Proposed Design for a Coupled Ground-Source Heat Pump/Energy Recovery Ventilator System to Reduce Building Energy Demand

Matthew L. McDaniel

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Francine Battaglia, Chair
Scott Huxtable
Amrinder Nain

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ABSTRACT

The work presented in this thesis focuses on reducing the energy demand of a residential building by using a coupled ground-source heat pump/energy recovery ventilation (GSHP-ERV) system to present a novel approach to space condition and domestic hot water supply for a residence. The proposed system is capable of providing hot water on-demand with a high coefficient of performance (COP), thus eliminating the need for a hot water storage tank and circulation system while requiring little power consumption. The necessary size of the proposed system and the maximum and normal heating and cooling loads for the home were calculated based on the assumptions of an energy efficient home, the assumed construction specifications, and the climate characteristics of the Blacksburg, Virginia region. The results from the load analysis were used to predict energy consumption and costs associated with annual operations.The results for the predicted heating annual energy consumption and costs for the GSHP-ERV system were compared to an air-source heat pump and a natural gas furnace. On average, it was determined that the proposed system was capable of reducing annual energy consumption by 56-78% over air-source heat pumps and 85-88% over a natural gas furnace. The proposed GSHP-ERV system reduced costs by 45-61% over air-source heat pump systems and 52-58% over natural gas furnaces. The annual energy consumption and costs associated with cooling were not calculated as cooling accounts for a negligible portion (6%) of the total annual energy demand for a home in Blacksburg.

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Acronyms

ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CFC	chlorofluorocarbon
COP	coefficient of performance
DR	daily temperature range (°F)
EER	energy efficiency ratio
EES	engineering Equation Solver®
ERV	energy recovery ventilator
FF	fenestration solar load factor
GSHP	ground-source heat pump
GSHP-ERV	ground-source heat pump/energy recovery ventilator
HCFC	hydrochlorofluorocarbon
HDD	heating degree day
HE	heat exchanger
HFC	hydrofluorocarbon
HRV	heat recovery ventilator
HSPF	heating seasonal performance factor
HVAC	heating, ventilation, and air-conditioning
IAC	interior attenuation coefficient
OF	opaque surface-cooling factor
PEX	polyethelene cross-linked
PU	furnace pickup factor
PXI	peak exterior irradiance
RHE	reversible heat exchanger
SHGC	solar heat gain coefficient
WAHP	water-to-air heat pump
WWHP	water-to-water heat pump

Chapter 1 Introduction

1.1. Background

Heat pump systems and energy recovery ventilators are emerging as some of the most popular and efficient ways to heat and cool a residential home. As fuel prices continue to climb and more emphasis is placed on renewable energy systems, many building designers and efficiency experts are turning to these technologies as a means of meeting the heating and cooling demands of a home with a single unit. Both units have proven both cost-effective and reliable over generations of use with little to no maintenance requirements [1].

Due to their minimal use of electricity as a fuel source, heat pumps are considered a clean technology that reduce the demand for fossil fuels and can have a negligible impact in terms of emissions and environmental pollution. Because they transfer heat rather than generate heat, heat pumps can provide up to four times the amount of energy they consume [1]. If coupled with a source of renewable energy, heat pumps can even help produce a net-zero home. Use of more efficient refrigerants and newer techniques such as ground-source exchange, only serve to boost the utility and implementation of heat pumps that were once limited by temperature extremes. Unlike dual-mode air-source heat pumps, which suffer lower performance characteristics at extreme temperatures, dual-mode ground-source heat pumps (GSHPs) use an open or closed heat exchanger system buried underground at a depth where the temperature is relatively constant to transfer energy to and from the home via a reversible vapor-compression cycle.

Energy recovery ventilators (ERVs) are becoming more prevalent as consumers continue to look for ways to simplify heating and ventilation needs while improving performance. These systems make optimal use of a tightly sealed home where ventilation would originally have been an issue [2]. As more and more construction practices focus on limiting the amount of air infiltration, the importance of adequate ventilation for indoor air quality increases. By simply exchanging energy in exhausted air from the home with the intake air for the system, energy recovery ventilators not only reduce the size of a building's heating/cooling system but can even replace it altogether [3].

