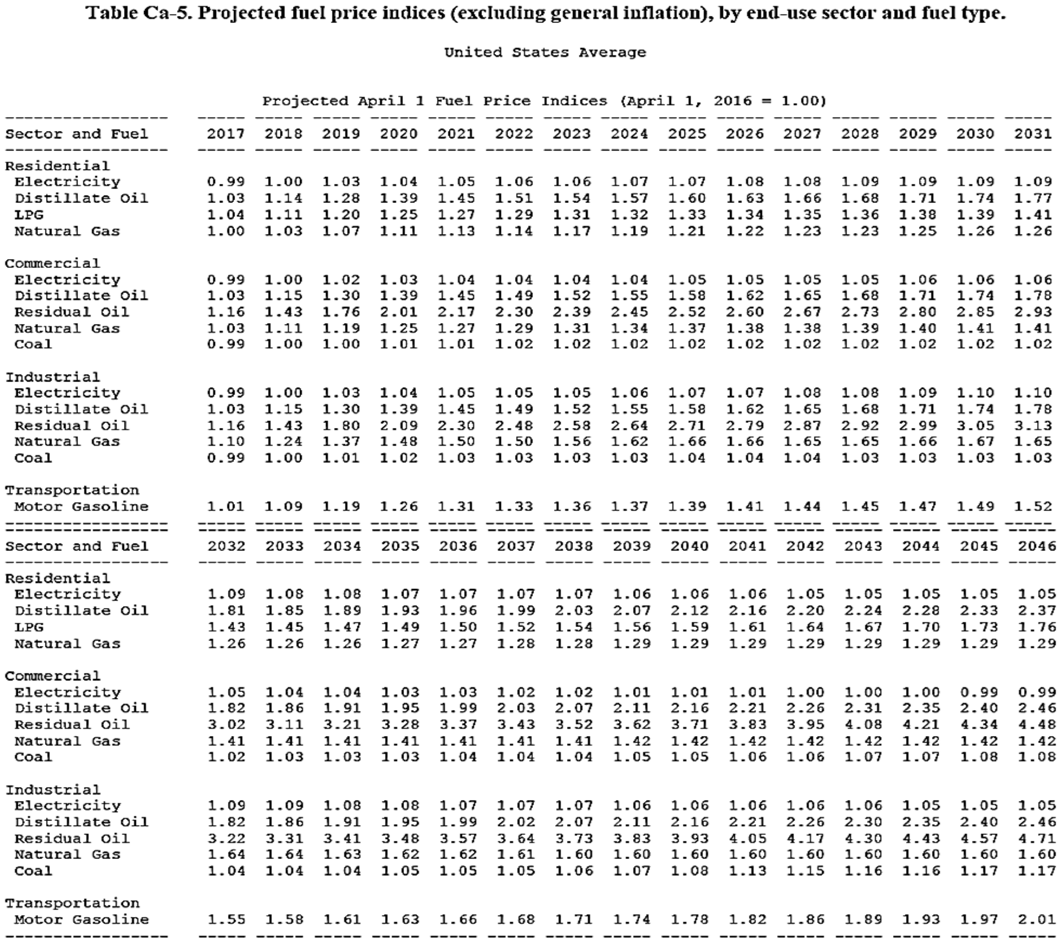


Table Ca-5. Projected fuel price indices (excluding general inflation), by end-use sector and fuel type.

United States Average

Projected April 1 Fuel Price Indices (April 1, 2016 = 1.00)

Sector and Fuel	2017	2018	2019	2020	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030	2031
Residential															
Electricity	0.99	1.00	1.03	1.04	1.05	1.06	1.06	1.07	1.07	1.08	1.08	1.09	1.09	1.09	1.09
Distillate Oil	1.03	1.14	1.28	1.39	1.45	1.51	1.54	1.57	1.60	1.63	1.66	1.68	1.71	1.74	1.77
LPG	1.04	1.11	1.20	1.25	1.27	1.29	1.31	1.32	1.33	1.34	1.35	1.36	1.38	1.39	1.41
Natural Gas	1.00	1.03	1.07	1.11	1.13	1.14	1.17	1.19	1.21	1.22	1.23	1.23	1.25	1.26	1.26
Commercial															
Electricity	0.99	1.00	1.02	1.03	1.04	1.04	1.04	1.04	1.05	1.05	1.05	1.05	1.06	1.06	1.06
Distillate Oil	1.03	1.15	1.30	1.39	1.45	1.49	1.52	1.55	1.58	1.62	1.65	1.68	1.71	1.74	1.78
Residual Oil	1.16	1.43	1.76	2.01	2.17	2.30	2.39	2.45	2.52	2.60	2.67	2.73	2.80	2.85	2.93
Natural Gas	1.03	1.11	1.19	1.25	1.27	1.29	1.31	1.34	1.37	1.38	1.38	1.39	1.40	1.41	1.41
Coal	0.99	1.00	1.00	1.01	1.01	1.02	1.02	1.02	1.02	1.02	1.02	1.02	1.02	1.02	1.02
Industrial															
Electricity	0.99	1.00	1.03	1.04	1.05	1.05	1.05	1.06	1.07	1.07	1.08	1.08	1.09	1.10	1.10
Distillate Oil	1.03	1.15	1.30	1.39	1.45	1.49	1.52	1.55	1.58	1.62	1.65	1.68	1.71	1.74	1.78
Residual Oil	1.16	1.43	1.80	2.09	2.30	2.48	2.58	2.64	2.71	2.79	2.87	2.92	2.99	3.05	3.13
Natural Gas	1.10	1.24	1.37	1.48	1.50	1.50	1.56	1.62	1.66	1.66	1.65	1.65	1.66	1.67	1.65
Coal	0.99	1.00	1.01	1.02	1.03	1.03	1.03	1.03	1.04	1.04	1.04	1.03	1.03	1.03	1.03
Transportation															
Motor Gasoline	1.01	1.09	1.19	1.26	1.31	1.33	1.36	1.37	1.39	1.41	1.44	1.45	1.47	1.49	1.52

Sector and Fuel	2032	2033	2034	2035	2036	2037	2038	2039	2040	2041	2042	2043	2044	2045	2046
Residential															
Electricity	1.09	1.08	1.08	1.07	1.07	1.07	1.07	1.06	1.06	1.06	1.05	1.05	1.05	1.05	1.05
Distillate Oil	1.81	1.85	1.89	1.93	1.96	1.99	2.03	2.07	2.12	2.16	2.20	2.24	2.28	2.33	2.37
LPG	1.43	1.45	1.47	1.49	1.50	1.52	1.54	1.56	1.59	1.61	1.64	1.67	1.70	1.73	1.76
Natural Gas	1.26	1.26	1.26	1.27	1.27	1.28	1.28	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29
Commercial															
Electricity	1.05	1.04	1.04	1.03	1.03	1.02	1.02	1.01	1.01	1.01	1.00	1.00	1.00	0.99	0.99
Distillate Oil	1.82	1.86	1.91	1.95	1.99	2.03	2.07	2.11	2.16	2.21	2.26	2.31	2.35	2.40	2.46
Residual Oil	3.02	3.11	3.21	3.28	3.37	3.43	3.52	3.62	3.71	3.83	3.95	4.08	4.21	4.34	4.48
Natural Gas	1.41	1.41	1.41	1.41	1.41	1.41	1.41	1.42	1.42	1.42	1.42	1.42	1.42	1.42	1.42
Coal	1.02	1.03	1.03	1.03	1.04	1.04	1.04	1.05	1.05	1.06	1.06	1.07	1.07	1.08	1.08
Industrial															
Electricity	1.09	1.09	1.08	1.08	1.07	1.07	1.07	1.06	1.06	1.06	1.06	1.06	1.05	1.05	1.05
Distillate Oil	1.82	1.86	1.91	1.95	1.99	2.02	2.07	2.11	2.16	2.21	2.26	2.30	2.35	2.40	2.46
Residual Oil	3.22	3.31	3.41	3.48	3.57	3.64	3.73	3.83	3.93	4.05	4.17	4.30	4.43	4.57	4.71
Natural Gas	1.64	1.64	1.63	1.62	1.62	1.61	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60
Coal	1.04	1.04	1.04	1.05	1.05	1.05	1.06	1.07	1.08	1.13	1.15	1.16	1.16	1.17	1.17
Transportation															
Motor Gasoline	1.55	1.58	1.61	1.63	1.66	1.68	1.71	1.74	1.78	1.82	1.86	1.89	1.93	1.97	2.01

Figure 7.4-1 Fuel Price Index (Lavappa and Kneifel, 2016)

The maintenance costs include planned equipment cleaning, repair and replacement counted by $/s.f./year, as listed in Table 7.3-1. Annual costs for maintenance will keep pace with the inflation rate of 3% annually over the study period. This research uses a discount rate of 3% in the Present value calculation.

Table 7.4-1 ~ Table 7.4-16 exhibit the Life-Cycle Cost Analysis of the four systems in the four climate zones (Miami, Phoenix, Seattle, and Spokane).

Table 7.4-1 Life-Cycle Cost Analysis of the VAV Reheat System in Miami, FL

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$977,476.75										
Energy costs		$44,319.32	$46,110.00	$48,443.16	$50,385.64	$52,401.07	$53,973.10	$55,592.29	$57,260.06	$59,544.96	$61,331.30
Maintenance costs		$24,746.47	$25,488.87	$26,253.54	$27,041.14	$27,852.38	$28,687.95	$29,548.58	$30,435.04	$31,348.09	$32,288.54
Net annual cash flow	$977,476.75	$69,065.79	$71,598.87	$74,696.70	$77,426.78	$80,253.44	$82,661.04	$85,140.88	$87,695.10	$90,893.05	$93,619.84
Present value of cash flow	$977,476.75	$67,054.17	$67,488.80	$68,358.06	$68,792.69	$69,227.32	$69,227.32	$69,227.32	$69,227.32	$69,661.95	$69,661.95
Accumulate cash flow	$977,476.75	$1,046,542.54	$1,118,141.41	$1,192,838.11	$1,270,264.89	$1,350,518.33	$1,433,179.37	$1,518,320.25	$1,606,015.35	$1,696,908.40	$1,790,528.24
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$63,171.24	$65,066.38	$67,656.64	$69,686.34	$71,776.93	$73,232.79	$74,711.39	$76,952.73	$78,499.19	$80,854.16
Maintenance costs		$33,257.19	$34,254.91	$35,282.56	$36,341.03	$37,431.26	$38,554.20	$39,710.83	$40,902.15	$42,129.22	$43,393.09
Net annual cash flow		$96,428.44	$99,321.29	$102,939.20	$106,027.37	$109,208.20	$111,786.99	$114,422.22	$117,854.88	$120,628.40	$124,247.25
Present value of cash flow		$69,661.95	$69,661.95	$70,096.59	$70,096.59	$70,096.59	$69,661.95	$69,227.32	$69,227.32	$68,792.69	$68,792.69
Accumulate cash flow		$1,886,956.68	$1,986,277.97	$2,089,217.17	$2,195,244.54	$2,304,452.74	$2,416,239.73	$2,530,661.94	$2,648,516.83	$2,769,145.23	$2,893,392.49