Both heat pumps and ERVs, individually, have several drawbacks. The main obstacle for both is capital costs, or the cost to purchase and install the system. While both of these technologies have the ability to recover their costs (or "pay for themselves") within a few years due to the energy they save, the initial costs are often too much for many consumers. GSHPs in particular face large capital costs that many consumers often cite as the key barrier to adaptation [4]. Most systems are typically oversized for the necessary applications thus contribute to the high costs. Air-source heat pumps are limited by temperature extremes, especially cold weather, and need to be supplemented by an auxiliary source in regions of extreme heat and/or cold [5]. An ERV is limited by both the exhaust and intake air flows as well as rapid changes in temperature. ERVs can also have difficulty controlling humidity levels in warm weather because they do not have an active form of humidity control. ERVs are reliant on the properties of the air cross-flow and typically do not incorporate elements such as a reheat coil.

1.2. Motivation

As the world population continues to grow and newer, more exciting technologies emerge, man's reliance on the supply of energy is reaching a critical point. Global energy use grew by 2% per year from 1970 to 2002 and 4.1% per year from 2002 to 2005 [6]. Both commercial and residential buildings account for 40% of the energy used in the United States today [7]. It is easy to see that with these unstable trends, the efficiency of many devices must be improved in order to help lower the demand. In 2010, the Virginia Tech Solar Decathlon built the lumenHAUS as a research project in an attempt to demonstrate the concept of a net-zero home [8]. The lumenHAUS is a prime example of efficiency, capable of generating more power in one year than it consumes. The house won the SDEurope 2010 competition with its innovative use of pavilion style architecture, efficiency appliances, and a heating, ventilation, and air-conditioning (HVAC) system that coupled a ground-source heat pump with an energy recovery ventilator [8]. The combination of these two heating and cooling systems gave the house an ultra-low power requirement for interior space conditioning, which allowed more power to be sent to the electrical grid to offset local demand. The system also produced the hot water for both domestic demand and radiant floor heating.

Both heat pumps and energy recovery ventilators are capable of efficiently handling the heating and cooling needs of a structure under moderate temperature conditions. However, once the conditions become extreme these efficiencies decrease [1]. The desire to improve the system capabilities is a large motivating factor for much research. In order to obtain maximum efficiency, a combination of the two systems should be considered. In addition, a coupled ground-source heat pump-ERV (GSHP-ERV) system allows for the waste heat generated during cooling operations to be used to supply hot water to the house, further reducing energy demands.

Cost is a large factor when selecting an HVAC system for a residence. While individually a heat pump setup or an energy recovery setup can have cost benefits and drawbacks, the combination of the two presents more benefits for the consumer. By combining the ground-source heat pump and ERV operations, the size of both the heat pump system and the hot water system are reduced. In addition, proper setup enables hot water generation to be conducted on-demand, and eliminates the need for a hot water tank and circulation pumps typically found in a residence located in the United States. The capital costs may be larger if not competitive (depending on the application) and the return on investment will be larger since the demand for energy is reduced.

1.3. Scope

The objectives of this study are to determine the system size requirements for an energy-efficient home in Blacksburg and to determine the overall energy and annual cost benefits from installing a coupled GSHP-ERV system. The study will present a novel coupled GSHP-ERV design that is capable of meeting space conditioning requirements with high efficiency while also providing domestic hot water heating on-demand. The unique design of the coupled GSHP-ERV system will eliminate the need for a hot water storage tank, reducing overall space requirements. The proposed system will be capable of providing direct hot water on-demand for the home. By producing hot water on-demand, the system will eliminate the need for circulating pumps and a hot water storage tank which will reduce spatial requirements.