Table 7.4-2 Life-Cycle Cost Analysis of the Chilled Beam System in Miami, FL

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$1,962,016.33										
Energy costs		$37,729.62	$39,254.05	$41,240.30	$42,893.96	$44,609.72	$45,948.01	$47,326.45	$48,746.24	$50,691.40	$52,212.14
Maintenance costs		$18,736.62	$19,298.72	$19,877.68	$20,474.01	$21,088.23	$21,720.87	$22,372.50	$23,043.68	$23,734.99	$24,447.03
Net annual cash flow	$1,962,016.33	$56,466.23	$58,552.76	$61,117.98	$63,367.96	$65,697.94	$67,668.88	$69,698.95	$71,789.92	$74,426.39	$76,659.18
Present value of cash flow	$1,962,016.33	$54,821.59	$55,191.59	$55,931.61	$56,301.62	$56,671.62	$56,671.62	$56,671.62	$56,671.62	$57,041.63	$57,041.63
Accumulate cash flow	$1,962,016.33	$2,018,482.56	$2,077,035.33	$2,138,153.30	$2,201,521.27	$2,267,219.21	$2,334,888.09	$2,404,587.04	$2,476,376.95	$2,550,803.34	$2,627,462.52
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$53,778.51	$55,391.86	$57,596.99	$59,324.90	$61,104.64	$62,344.03	$63,602.79	$65,510.87	$66,827.39	$68,832.21
Maintenance costs		$25,180.45	$25,935.86	$26,713.94	$27,515.35	$28,340.81	$29,191.04	$30,066.77	$30,968.77	$31,897.84	$32,854.77
Net annual cash flow		$78,958.96	$81,327.72	$84,310.92	$86,840.25	$89,445.46	$91,535.07	$93,669.56	$96,479.64	$98,725.22	$101,686.98
Present value of cash flow		$57,041.63	$57,041.63	$57,411.64	$57,411.64	$57,411.64	$57,041.63	$56,671.62	$56,671.62	$56,301.62	$56,301.62
Accumulate cash flow		$2,706,421.48	$2,787,749.20	$2,872,060.12	$2,958,900.37	$3,048,345.83	$3,139,880.90	$3,233,550.46	$3,330,030.10	$3,428,755.32	$3,530,442.30

Table 7.4-3 Life-Cycle Cost Analysis of the ASHP System in Miami, FL

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$951,186.33										
Energy costs		$42,347.12	$44,058.12	$46,287.46	$48,143.49	$50,069.23	$51,571.31	$53,118.45	$54,712.00	$56,895.22	$58,602.08
Maintenance costs		$21,211.26	$21,847.60	$22,503.03	$23,178.12	$23,873.46	$24,589.67	$25,327.36	$26,087.18	$26,869.79	$27,675.89
Net annual cash flow	$951,186.33	$63,558.39	$65,905.72	$68,790.49	$71,321.61	$73,942.70	$76,160.98	$78,445.81	$80,799.18	$83,765.02	$86,277.97
Present value of cash flow	$951,186.33	$61,707.17	$62,122.46	$62,953.04	$63,368.33	$63,783.62	$63,783.62	$63,783.62	$63,783.62	$64,198.91	$64,198.91
Accumulate cash flow	$951,186.33	$1,014,744.71	$1,080,650.43	$1,149,440.92	$1,220,762.53	$1,294,705.23	$1,370,866.21	$1,449,312.02	$1,530,111.20	$1,613,876.22	$1,700,154.18
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$60,360.14	$62,170.95	$64,645.94	$66,585.32	$68,582.88	$69,973.95	$71,386.75	$73,528.36	$75,005.99	$77,256.17
Maintenance costs		$28,506.17	$29,361.35	$30,242.19	$31,149.46	$32,083.94	$33,046.46	$34,037.85	$35,058.99	$36,110.76	$37,194.08
Net annual cash flow		$88,866.31	$91,532.30	$94,888.13	$97,734.78	$100,666.82	$103,020.40	$105,424.61	$108,587.34	$111,116.75	$114,450.25
Present value of cash flow		$64,198.91	$64,198.91	$64,614.20	$64,614.20	$64,614.20	$64,198.91	$63,783.62	$63,783.62	$63,368.33	$63,368.33
Accumulate cash flow		$1,789,020.49	$1,880,552.78	$1,975,440.91	$2,073,175.69	$2,173,842.51	$2,276,862.91	$2,382,287.52	$2,490,874.86	$2,601,991.61	$2,716,441.87

Table 7.4-4 Life-Cycle Cost Analysis of the GSHP System in Miami, FL

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,168,362.41										
Energy costs		$45,081.34	$46,902.81	$49,276.10	$51,251.97	$53,302.05	$54,901.11	$56,548.14	$58,244.59	$60,568.77	$62,385.83
Maintenance costs		$9,898.59	$10,195.55	$10,501.41	$10,816.46	$11,140.95	$11,475.18	$11,819.43	$12,174.02	$12,539.24	$12,915.41
Net annual cash flow	$1,168,362.41	$54,979.93	$57,098.36	$59,777.51	$62,068.43	$64,443.00	$66,376.29	$68,367.58	$70,418.61	$73,108.01	$75,301.25
Present value of cash flow	$1,168,362.41	$53,378.58	$53,820.68	$54,704.89	$55,146.99	$55,589.10	$55,589.10	$55,589.10	$55,589.10	$56,031.20	$56,031.20
Accumulate cash flow	$1,168,362.41	$1,223,342.35	$1,280,440.71	$1,340,218.22	$1,402,286.64	$1,466,729.64	$1,533,105.93	$1,601,473.51	$1,671,892.12	$1,745,000.13	$1,820,301.38
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$64,257.41	$66,185.13	$68,819.93	$70,884.53	$73,011.06	$74,491.95	$75,995.98	$78,275.86	$79,848.90	$82,244.37
Maintenance costs		$13,302.88	$13,701.96	$14,113.02	$14,536.41	$14,972.51	$15,421.68	$15,884.33	$16,360.86	$16,851.69	$17,357.24
Net annual cash flow		$77,560.29	$79,887.10	$82,932.95	$85,420.94	$87,983.57	$89,913.63	$91,880.31	$94,636.72	$96,700.59	$99,601.60
Present value of cash flow		$56,031.20	$56,031.20	$56,473.31	$56,473.31	$56,473.31	$56,031.20	$55,589.10	$55,589.10	$55,146.99	$55,146.99
Accumulate cash flow		$1,897,861.66	$1,977,748.76	$2,060,681.71	$2,146,102.65	$2,234,086.22	$2,323,999.85	$2,415,880.16	$2,510,516.88	$2,607,217.46	$2,706,819.07

Table 7.4-5 Life-Cycle Cost Analysis of the VAV Reheat System in Phoenix, AZ

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,072,754.26										
Energy costs		$53,194.66	$55,343.94	$58,144.34	$60,475.81	$62,894.85	$64,781.69	$66,725.14	$68,726.90	$71,469.36	$73,613.44
Maintenance costs		$27,158.59	$27,973.34	$28,812.54	$29,676.92	$30,567.23	$31,484.24	$32,428.77	$33,401.64	$34,403.68	$35,435.79
Net annual cash flow	$1,072,754.26	$80,353.24	$83,317.28	$86,956.88	$90,152.73	$93,462.07	$96,265.94	$99,153.91	$102,128.53	$105,873.05	$109,049.24
Present value of cash flow	$1,072,754.26	$78,012.86	$78,534.53	$79,577.87	$80,099.54	$80,621.21	$80,621.21	$80,621.21	$80,621.21	$81,142.88	$81,142.88
Accumulate cash flow	$1,072,754.26	$1,153,107.51	$1,236,424.79	$1,323,381.67	$1,413,534.40	$1,506,996.48	$1,603,262.41	$1,702,416.33	$1,804,544.86	$1,910,417.91	$2,019,467.15
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$75,821.85	$78,096.50	$81,205.49	$83,641.65	$86,150.90	$87,898.30	$89,673.01	$92,363.20	$94,219.35	$97,045.93
Maintenance costs		$36,498.87	$37,593.83	$38,721.65	$39,883.30	$41,079.80	$42,312.19	$43,581.56	$44,889.00	$46,235.67	$47,622.74
Net annual cash flow		$112,320.72	$115,690.34	$119,927.14	$123,524.95	$127,230.70	$130,210.49	$133,254.57	$137,252.21	$140,455.02	$144,668.67
Present value of cash flow		$81,142.88	$81,142.88	$81,664.55	$81,664.55	$81,664.55	$81,142.88	$80,621.21	$80,621.21	$80,099.54	$80,099.54
Accumulate cash flow		$2,131,787.86	$2,247,478.20	$2,367,405.34	$2,490,930.29	$2,618,160.99	$2,748,371.49	$2,881,626.06	$3,018,878.26	$3,159,333.28	$3,304,001.96

Table 7.4-6 Life-Cycle Cost Analysis of the Chilled Beam System in Phoenix, AZ

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$2,153,259.80										
Energy costs		$45,589.17	$47,431.15	$49,831.17	$51,829.30	$53,902.48	$55,519.55	$57,185.14	$58,900.69	$61,251.05	$63,088.59
Maintenance costs		$20,562.93	$21,179.82	$21,815.21	$22,469.67	$23,143.76	$23,838.07	$24,553.21	$25,289.81	$26,048.50	$26,829.96
Net annual cash flow	$2,153,259.80	$66,152.10	$68,610.97	$71,646.38	$74,298.97	$77,046.23	$79,357.62	$81,738.35	$84,190.50	$87,299.56	$89,918.54
Present value of cash flow	$2,153,259.80	$64,225.34	$64,672.42	$65,566.59	$66,013.67	$66,460.76	$66,460.76	$66,460.76	$66,460.76	$66,907.84	$66,907.84
Accumulate cash flow	$2,153,259.80	$2,219,411.90	$2,288,022.87	$2,359,669.25	$2,433,968.22	$2,511,014.45	$2,590,372.07	$2,672,110.42	$2,756,300.92	$2,843,600.48	$2,933,519.02
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$64,981.24	$66,930.68	$69,595.16	$71,683.01	$73,833.50	$75,331.07	$76,852.04	$79,157.60	$80,748.37	$83,170.82
Maintenance costs		$27,634.86	$28,463.90	$29,317.82	$30,197.36	$31,103.28	$32,036.37	$32,997.47	$33,987.39	$35,007.01	$36,057.22
Net annual cash flow		$92,616.10	$95,394.58	$98,912.98	$101,880.37	$104,936.78	$107,367.44	$109,849.51	$113,144.99	$115,755.38	$119,228.04
Present value of cash flow		$66,907.84	$66,907.84	$67,354.93	$67,354.93	$67,354.93	$66,907.84	$66,460.76	$66,460.76	$66,013.67	$66,013.67
Accumulate cash flow		$3,026,135.13	$3,121,529.71	$3,220,442.69	$3,322,323.06	$3,427,259.84	$3,534,627.28	$3,644,476.79	$3,757,621.78	$3,873,377.15	$3,992,605.19