Chapter 2 will present an overview of current system designs and concepts and will discuss the types of refrigerants found in heat pump systems. Chapter 3 will cover the fundamental theory of the heat pump vapor compression cycle as well as the theory of energy recovery ventilators. The chapter will also cover the concepts behind waste heat recovery for ground-source heat pumps in cooling mode via a desuperheater. Chapter 4 will cover the proposed design concepts for the proposed combination system of a ground-source heat pump with hot water generation and ERV. The focus will be on the overall design of the system based on code stipulations and desired criteria, and will cover some of the benefits of using this design. Chapter 5 will present the analysis of the proposed design performance from the standpoints of energy and cost. The chapter will also attempt to offer some estimated spatial dimensions for the design. Finally, Chapter 6 will provide concluding remarks and recommendations based on the results.

Chapter 2 Current System Design and Concepts

A review of the current methods used for space heating and cooling will be presented in this chapter. Current system design will be discussed for ground-source heat pumps and energy recovery ventilators, and the type of refrigerant chosen as the working fluid will be presented. Note that more detail will be given to the technical specifics of each unit (such as the vapor compression cycle components and theory, etc.) in Chapter 3.

2.1. Ground-source heat pumps

Ground-source heat pumps (GSHPs) are a type of space conditioning device that use a basic vapor-compression cycle to heat and cool a building [5]. Unlike air-source heat pumps, which use the outdoor ambient air as a sink, a ground-source heat pump exchanges energy in the form of heat with the earth [2]. The earth creates a sink and provides a major advantage for GSHPs because the ground temperature below a certain depth remains constant due to thermal inertia [9]. In contrast, ambient air fluctuates in temperature and therefore air-source heat pumps are limited to climates where the ambient temperature does not experience extreme heat or cold [2]. The constant temperature source that the earth provides helps maintain a relatively constant coefficient of performance (COP) for the heat pump, i.e. the amount of electricity needed to run the system remains constant no matter what the outdoor conditions are on any given day.

Ground-source heat pumps exchange heat with the ground via an open or closed-loop hydronic or refrigerant system [5]. There are three major types of loop systems: open, closed with vertical wells, or closed with horizontal wells. The diagram for a typical ground-source heat pump is shown in Figure 2-1. Open-loop systems, such as the one shown in Figure 2-2, use groundwater or a nearby body of water such as a pond as a source for geothermal exchange.

Water is pumped from the source to the system, where the system exchanges energy with the water then rejects the water back to the source [5].

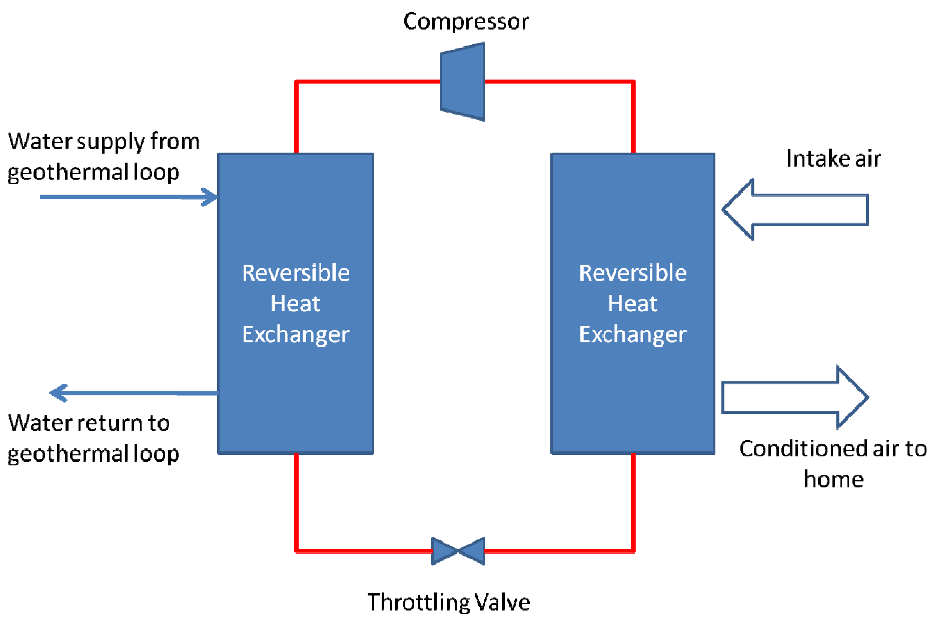

Figure 2-1: Typical ground-source heat pump and its components.