Table 7.4-7 Life-Cycle Cost Analysis of the ASHP System in Phoenix, AZ

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,043,901.24										
Energy costs		$48,593.64	$50,557.02	$53,115.20	$55,245.02	$57,454.82	$59,178.46	$60,953.82	$62,782.43	$65,287.69	$67,246.32
Maintenance costs		$23,278.79	$23,977.15	$24,696.47	$25,437.36	$26,200.48	$26,986.50	$27,796.09	$28,629.97	$29,488.87	$30,373.54
Net annual cash flow	$1,043,901.24	$71,872.43	$74,534.17	$77,811.67	$80,682.38	$83,655.30	$86,164.96	$88,749.91	$91,412.40	$94,776.56	$97,619.86
Present value of cash flow	$1,043,901.24	$69,779.05	$70,255.60	$71,208.70	$71,685.25	$72,161.80	$72,161.80	$72,161.80	$72,161.80	$72,638.34	$72,638.34
Accumulate cash flow	$1,043,901.24	$1,115,773.67	$1,190,307.84	$1,268,119.51	$1,348,801.89	$1,432,457.19	$1,518,622.14	$1,607,372.05	$1,698,784.46	$1,793,561.02	$1,891,180.88
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$69,263.71	$71,341.62	$74,181.70	$76,407.15	$78,699.37	$80,295.63	$81,916.83	$84,374.34	$86,069.94	$88,652.04
Maintenance costs		$31,284.74	$32,223.29	$33,189.99	$34,185.68	$35,211.26	$36,267.59	$37,355.62	$38,476.29	$39,630.58	$40,819.50
Net annual cash flow		$100,548.46	$103,564.91	$107,371.69	$110,592.84	$113,910.62	$116,563.22	$119,272.45	$122,850.63	$125,700.52	$129,471.53
Present value of cash flow		$72,638.34	$72,638.34	$73,114.89	$73,114.89	$73,114.89	$72,638.34	$72,161.80	$72,161.80	$71,685.25	$71,685.25
Accumulate cash flow		$1,991,729.34	$2,095,294.25	$2,202,665.94	$2,313,258.77	$2,427,169.40	$2,543,732.61	$2,663,005.07	$2,785,855.70	$2,911,556.21	$3,041,027.75

Table 7.4-8 Life-Cycle Cost Analysis of the GSHP System in Phoenix, AZ

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,282,246.11										
Energy costs		$50,105.77	$52,130.25	$54,768.04	$56,964.13	$59,242.70	$61,019.98	$62,850.58	$64,736.09	$67,319.31	$69,338.89
Maintenance costs		$10,863.43	$11,189.34	$11,525.02	$11,870.77	$12,226.89	$12,593.70	$12,971.51	$13,360.65	$13,761.47	$14,174.32
Net annual cash flow	$1,282,246.11	$60,969.21	$63,319.59	$66,293.06	$68,834.90	$71,469.59	$73,613.68	$75,822.09	$78,096.75	$81,080.79	$83,513.21
Present value of cash flow	$1,282,246.11	$59,193.41	$59,684.78	$60,667.54	$61,158.92	$61,650.29	$61,650.29	$61,650.29	$61,650.29	$62,141.67	$62,141.67
Accumulate cash flow	$1,282,246.11	$1,343,215.32	$1,406,534.91	$1,472,827.97	$1,541,662.87	$1,613,132.45	$1,686,746.13	$1,762,568.21	$1,840,664.96	$1,921,745.75	$2,005,258.96
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$71,419.06	$73,561.63	$76,490.08	$78,784.79	$81,148.33	$82,794.26	$84,465.92	$86,999.90	$88,748.26	$91,410.71
Maintenance costs		$14,599.55	$15,037.53	$15,488.66	$15,953.32	$16,431.92	$16,924.88	$17,432.62	$17,955.60	$18,494.27	$19,049.10
Net annual cash flow		$86,018.61	$88,599.16	$91,978.74	$94,738.11	$97,580.25	$99,719.14	$101,898.54	$104,955.50	$107,242.53	$110,459.81
Present value of cash flow		$62,141.67	$62,141.67	$62,633.05	$62,633.05	$62,633.05	$62,141.67	$61,650.29	$61,650.29	$61,158.92	$61,158.92
Accumulate cash flow		$2,091,277.57	$2,179,876.73	$2,271,855.48	$2,366,593.58	$2,464,173.83	$2,563,892.97	$2,665,791.52	$2,770,747.01	$2,877,989.54	$2,988,449.35

Table 7.4-9 Life-Cycle Cost Analysis of the VAV Reheat System in Seattle, WA

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,246,841.58										
Energy costs		$38,416.22	$39,968.39	$41,990.79	$43,674.54	$45,421.52	$46,784.16	$48,187.69	$49,633.32	$51,613.88	$53,162.30
Maintenance costs		$31,565.90	$32,512.88	$33,488.26	$34,492.91	$35,527.70	$36,593.53	$37,691.34	$38,822.08	$39,986.74	$41,186.34
Net annual cash flow	$1,246,841.58	$69,982.12	$72,481.27	$75,479.05	$78,167.45	$80,949.22	$83,377.69	$85,879.03	$88,455.40	$91,600.62	$94,348.64
Present value of cash flow	$1,246,841.58	$67,943.80	$68,320.55	$69,074.03	$69,450.77	$69,827.51	$69,827.51	$69,827.51	$69,827.51	$70,204.25	$70,204.25
Accumulate cash flow	$1,246,841.58	$1,316,823.70	$1,389,304.96	$1,464,784.02	$1,542,951.47	$1,623,900.68	$1,707,278.38	$1,793,157.40	$1,881,612.80	$1,973,213.42	$2,067,562.06
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$54,757.17	$56,399.88	$58,645.13	$60,404.49	$62,216.62	$63,478.56	$64,760.23	$66,703.03	$68,043.51	$70,084.81
Maintenance costs		$42,421.93	$43,694.59	$45,005.43	$46,355.59	$47,746.26	$49,178.64	$50,654.00	$52,173.62	$53,738.83	$55,351.00
Net annual cash flow		$97,179.10	$100,094.47	$103,650.56	$106,760.08	$109,962.88	$112,657.21	$115,414.23	$118,876.66	$121,782.34	$125,435.81
Present value of cash flow		$70,204.25	$70,204.25	$70,580.99	$70,580.99	$70,580.99	$70,204.25	$69,827.51	$69,827.51	$69,450.77	$69,450.77
Accumulate cash flow		$2,164,741.15	$2,264,835.62	$2,368,486.18	$2,475,246.26	$2,585,209.14	$2,697,866.35	$2,813,280.58	$2,932,157.23	$3,053,939.57	$3,179,375.38

Table 7.4-10 Life-Cycle Cost Analysis of the Chilled Beam System in Seattle, WA

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$2,502,692.31										
Energy costs		$34,045.05	$35,420.61	$37,212.89	$38,705.05	$40,253.26	$41,460.85	$42,704.68	$43,985.82	$45,741.02	$47,113.25
Maintenance costs		$23,899.90	$24,616.89	$25,355.40	$26,116.06	$26,899.54	$27,706.53	$28,537.73	$29,393.86	$30,275.67	$31,183.94
Net annual cash flow	$2,502,692.31	$57,944.95	$60,037.50	$62,568.29	$64,821.12	$67,152.80	$69,167.38	$71,242.41	$73,379.68	$76,016.70	$78,297.20
Present value of cash flow	$2,502,692.31	$56,257.23	$56,591.10	$57,258.85	$57,592.72	$57,926.60	$57,926.60	$57,926.60	$57,926.60	$58,260.47	$58,260.47
Accumulate cash flow	$2,502,692.31	$2,560,637.26	$2,620,674.76	$2,683,243.05	$2,748,064.16	$2,815,216.96	$2,884,384.35	$2,955,626.75	$3,029,006.43	$3,105,023.13	$3,183,320.33
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$48,526.65	$49,982.45	$51,972.23	$53,531.40	$55,137.34	$56,255.69	$57,391.52	$59,113.26	$60,301.21	$62,110.25
Maintenance costs		$32,119.46	$33,083.05	$34,075.54	$35,097.80	$36,150.74	$37,235.26	$38,352.32	$39,502.89	$40,687.97	$41,908.61
Net annual cash flow		$80,646.11	$83,065.50	$86,047.77	$88,629.20	$91,288.08	$93,490.95	$95,743.84	$98,616.15	$100,989.19	$104,018.86
Present value of cash flow		$58,260.47	$58,260.47	$58,594.34	$58,594.34	$58,594.34	$58,260.47	$57,926.60	$57,926.60	$57,592.72	$57,592.72
Accumulate cash flow		$3,263,966.44	$3,347,031.94	$3,433,079.70	$3,521,708.90	$3,612,996.98	$3,706,487.93	$3,802,231.76	$3,900,847.91	$4,001,837.10	$4,105,855.96

Table 7.4-11 Life-Cycle Cost Analysis of the ASHP System in Seattle, WA

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$1,213,306.27										
Energy costs		$34,337.66	$35,725.04	$37,532.73	$39,037.72	$40,599.23	$41,817.21	$43,071.72	$44,363.87	$46,134.16	$47,518.19
Maintenance costs		$27,056.49	$27,868.18	$28,704.23	$29,565.35	$30,452.31	$31,365.88	$32,306.86	$33,276.07	$34,274.35	$35,302.58
Net annual cash flow	$1,213,306.27	$61,394.15	$63,593.22	$66,236.96	$68,603.07	$71,051.54	$73,183.09	$75,378.58	$77,639.94	$80,408.51	$82,820.76
Present value of cash flow	$1,213,306.27	$59,605.97	$59,942.71	$60,616.20	$60,952.94	$61,289.68	$61,289.68	$61,289.68	$61,289.68	$61,626.43	$61,626.43
Accumulate cash flow	$1,213,306.27	$1,274,700.42	$1,338,293.64	$1,404,530.60	$1,473,133.67	$1,544,185.21	$1,617,368.30	$1,692,746.88	$1,770,386.82	$1,850,795.33	$1,933,616.10
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$48,943.73	$50,412.04	$52,418.92	$53,991.49	$55,611.24	$56,739.20	$57,884.79	$59,621.34	$60,819.50	$62,644.08
Maintenance costs		$36,361.65	$37,452.50	$38,576.08	$39,733.36	$40,925.36	$42,153.12	$43,417.72	$44,720.25	$46,061.86	$47,443.71
Net annual cash flow		$85,305.39	$87,864.55	$90,995.00	$93,724.85	$96,536.60	$98,892.32	$101,302.51	$104,341.58	$106,881.35	$110,087.79
Present value of cash flow		$61,626.43	$61,626.43	$61,963.17	$61,963.17	$61,963.17	$61,626.43	$61,289.68	$61,289.68	$60,952.94	$60,952.94
Accumulate cash flow		$2,018,921.48	$2,106,786.03	$2,197,781.04	$2,291,505.89	$2,388,042.49	$2,486,934.81	$2,588,237.32	$2,692,578.91	$2,799,460.26	$2,909,548.05

Table 7.4-12 Life-Cycle Cost Analysis of the GSHP System in Seattle, WA

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$1,490,329.91										
Energy costs		$25,002.47	$26,012.67	$27,328.92	$28,424.75	$29,561.74	$30,448.59	$31,362.05	$32,302.91	$33,591.92	$34,599.68
Maintenance costs		$12,626.36	$13,005.15	$13,395.31	$13,797.16	$14,211.08	$14,637.41	$15,076.53	$15,528.83	$15,994.70	$16,474.54
Net annual cash flow	$1,490,329.91	$37,628.83	$39,017.82	$40,724.22	$42,221.92	$43,772.82	$45,086.01	$46,438.59	$47,831.74	$49,586.62	$51,074.22
Present value of cash flow	$1,490,329.91	$36,532.85	$36,778.04	$37,268.43	$37,513.63	$37,758.82	$37,758.82	$37,758.82	$37,758.82	$38,004.01	$38,004.01
Accumulate cash flow	$1,490,329.91	$1,527,958.75	$1,566,976.57	$1,607,700.79	$1,649,922.71	$1,693,695.53	$1,738,781.53	$1,785,220.12	$1,833,051.86	$1,882,638.48	$1,933,712.70
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$35,637.67	$36,706.80	$38,168.08	$39,313.12	$40,492.52	$41,313.83	$42,147.97	$43,412.41	$44,284.84	$45,613.38
Maintenance costs		$16,968.77	$17,477.84	$18,002.17	$18,542.24	$19,098.50	$19,671.46	$20,261.60	$20,869.45	$21,495.53	$22,140.40
Net annual cash flow		$52,606.44	$54,184.64	$56,170.25	$57,855.36	$59,591.02	$60,985.29	$62,409.58	$64,281.86	$65,780.37	$67,753.78
Present value of cash flow		$38,004.01	$38,004.01	$38,249.21	$38,249.21	$38,249.21	$38,004.01	$37,758.82	$37,758.82	$37,513.63	$37,513.63
Accumulate cash flow		$1,986,319.14	$2,040,503.78	$2,096,674.03	$2,154,529.39	$2,214,120.41	$2,275,105.70	$2,337,515.27	$2,401,797.13	$2,467,577.50	$2,535,331.28

Table 7.4-13 Life-Cycle Cost Analysis of the VAV Reheat System in Spokane, WA

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$1,100,984.64										
Energy costs		$37,549.36	$39,066.50	$41,043.27	$42,689.02	$44,396.58	$45,728.48	$47,100.34	$48,513.35	$50,449.21	$51,962.69
Maintenance costs		$27,873.29	$28,709.48	$29,570.77	$30,457.89	$31,371.63	$32,312.78	$33,282.16	$34,280.63	$35,309.04	$36,368.32
Net annual cash flow	$1,100,984.64	$65,422.64	$67,775.99	$70,614.04	$73,146.91	$75,768.21	$78,041.26	$80,382.50	$82,793.97	$85,758.26	$88,331.01
Present value of cash flow	$1,100,984.64	$63,517.13	$63,885.37	$64,621.85	$64,990.09	$65,358.33	$65,358.33	$65,358.33	$65,358.33	$65,726.56	$65,726.56
Accumulate cash flow	$1,100,984.64	$1,166,407.28	$1,234,183.27	$1,304,797.31	$1,377,944.22	$1,453,712.43	$1,531,753.69	$1,612,136.19	$1,694,930.16	$1,780,688.42	$1,869,019.42

year		11	12	13	14	15	16	17	18	19	20
Capital costs											
Energy costs		$53,521.57	$55,127.22	$57,321.81	$59,041.46	$60,812.71	$62,046.17	$63,298.91	$65,197.88	$66,508.11	$68,503.35
Maintenance costs		$37,459.37	$38,583.15	$39,740.64	$40,932.86	$42,160.85	$43,425.67	$44,728.44	$46,070.29	$47,452.40	$48,875.98
Net annual cash flow		$90,980.94	$93,710.37	$97,062.45	$99,974.32	$102,973.55	$105,471.84	$108,027.35	$111,268.17	$113,960.51	$117,379.32
Present value of cash flow		$65,726.56	$65,726.56	$66,094.80	$66,094.80	$66,094.80	$65,726.56	$65,358.33	$65,358.33	$64,990.09	$64,990.09
Accumulate cash flow		$1,960,000.36	$2,053,710.73	$2,150,773.17	$2,250,747.49	$2,353,721.05	$2,459,192.89	$2,567,220.24	$2,678,488.41	$2,792,448.92	$2,909,828.25

Table 7.4-14 Life-Cycle Cost Analysis of the Chilled Beam System in Spokane, WA

year	0	1	2	3	4	5	6	7	8	9	10
Capital costs	$2,209,924.53										
Energy costs		$31,849.59	$33,136.44	$34,813.15	$36,209.08	$37,657.45	$38,787.17	$39,950.79	$41,149.31	$42,791.33	$44,075.07
Maintenance costs		$21,104.06	$21,737.18	$22,389.30	$23,060.98	$23,752.80	$24,465.39	$25,199.35	$25,955.33	$26,733.99	$27,536.01
Net annual cash flow	$2,209,924.53	$52,953.65	$54,873.62	$57,202.44	$59,270.06	$61,410.25	$63,252.56	$65,150.14	$67,104.64	$69,525.32	$71,611.08
Present value of cash flow	$2,209,924.53	$51,411.31	$51,723.65	$52,348.34	$52,660.68	$52,973.02	$52,973.02	$52,973.02	$52,973.02	$53,285.37	$53,285.37
Accumulate cash flow	$2,209,924.53	$2,262,878.18	$2,317,751.80	$2,374,954.24	$2,434,224.30	$2,495,634.55	$2,558,887.11	$2,624,037.25	$2,691,141.89	$2,760,667.21	$2,832,278.28

year		11	12	13	14	15	16	17	18	19	20
Capital costs											
Energy costs		$45,397.32	$46,759.24	$48,620.70	$50,079.32	$51,581.70	$52,627.93	$53,690.52	$55,301.23	$56,412.57	$58,104.95
Maintenance costs		$28,362.09	$29,212.95	$30,089.34	$30,992.02	$31,921.78	$32,879.44	$33,865.82	$34,881.79	$35,928.25	$37,006.10
Net annual cash flow		$73,759.41	$75,972.19	$78,710.04	$81,071.34	$83,503.48	$85,507.37	$87,556.34	$90,183.03	$92,340.82	$95,111.05
Present value of cash flow		$53,285.37	$53,285.37	$53,597.71	$53,597.71	$53,597.71	$53,285.37	$52,973.02	$52,973.02	$52,660.68	$52,660.68
Accumulate cash flow		$2,906,037.69	$2,982,009.88	$3,060,719.92	$3,141,791.26	$3,225,294.75	$3,310,802.12	$3,398,358.45	$3,488,541.48	$3,580,882.30	$3,675,993.34

Table 7.4-15 Life-Cycle Cost Analysis of the ASHP System in Spokane, WA

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$1,071,372.33										
Energy costs		$33,180.83	$34,521.47	$36,268.26	$37,722.55	$39,231.45	$40,408.39	$41,620.64	$42,869.26	$44,579.91	$45,917.31
Maintenance costs		$23,891.39	$24,608.13	$25,346.37	$26,106.76	$26,889.97	$27,696.67	$28,527.57	$29,383.39	$30,264.90	$31,172.84
Net annual cash flow	$1,071,372.33	$57,072.22	$59,129.60	$61,614.63	$63,829.31	$66,121.42	$68,105.06	$70,148.21	$72,252.66	$74,844.81	$77,090.15
Present value of cash flow	$1,071,372.33	$55,409.92	$55,735.32	$56,386.12	$56,711.52	$57,036.91	$57,036.91	$57,036.91	$57,036.91	$57,362.31	$57,362.31
Accumulate cash flow	$1,071,372.33	$1,128,444.55	$1,187,574.15	$1,249,188.79	$1,313,018.10	$1,379,139.51	$1,447,244.57	$1,517,392.78	$1,589,645.44	$1,664,490.24	$1,741,580.39
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$47,294.83	$48,713.67	$50,652.94	$52,172.53	$53,737.71	$54,827.67	$55,934.66	$57,612.70	$58,770.50	$60,533.61
Maintenance costs		$32,108.03	$33,071.27	$34,063.41	$35,085.31	$36,137.87	$37,222.00	$38,338.66	$39,488.82	$40,673.49	$41,893.69
Net annual cash flow		$79,402.86	$81,784.94	$84,716.35	$87,257.84	$89,875.57	$92,049.67	$94,273.33	$97,101.53	$99,443.99	$102,427.31
Present value of cash flow		$57,362.31	$57,362.31	$57,687.71	$57,687.71	$57,687.71	$57,362.31	$57,036.91	$57,036.91	$56,711.52	$56,711.52
Accumulate cash flow		$1,820,983.25	$1,902,768.19	$1,987,484.54	$2,074,742.37	$2,164,617.95	$2,256,667.62	$2,350,940.95	$2,448,042.48	$2,547,486.46	$2,649,913.77

Table 7.4-16 Life-Cycle Cost Analysis of the GSHP System in Spokane, WA

year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Capital costs	$1,315,989.43										
Energy costs		$21,872.99	$22,756.75	$23,908.24	$24,866.91	$25,861.59	$26,637.44	$27,436.56	$28,259.66	$29,387.32	$30,268.94
Maintenance costs		$11,149.31	$11,483.79	$11,828.31	$12,183.16	$12,548.65	$12,925.11	$13,312.86	$13,712.25	$14,123.62	$14,547.33
Net annual cash flow	$1,315,989.43	$33,022.30	$34,240.54	$35,736.55	$37,050.07	$38,410.24	$39,562.55	$40,749.42	$41,971.91	$43,510.94	$44,816.27
Present value of cash flow	$1,315,989.43	$32,060.49	$32,274.99	$32,704.00	$32,918.51	$33,133.01	$33,133.01	$33,133.01	$33,133.01	$33,347.51	$33,347.51
Accumulate cash flow	$1,315,989.43	$1,349,011.74	$1,383,252.28	$1,418,988.82	$1,456,038.89	$1,494,449.13	$1,534,011.67	$1,574,761.10	$1,616,733.00	$1,660,243.94	$1,705,060.21
year		**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
Capital costs											
Energy costs		$31,177.01	$32,112.32	$33,390.70	$34,392.42	$35,424.19	$36,142.70	$36,872.44	$37,978.61	$38,741.84	$39,904.09
Maintenance costs		$14,983.75	$15,433.26	$15,896.26	$16,373.14	$16,864.34	$17,370.27	$17,891.38	$18,428.12	$18,980.96	$19,550.39
Net annual cash flow		$46,160.76	$47,545.58	$49,286.95	$50,765.56	$52,288.53	$53,512.97	$54,763.82	$56,406.73	$57,722.80	$59,454.48
Present value of cash flow		$33,347.51	$33,347.51	$33,562.02	$33,562.02	$33,562.02	$33,347.51	$33,133.01	$33,133.01	$32,918.51	$32,918.51
Accumulate cash flow		$1,751,220.97	$1,798,766.55	$1,848,053.51	$1,898,819.07	$1,951,107.60	$2,004,620.57	$2,059,384.39	$2,115,791.12	$2,173,513.92	$2,232,968.40

According to the calculation above, the present value of the 20-year total life cost of the systems are shown in Table 7.4-17, and Figure 7.4-2 shows.