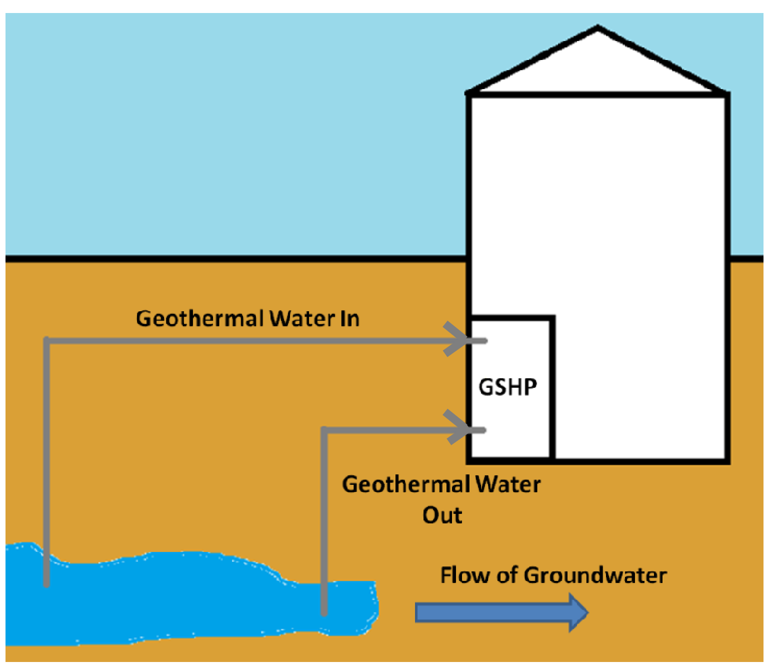

Figure 2-2: Typical GSHP configuration for open-loop systems.

There are two key disadvantages to open-loop systems. First, water from the source is not typically conditioned before entering the pipes, and it contains many particles and contaminants that can build in the pipe over time. These deposits may also be corrosive, leading to a rapid breakdown of the pipe system and possible contamination of the heat pump unit itself. Open-loop systems often require maintenance such as acid flushes or frequent pipe replacement in order to maintain optimal operation. The second disadvantage is location; the system must be close to a sufficient supply of either groundwater or a pond, which presents a major drawback as most people do not readily have access to such resources [5].

Closed-loop systems utilize a buried pipeline to exchange heat with the earth via conduction [5]. As the liquid (e.g., water, antifreeze, or refrigerant) exits the unit, it travels through a series of underground loops buried at a specific depth until it reaches a desired temperature, at which point the liquid is brought back to the system to repeat the process of heat exchange. Note that a closed-loop system can either run the refrigerant line directly through the ground (known as direct exchange) or through a secondary loop, with the primary refrigerant loop located in the appliance [5]. Direct exchange systems use brazed copper tubing that must be protected from corrosion whilst in the ground. While these types of systems experience higher efficiencies with a shorter loop, the amount of refrigerant and the cost of the tubing are higher than required by other systems. Closed-loop and open-loop water systems utilize a secondary loop to exchange heat with the ground [1]. The loop system can either be buried in a series of vertical wells or a horizontal field.

Vertical wells require far less space and little-to-no maintenance [1]. Typically a series of wells are dug to a predetermined depth and then linked together in order to provide the necessary return water temperature. However, this type of configuration is particularly expensive since it involves having a certified well-digging/drilling service install the necessary wells. In addition, the system requires drilling to a specific depth that may be very costly to achieve, such as a home situated on bedrock or in a rocky region [2]. A typical vertical well configuration is shown in Figure 2-3. Horizontal field closed-loop systems are typically the cheapest closed-loop system to install. A wide trench is dug to a specific depth and the field lines are then placed at the bottom of the trench and covered with earth. Again, the system requires little to no maintenance over the lifecycle of the heat pump. While the excavation is typically cheaper than drilling wells, the area required for the field lines can be very large and hence this is not a practical option for those living in more urban areas [2]. A typical horizontal field loop is shown in Figure 2-4. In order to reduce space requirements, some installers will bury the field line as a series of coils. However, this configuration typically suffers from more residual heat transfer because the coils lay on top of each other, causing heat to transfer across the pipes rather than into the ground.

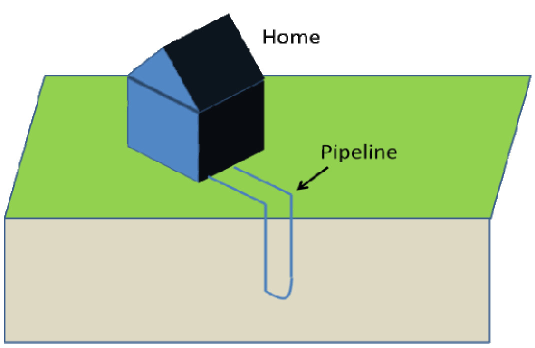

Figure 2-3: Closed-loop system with vertical well configuration.

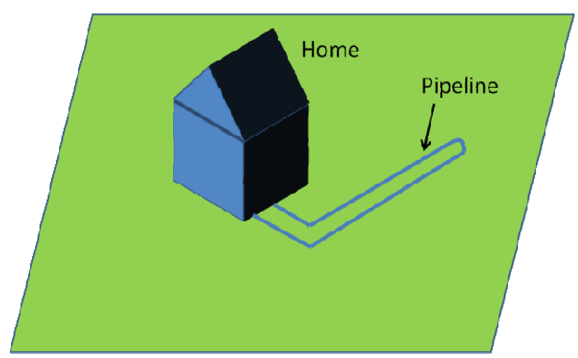

Figure 2-4: Typical horizontal field loop setup.

Ground-source heat pumps have several advantages that other types of heating systems do not. Ground-source heat pumps can provide both forced air and hydronic heating, known as "water-to-air" and "water-to-water" processes, respectively. A forced-air process transfers heat to and from the conditioned air in a residence via the vapor refrigeration cycle [5]. A system that uses a central unit to condition air before being dispersed throughout the house is most common and is often called central air conditioning [5]. Forced air systems can also provide cooling by reversing the heat pump cycle. What makes heat pumps truly unique is that they have the ability to both heat and cool a space [5]. A hydronic heating process uses water to transfer heat throughout the building in the form of baseboard heating, radiators, and radiant floor heating. The system must be sized based on both the building size and the heat delivery method, as some are not as effective as others [1].

One of the key features of a ground-source heat pump is the ability to recover waste heat in order to heat domestic hot water via a desuperheater [1]. The recovery process allows the refrigerant to pass through a separate heat exchanger when the unit is in cooling mode. A desuperheater is a heat exchanger that transfers some of the heat energy being extracted from the conditioned air into the hot water supply for the home, which is particularly efficient in terms of hot water production and can help reduce water heating bills by 25 to 50 percent [2].

2.2. Energy recovery ventilators

Energy recovery ventilators (ERVs) provide a controlled way of ventilating a home while minimizing energy loss [1]. The operation principle is simple: using a heat exchanger, an ERV allows the exhaust air being ventilated from the home to exchange energy with air being drawn into the system. A diagram for an energy recovery ventilator can be found in Figure 2-5. ERVs are especially effective for tightly sealed homes as such structures often require forced ventilation in order to maintain proper indoor air quality [6]. By allowing the expelled air to "pre-condition" the intake air, the amount of work required from the heat pump or other heating/cooling device is drastically reduced. In some cases the need for a separate heating system can be eliminated altogether and the house can run on an energy recovery ventilator alone. However, to run a home completely without an auxiliary heating or cooling source requires significant insulation and rigorous energy practices, and may not be suitable for all regions [3].