Table 7.4-17 Present Value of Total Life-Cycle Cost

Life-Cycle Cost (PV)	Miami, FL	Phoenix, AZ	Seattle, WA	Spokane, WA
VAV Reheat	$2,360,719.32	$2,683,613.37	$2,642,261.49	$2,407,046.43
Chilled Beam	$3,094,338.75	$3,481,133.70	$3,660,222.60	$3,268,447.96
ASHP	$2,225,612.86	$2,485,707.53	$2,438,089.73	$2,211,134.41
GSHP	$2,278,818.05	$2,513,777.86	$2,244,770.72	$1,978,006.11

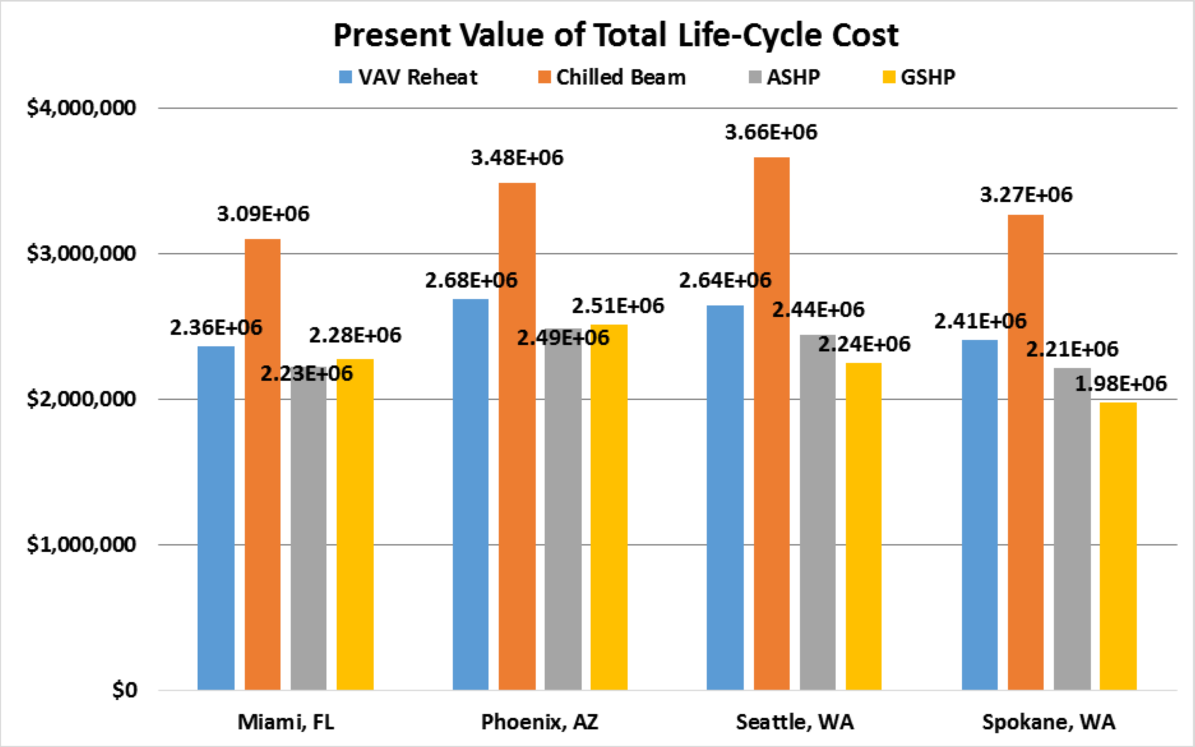


Figure 7.4-2 Present Value of Total Life-Cycle Cost

In all the four climate zones, chilled beam system has the highest life cycle cost. In Miami and Phoenix, the ASHP system is the most cost-efficient system; in Seattle and Spokane, the GSHP system has the lowest overall cost.

Figure 7.4-3 ~ Figure 7.4-6 are the accumulated cash flow curves for the four systems in different locations.

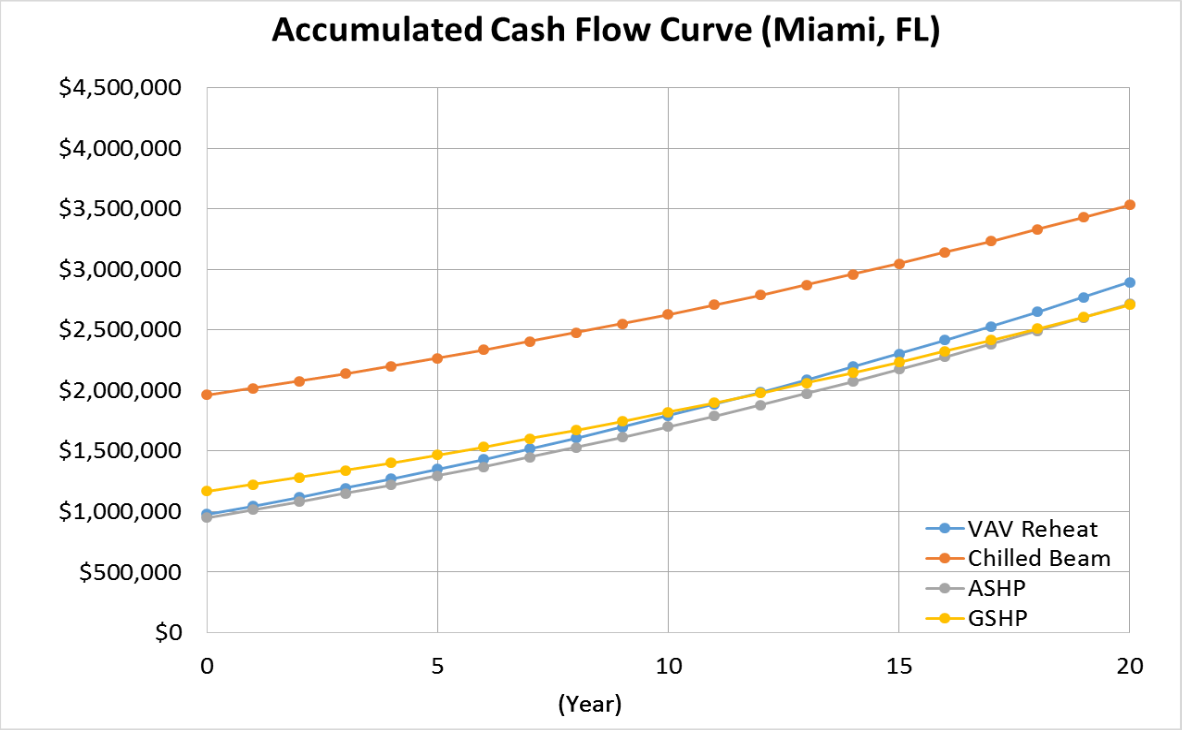


Figure 7.4-3 Accumulate Cash Flow Curve (Miami, FL)

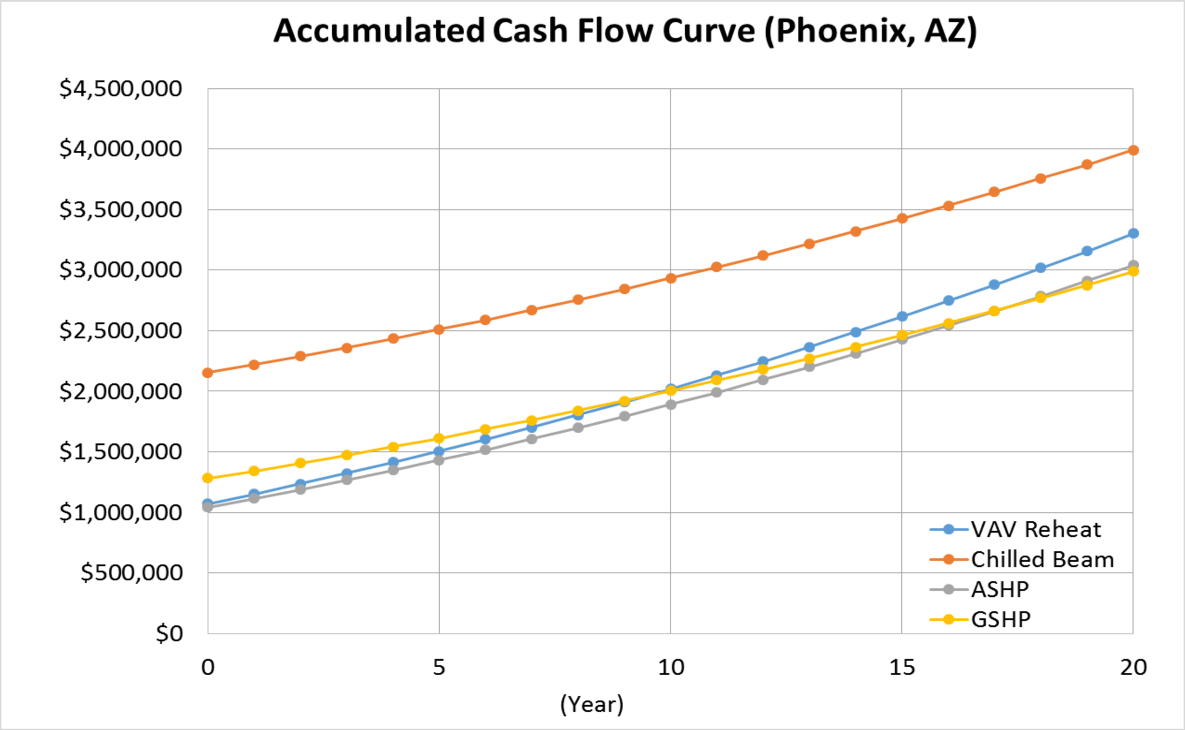


Figure 7.4-4 Accumulate Cash Flow Curve (Phoenix, AZ)

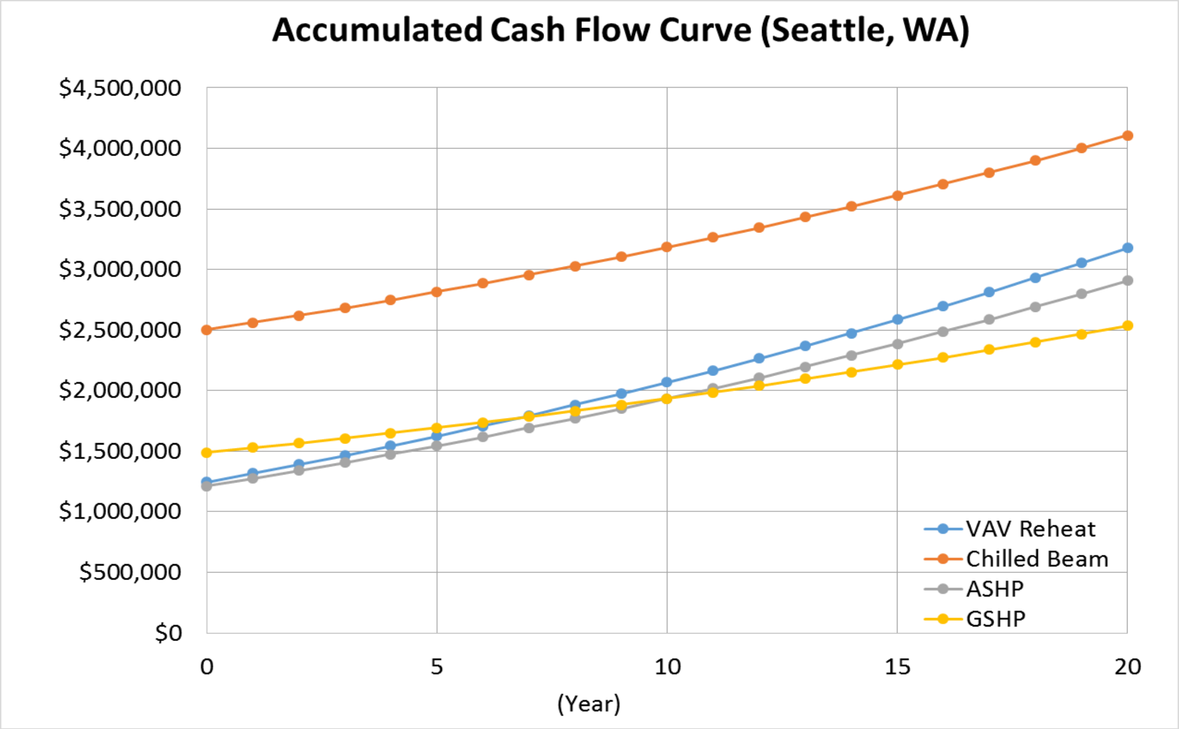


Figure 7.4-5 Accumulate Cash Flow Curve (Seattle, WA)