Energy recovery ventilators require little maintenance and very little energy to operate. Unlike heat recovery ventilators (HRVs) that simply recover the latent heat associated with the exhaust air, ERVs allow for the transfer of moisture between the air streams, which helps to maintain a better humidity balance in the conditioned space and can also help reduce problems associated with water freezing in the ERV unit [1]. ERVs are best suited for climates that experience extreme winters and summers and have high fuel costs; in mild climates the cost of energy consumed by the system may exceed the energy savings from not conditioning the intake air [1].

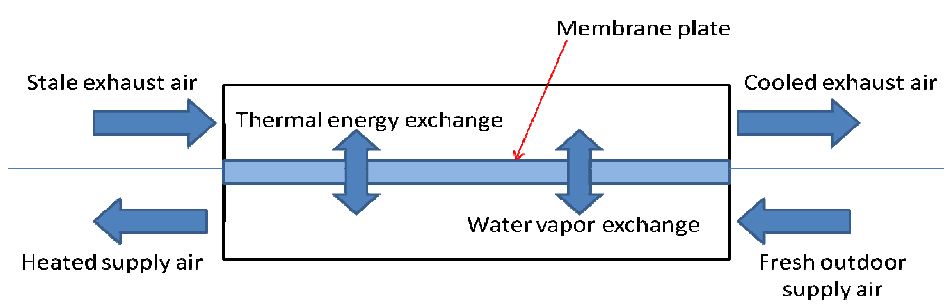

Figure 2-5: Typical ERV setup. Note that the air streams never make contact with each other; only enthalpy is exchanged.

There are several drawbacks to energy recovery ventilators. For example, ERVs can freeze in extremely cold climates if the intake air is not properly conditioned [2]. In hot, humid climates an ERV may not be able to adequately maintain the humidity in a structure [1]. Since the system does not provide an active means of heat removal, the unit may not be able to maintain a comfortable temperature if the surroundings are excessively hot for long periods of time due to internal gains heating, rather than cooling, the house. Finally, if the house is not properly designed and insulated, an ERV will not be able to provide the amount of heating and cooling required to keep the space comfortable and may even lose the ability to maintain a comfortable temperature for the home [3].

2.3. Types of refrigerants

Many heat pumps use a vapor compression refrigeration cycle to transfer thermal energy to and from a space. For this cycle a refrigerant is used as the working fluid. Older designs utilize halocarbons (commonly known as Freon) which contain chlorofluorocarbon compounds, or CFCs [10]. These types of compounds are considered environmentally harmful because upon being released the chlorine attacks the ozone in the upper atmosphere, depleting a critical screen for ultraviolet radiation [11].

Due to the potentially dangerous impacts of CFCs, in the 1970’s there was a shift from CFC groups to HCFC, or hydrochlorofluorocarbon groups. Common refrigerants such as R-22 and R-124 were developed to reduce the impact on ozone depletion but environmental concerns continued to increase. This led to the development of HFCs (hydrofluorocarbons) in the 1990’s such as R-134a, which is still widely used in today’s automobiles [11]. Alternative refrigerants that provide satisfactory performance with minimal impact on the environment are the result of the blending of refrigerants to create mixtures, such as R-410A. There are two types of blended refrigerants: azeotropic and zeotropic. Azeotropes have multiple components but do not change temperature during isobaric phase change (i.e., they have a single boiling point called the azeotropic point), whereas zeotropes experience a shift in temperature during isobaric phase change known as “glide” [11].

Refrigerant R-410A is becoming one of the most prevalent refrigerants in use today based on good performance, while maintaining an environmentally friendly composition. The thermodynamic analyses in this research will utilize R-410A and the licensed software EES® (Engineering Equation Solver) provided by F-Chart Software®, which includes the properties and physical data for many substances including R-410A.