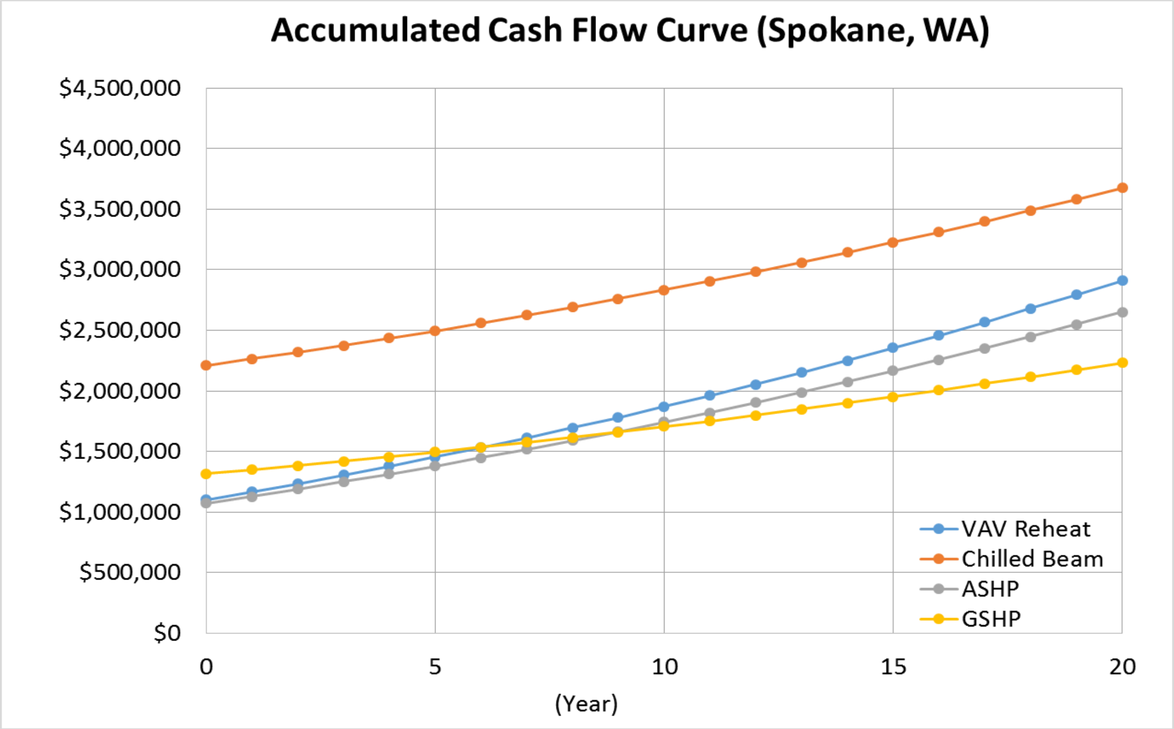


Figure 7.4-6 Accumulate Cash Flow Curve (Spokane, WA)

In Miami, the ASHP system has the lowest capital and energy cost. It has the lowest present value of total life-cycle cost, Figure 7.4-2. However, the GSHP system has a lower accumulated cash flow at the end of 20th year, Figure 7.4-3.

Phoenix has a ten-year payback period when compared to the VAV reheat system, and has an eighteen-year payback period when compared to the ASHP system, Figure 7.4-4. Chilled beam system stays on the top of the cost during the whole life cycle of the analysis, even though it consumes the least energy in both Miami and Phoenix.

For Seattle and Spokane, the ground source heat pump has short payback periods. Seattle has a seven-year payback period when compared to the VAV reheat system, and has a ten-year payback period when compared to the ASHP system, Figure 7.4-5. Spokane has a six-year payback period when compared to the VAV reheat system, and has a nine-year payback period when compared to the ASHP system, Figure 7.4-6.

CHAPTER 8. CONCLUSION

8.1 ASSUMPTIONS AND LIMITATIONS

The goal of this research is to illustrate a way of selecting the most a suitable HVAC system for a project in a specific climate condition. Suitability of a system is based on having the lowest life cycle cost.

A methodology compared the life-cycle cost of VAV reheat system, Chilled Beam system, ASHP system and GSHP system in four different climate zones, Miami (FL), Phoenix (AZ), Seattle (WA), Spokane (WA).

This research only compared four types of system in four different climate zones on one typical educational office building. Thus, the conclusion of this research may not apply to any project in the United States. The project purpose is not to pick one best system for all projects but is to illustrate a practical way of selecting the most suitable HVAC system for a project in the specific climate condition.

The eQUEST energy modeling software is a graphical-user interface that runs on DOE-2.2, and there are more than sixty HVAC system types available to choose in eQUEST. The simulation of VAV reheat system, ASHP system and GSHP system was easy to operate. However, since eQUEST does not have an exactly chilled beam component, (Vaughn, 2012) modeling the chilled beam system was one of the difficulties in this project.

Based on the previous research of chilled beam energy modeling, eQUEST can model chilled beams with a reasonable degree of accuracy by using induction unit system. (Betz, F., McNeill, J., Talbert, B., Thimmanna, H., and Repka, N., 2012) All chilled beam models in this research used the induction unit, which are the most similar in concept to chilled beams.

The existing research on the life-cycle analysis in built environment reviewed was to compare the simple construction tools or different building materials, but there is limited research done on the life cycle cost analysis of HVAC systems.

Since there is not a specific design for each HVAC system studied, it is difficult to estimate the accurate initial cost and the maintenance cost for all the models. The cost information varies significantly based on the specific design, the capital cost and the maintenance cost of a system may vary, so the specific conclusions should not be derived from this work.

This research investigates three aspects of cost information, 1) initial capital cost, 2) life energy cost, and 3) life maintenance cost. This research uses a discount rate of 3% in the Present value calculation.

The capital cost of each system are derived from the past research and publications. This cost is adjusted with the historical index data and the location index data according to 2016. RS Means Construction Cost Data. (R.S. Means Company, 2016) The annual energy consumption is acquired through eQUEST energy modeling analysis of a sample educational office building. The annual electricity costs are calculated with the inflation rate of 3% per year with an additional fuel price index from the NIST Handbook (Lavappa and Kneifel, 2016). The annual costs for maintenance is calculated with the inflation rate of 3% per year over the study period.

8.2 CONCLUSIONS

Based on the energy load simulation result for the four locations, Miami and Phoenix have high cooling load and low heating load. However, in Seattle and Spokane, the cooling load is about half of that in Miami and Phoenix, and the heating load is much higher than that in Miami and Phoenix, as Figure 8.2-1

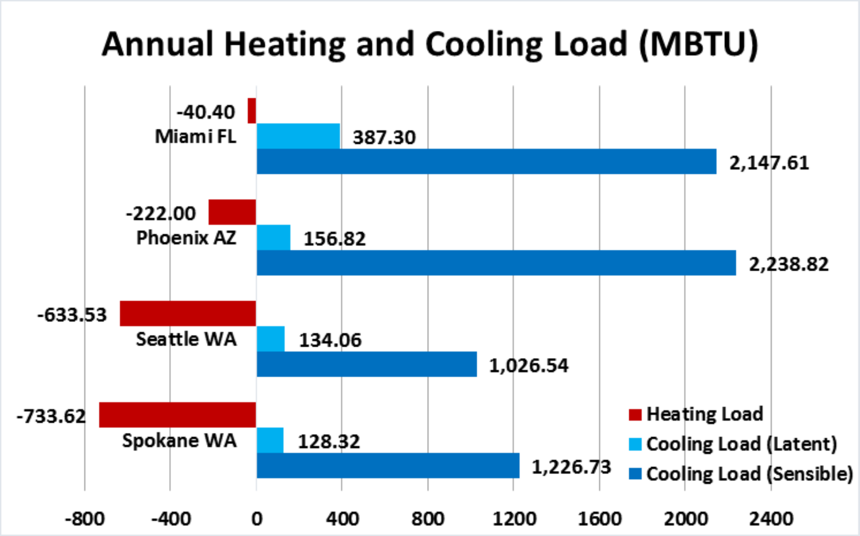


Figure 8.2-1 Annual Heating and Cooling Load

Other than the annual heating and cooling load, the peak heating and peak cooling load data is important when deciding the equipment size for a specific project. A project with higher peak loads will require installing a larger chiller or boiler.

The peak heating loads and the peak cooling loads of the four locations do not have huge differences, except that Seattle has a lower peak cooling load, and Spokane has a higher peak heating load, as shown in Figure 8.2-2.

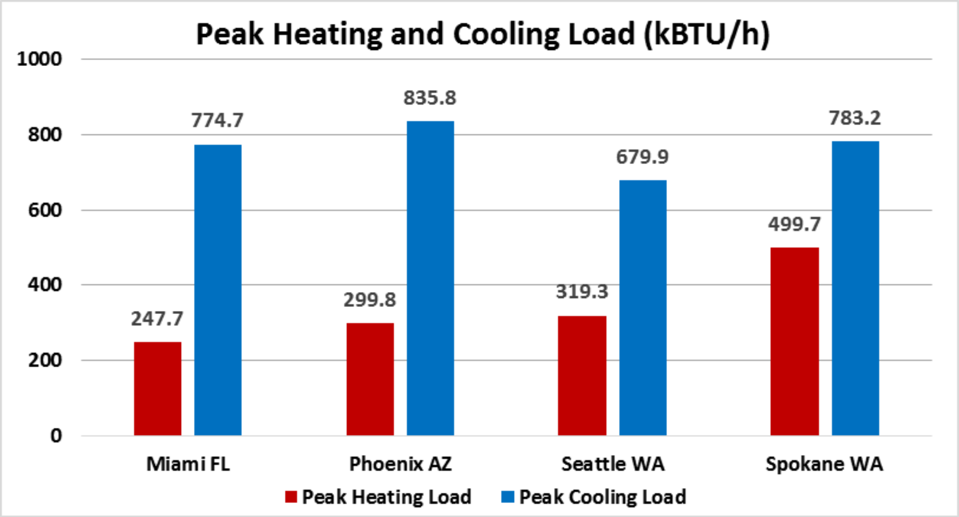


Figure 8.2-2 Peak Heating and Cooling Load

Through the eQUEST simulation result for the sixteen energy models, the lowest energy consuming system varies for different locations. The most energy efficient system in Miami and Phoenix is the Chilled Beam system, and in Seattle and Spokane is the GSHP system, as Figure 8.2-3 shows.