Chapter 3 Design Principles and Theory

The underlying theory behind current and future design principles of this research will be presented in this chapter. The vapor compression cycle will be examined for heat pump operations including ideal designs and actual performance characteristics. The theory behind energy recovery ventilation will be examined with a focus on how it helps improve both air quality and space conditioning performance. Finally, the concepts of waste heat recovery with respect to hot water generation from space conditioning applications will be presented and analyzed.

3.1. Heat pump theory

A heat pump is a machine or device that transfers heat from one location (the 'source') at a lower temperature to another location (the 'sink' or 'heat sink') at a higher temperature using mechanical work or a high-temperature heat source [12]. Unlike furnaces, which must generate heat through a combustion process, heat pumps are designed to transfer thermal energy from a cold source to a hot sink and thus do not generate any energy. The unique design of a heat pump is its ability to change which coil is the condenser and which is the evaporator via a reversing valve.

3.1.1. Overview

The heat pump uses the concepts of the vapor compression cycle to transfer heat from one source to another. Heat pumps exchange energy between the conditioned interior space and either the ground or the air. For the purposes of this study, only ground-source heat exchange is considered as stated in Section 2.1. In heating mode, the coil in the ground loop becomes the evaporator, while the coil in the conditioned interior of the home becomes the condenser, thus absorbing the heat from the refrigerant. The refrigerant absorbs heat from the water in the ground-source loop, is compressed by the compressor, and sent to the evaporator. The refrigerant then rejects heat to the space where it is distributed throughout the home. In cooling mode, the coils are reversed with the conditioned space being the evaporator and the ground-source loop becoming the condenser. Figure 3-1 shows a simplified heat pump system with the primary heat and work interactions between the cycle and the surroundings. The box represents the control volume defined for the vapor compression system. The "hot" and "cold" bodies represent the thermal reservoirs and the arrows indicate the rate of heat transfer $\dot{Q}$ to and from the system as well as the total power $\dot{W}_{cycle}$ required for the cycle.

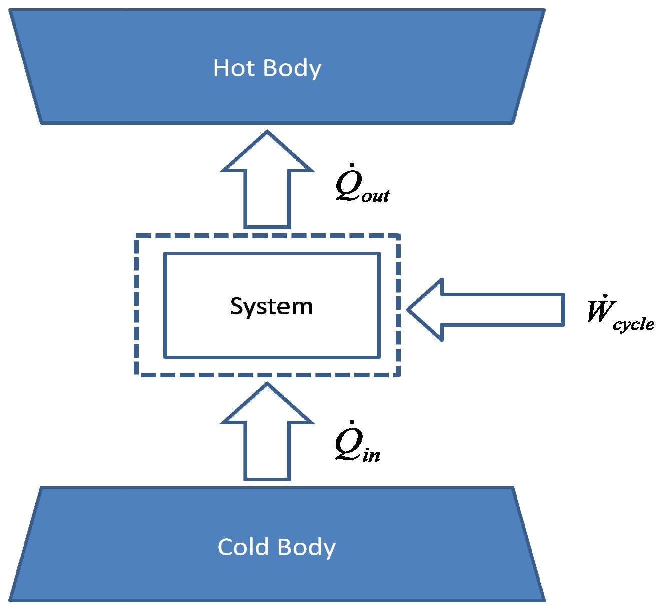

Figure 3-1: Simplified model of a heat pump system.

Based on Figure 3-1, the total work transfer equals the total heat transfer, or

$$\oint \dot{W}_{cycle} = \oint \dot{Q}_{cycle}$$

$$\therefore \dot{W}_{cycle} = \dot{Q}_{out} - \dot{Q}_{in}$$

A coefficient of performance (COP) is established to evaluate the amount of work converted into heat for the two different system operations. For refrigeration, the desired effect is the transfer of heat from the conditioned space (cold body) to the geothermal water exchange loop (hot body), and the coefficient of performance β is defined as:

$$\beta = \frac{\text{heat removed}}{\text{work required}} = \frac{\dot{Q}_{in}}{\dot{W}_{cycle}} = \frac{\dot{Q}_{in}}{\dot{Q}_{out} - \dot{Q}_{in}} \qquad 3.1$$