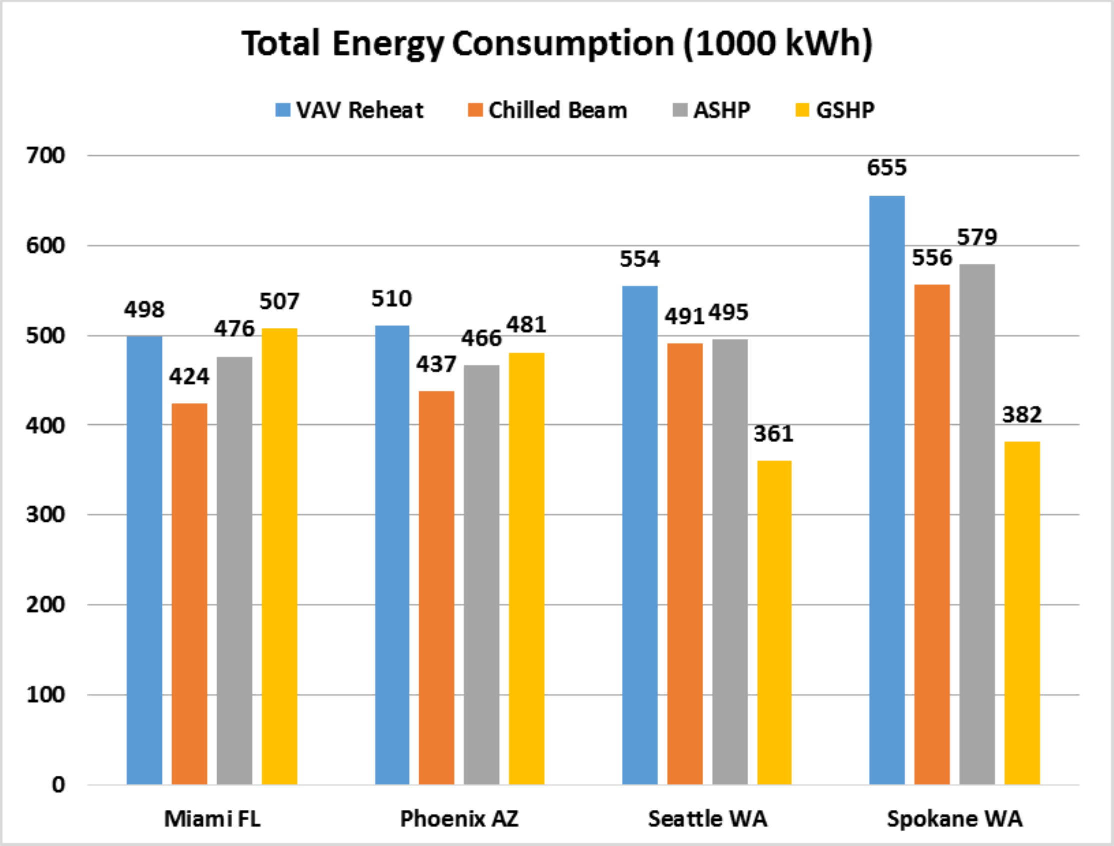


Figure 8.2-3 Energy Consumptions

According to the result, chilled beam system is more efficient in cooling dominant climate zones, and it does not have a superiority in energy consumption for the heating dominant climate zones. The energy consumption of the GSHP system is highly related to the temperature difference between the ground layer and the underground layer, which is why GSHP system consumes less energy in Seattle and Spokane but spends more energy in Miami.

Chilled beam system has the highest life cycle cost in all studied climate zones due to the high initial cost. VAV reheat system comes after chilled beam system since the energy cost is the highest (except in Miami). Both types of the heat pump systems have lower overall life cost compared to the other two systems. In Miami and Phoenix, the ASHP performs better than the GSHP system. In Seattle and Spokane, the GSHP performs the best, as Figure 8.2-4 shows.

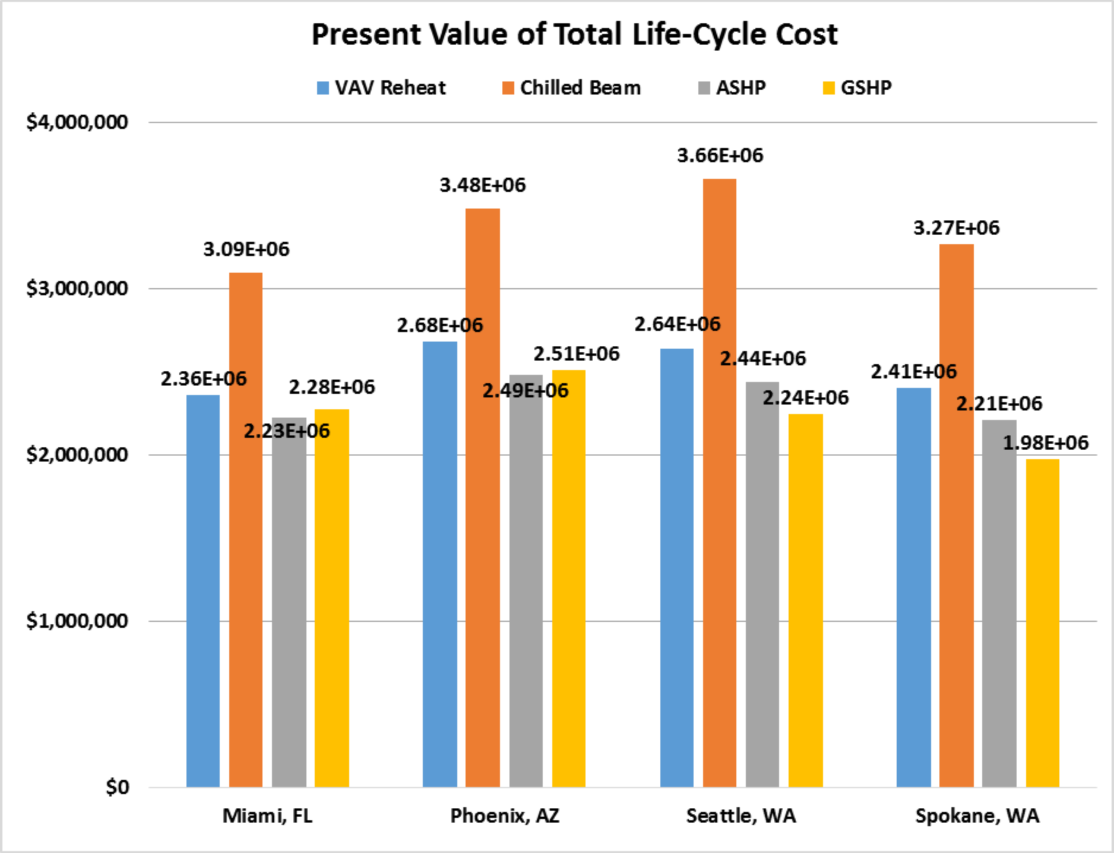


Figure 8.2-4 Present Value of Life-Cycle Cost

Figure 8.2-5 collects all the cash flow curves of the sixteen models from the conducted life cycle cost analysis.

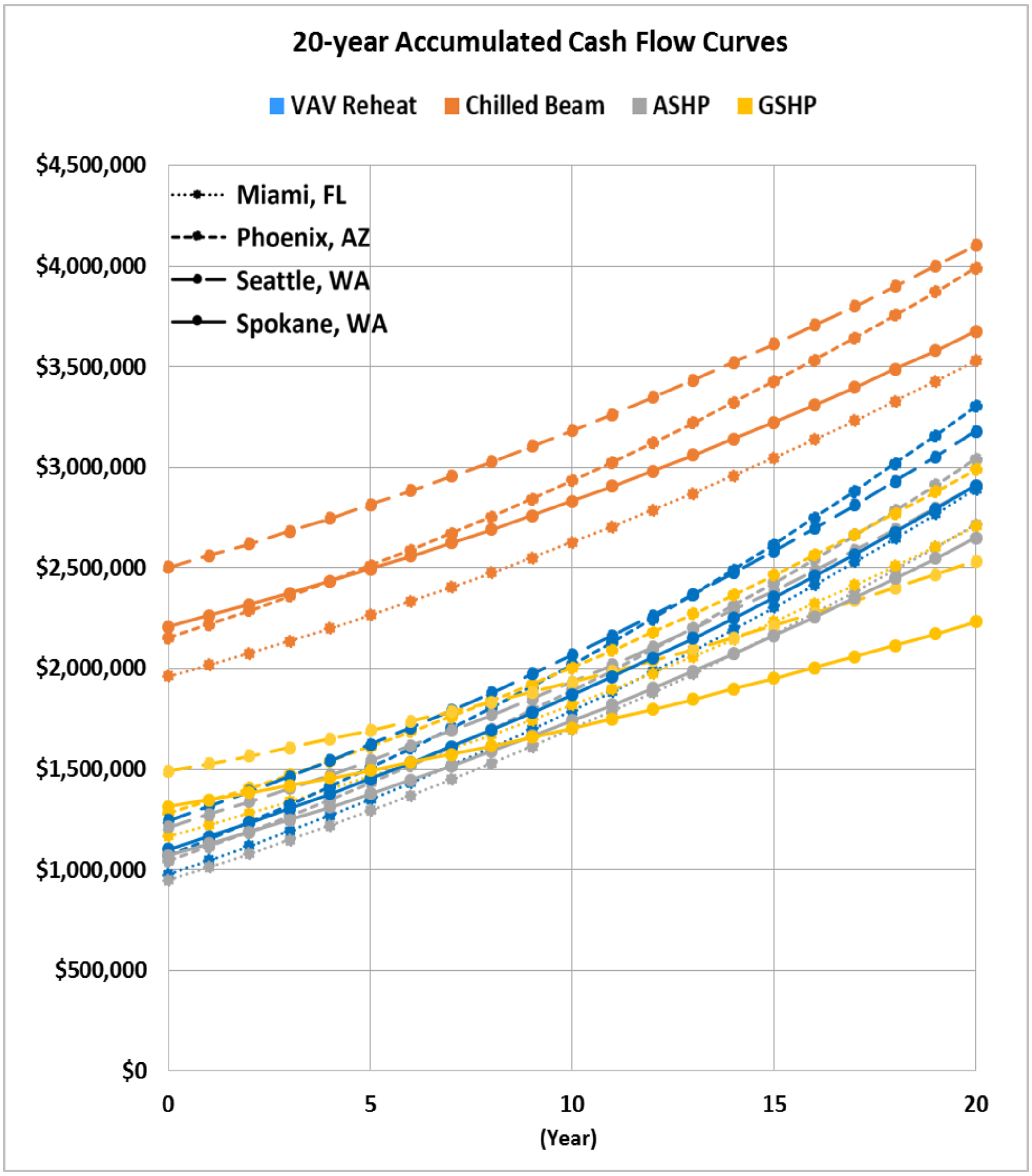


Figure 8.2-5 Accumulated Cash Flow Curve

According to the cash flow curves, chilled beam system has the highest life-cycle cost; VAV reheat system is the second largest, then follows by ASHP system, GSHP system is the most cost-effective system in the 20-year's life spam. From the present value comparison, ASHP system has the lowest present value of the life-cycle cost in Miami and Phoenix. However, according to the accumulated cash flow curves, in Miami and Phoenix, ASHP system and GSHP system have similar accumulated cash flow around the 19th year, and GSHP system has the lowest accumulated cash flow over the whole 20 years.

The energy simulation result and the life-cycle cost analysis demonstrate that chilled beam system is the least cost efficient one of the four systems assessed. It is necessary to carefully considered before applying chilled beam system in heating dominant climate zones, such as Spokane, or if the owner wants to have a shorter payback period.

Based on the life-cycle cost analysis, it is apparent that GSHP systems can be highly recommended for projects in most climate zones, except the climate zone where there is a small temperature difference between the ground level and the underground level like Miami.

This research identified the benefits and the potential limitations of the four HVAC systems in four different locations, and provided stakeholders practical hints on selecting HVAC system for a new project. The research can be used as a guideline for the projects with other options of HVAC systems or in the locations other than the four cities discussed above.

The next steps of this research will include practical applications using the illustrated methodology to select HVAC system. In the future study, sensitivity analysis on the discount rate and the inflation rate need to be done to see if those factors will influence the result of this life-cycle cost analysis. The accuracy of the energy models should be verified through auditing real projects to examine if the energy models are capable of predicting the future energy cost.

BIBLIOGRAPHY

American Institute of Architects. (1977*). Life cycle cost analysis: A guide for architects.* Washington]: AIA.

American Society of Heating, Refrigerating Air-Conditioning Engineers (ASHRAE). (2005). *2005 ASHRAE handbook: Fundamentals*, Chapter 20, figure 8. Atlanta, GA: American Society of Heating, Refrigeration and Air-Conditioning Engineers.

American Society of Heating, Refrigerating Air-Conditioning Engineers (ASHRAE). (2010). *Energy standard for buildings except low-rise residential buildings.* (I-P ed., ASHRAE standard; 90.1). Atlanta, GA: ASHRAE.

American Society of Heating, Refrigerating Air-Conditioning Engineers (ASHRAE). (2013). *ASHRAE equipment life expectancy chart*, Retrieved from http://culluminc.com/wp-content/uploads/2013/02/ashrae%20hvac%20life%20expectancy1.pdf

Apprill, J. (2013). *Order of Magnitude Mechanical Estimating.* Retrieved from http://www.buildersassociation.com/docs/Education/Estimating%20Academy/Mechanical%20Budgetary%20Estimating%20January%202013.pdf

Autodesk Ecotect Analysis. (Version 2011). [Computer software]. Retrieved from http://vymzvlbtahnwxtmd.depotapps.com/product/autodesk-ecotect-analysis-2011/

Betz, F., McNeill, J., Talbert, B., Thimmanna, H., and Repka, N. (2012). *Issues arising from the use of chilled beams in energy models*. Retrieved from https://www.researchgate.net/publication/256375218_Issues_arising_from_the_use_of_chilled_beams_in_energy_models

Bloomquist, R.G. (2011). *The economics of geothermal heat pump systems for commercial and institutional buildings.* Retrieved from http://blogs.worldwatch.org/revolt/wp-content/uploads/2011/07/Additional-Geothermal-Doc.docx

Brenden, K. (2010). *Why R is not simply the inverse of U*. www.dwmmag.com. Retrieved from http://www.aamanet.org/upload/file/Why_R_Is_Not_Simply_The_Inverse_of_U_September.pdf

Burdick, A. (2011, June). *Strategy guideline: accurate heating and cooling load calculations.* Report prepared for Building Technologies Program, Office of Energy Efficiency and Renewable Energy, U.S. Department of Energy. Retrieved from IBACOS, Inc. website: http://www.nrel.gov/docs/fy11osti/51603.pdf

Chiasson, A. (2006). *Life-cycle cost study of a geothermal heat pump system.* Retrieved from http://www.oit.edu/docs/default-source/geoheat-center-documents/toa/winnebago-life-cycle-cost-analysis-report.pdf?sfvrsn=6

China Zhongyuan Engineering Corp. (2014). *An investigation report for the project of the University of Dar es Salaam library.*

Climate Consultant. (Version 6.0). [Computer software]. Retrieved from http://www.energy-design-tools.aud.ucla.edu/climate-consultant/request-climate-consultant.php

Davis, M., Coony, R., Gould, S., and Daly, A. (2005). Guidelines for life cycle cost analysis. Stanford University Land and Buildings. Retrieved from https://lbre.stanford.edu/finance/sites/all/lbre-shared/files/docs_public/LCCA121405.pdf

Demkin, J., & American Institute of Architects. (2001). *The architect's handbook of professional practice (13th ed.).* New York: J. Wiley.

Electricity Local. (2017). *Local electricity rates and statistics.* Retrieved from http://www.electricitylocal.com/

Energy-Models.com. (n. d.). *What is energy modeling & building simulation?* Retrieved from http://energy-models.com/what-is-energy-modeling-building-simulation

eQUEST. (2008). *eQUEST-The quick energy simulation tool, an overview.* Retrieved from http://www.doe2.com/download/equest/eQUESTv3-Overview.pdf eQUEST. (Version 3-65). [Computer software]. Retrieved from http://www.doe2.com/equest/

Feldkamp, J. (2009) *Office Insights—Warming Ups to Chilled Bean Technology.* Retrieved from HIXSON website: http://www.hixson-inc.com/_images/winsights/warminguptochilledbeams.pdf

Harvey, L. (2009). *Reducing energy use in the buildings sector: Measures, costs, and examples.* Energy Efficiency, 2(2), 139-163.

Ihnen, J. L., Weitner, S. L. and O'Donnell P. C. (2012). *Rescuing New Construction from Widgetitis with Simple HVAC Design.* Retrieved from http://michaelsenergy.com/wp-content/uploads/2015/10/Rescuing-New-Construction-from-Widgetitis-with-Simple-HVAC-Design-ACEEE.pdf

Karr, M. (2011, March*) Ground-Source Variable Refrigerant Flow Heat Pumps,* Retrieved from http://www.energy.wsu.edu/Documents/EEFactsheet-GSHP-Feb2011.pdf

Kavanaugh, S., Green, M., & Mescher, K. (2012). *Long-Term Commercial GSHP Performance: Part 4: Installation Costs*. ASHRAE Journal, 54(10), 26-36.

Lavappa, P.D. and Kneifel, J.D. (2016). *Energy Price Indices and Discount Factors for Life-Cycle Cost Analysis – 2016, Annual Supplement to NIST Handbook 135.* U.S. Department of Commerce, U.S. Department of Energy, Washington, DC. doi: 10.6028/NIST.IR.85-3273-31

Loudermilk, K. (2009). *Designing chilled beams for thermal comfort.* ASHRAE Journal 51(10):58–64. Retrieved from http://www.taco-hvac.com/uploads/FileLibrary/DesigningChilledBeamsforThermalComfort_Loudermilk_ASHRAEJournal_Oct2009.pdf

Loudermilk, K. (2016). *Chilled beam systems and the 2016 WSEC. ASHRAE Presentation.* Retrieved from https://pugetsoundashrae.org/wp-content/uploads/2016/02/Seattle-ASHRAE-2016-February-Presentation.pdf

Madison Gas and Electric Company, MGEC. (2015, October 28). *Reheat Systems for Commercial Buildings*. Retrieved from https://www.mge.com/images/PDF/Brochures/business/ReheatSystemForCommercialBuildingsFactSheet.pdf

Murphy, J. (2011, April). *Understanding chilled beam systems.* Retrieved from Trane Engineers Newsletter volume 38-4, website: https://www.trane.com/content/dam/Trane/Commercial/global/products-systems/education-training/engineers-newsletters/airside-design/adm_apn034en_1209.pdf

Natural Resources Canada, Office of Energy Efficiency. (2004) *Heating and Cooling with a Heat Pump.* Retrieved from https://www.nrcan.gc.ca/sites/oee.nrcan.gc.ca/files/pdf/publications/infosource/pub/home/heating-heat-pump/booklet.pdf

New York City Mayor's Office of Sustainability (MOS). (2015, February). *Geothermal Systems and Their Applications in New York City.* Retrieved from http://www.nyc.gov/html/planyc/downloads/pdf/publications/2015_Geothermal.pdf

Nicolow, J., and Sami, V. (2006). *Energy efficiency at the Eco Office: Harnessing state-of-the-shelf technology for state-of-the-art performance.(Leadership in Energy and Environmental Design).* Environmental Design & Construction, 9(2), 34.

Park, J. (2013). *Comparative analysis of the VRF System and conventional HVAC Systems, focused on life cycle-cost.* Atlanta, GA-USA: Georgia Institute of Technology.

R.S. Means Company. (2016). *RSMeans building construction cost data 2016 (74rd annual ed.).* Norwell, MA: RSMeans.

Rumsey, P., Bulger, N., Wenisch, J., and Disney, T. (2009, June). *Chilled beams in Laboratories - Key strategies to ensure effective design, construction, and operation.* Retrieved from http://www.i2sl.org/documents/toolkit/bp_chilled-beam_508.pdf

Sami, V. (2015, October). *Integrated Building Systems.* University of Washington, ARCH 530 - Integrated Building Systems Autumn Quarter, 2015

Setty, B.S. (2011). *Application issues for chilled beam technologies.* ASHRAE Transactions 117:494–501. Retrieved from http://www.setty.com/pdf/ASHRAE-Application-Issues.pdf

Smeed, D. (2007, September) *Chilled beams or variable air volume*. EcoLibrium®. Retrieved from http://seedengr.com/Chilled%20beams%20or%20variable%20air%20volume.pdf

Solar chimney. (2017, January 5). Retrieved January 21, 2017, from In Designing Buildings Wiki: http://www.designingbuildings.co.uk/wiki/Solar_chimney

Stein, B., Reynolds, J. S., Grondzik, W. T., & Kwok, A. G. (2006). *Mechanical and electrical equipment for buildings (10th ed.).* Hoboken, NJ: John Wiley & Sons, Inc.

Stein, J., and Taylor, S. T. (2013). *VAV reheat versus active chilled beams & DOAS. (TECHNICAL FEATURE)*. ASHRAE Journal, 55(5), 18.

Taylor Engineering. (2014). *How to Estimate the Cost of a VAV Reheat HVAC System.* Taylor Engineering - Northern California Project. Retrieved from https://c.ymcdn.com/sites/www.aspenational.org/resource/resmgr/Techical_Papers/2014_July_TP.pdf

U.S. Department of Energy. (2011, June). *Buildings Energy Data Book, 1.1.3, Buildings share of U.S. primary energy consumption.* Retrieved from EIA, State Energy Consumption Database for 1980-2009, website: http://buildingsdatabook.eren.doe.gov/TableView.aspx?table=1.1.3

U.S. Department of Energy. (2012, January). *Buildings Energy Data Book, 1.1.4, 2010 U.S. buildings energy end-use splits, by fuel type*. Retrieved from EIA, annual energy outlook 2012 early release, website: http://buildingsdatabook.eren.doe.gov/TableView.aspx?table=1.1.4

Vaughn, M. R. (2012, November 12). *Testing and Modeling Energy Performance of Active Chilled Beam Systems.* Retrieved from https://www.ashrae.org/file%20library/doclib/research/2012fallimplementationupdates/1629-ws.pdf

Wang, S., Liu, X., and Gates, S. (2015). *An introduction of new features for conventional and hybrid GSHP simulations in eQUEST 3.7*. Energy & Buildings, 105, 368-376.

Zhu, Y. (2010, October). *Built Environment. (3rd ed.).* Beijing: China Building Industry Press.