For a heat pump, the heat transfer $\dot{Q}_{out}$ from the system to the conditioned space (hot body) is desired, and the coefficient of performance γ is:

$$\gamma = \frac{\text{heat added}}{\text{work required}} = \frac{\dot{Q}_{out}}{\dot{W}_{cycle}} = \frac{\dot{Q}_{out}}{\dot{Q}_{out} - \dot{Q}_{in}} \qquad 3.2$$

It is helpful to examine the Carnot cycle and ideal cycles before examining the actual vapor compression cycle. In addition, the cooling cycles can utilize a desuperheater and the effects of this extra component on the system must be analyzed.

3.1.2. Carnot cycle

The Carnot vapor compression cycle is the theoretical system that describes all heat pump cycles. All processes in the Carnot cycle are assumed to be internally reversible (i.e., no losses), as are the isothermal heat transfer processes. The expansion valve is replaced by a turbine because of the assumed reversibility. The components for this cycle are shown in Figure 3-2.

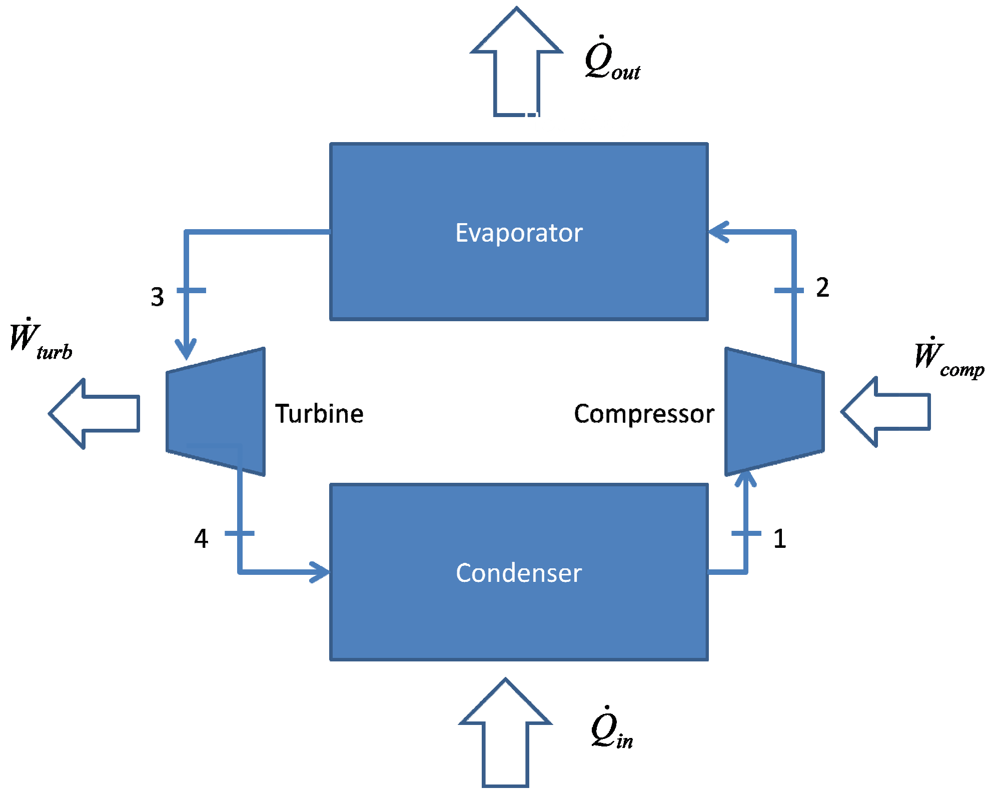

Figure 3-2: Components of the Carnot vapor compression cycle.

The Carnot cycle operates within the vapor dome of the refrigerant, which indicates that the refrigerant is always a saturated mixture. Due to the reversibility and specified operating conditions, the Carnot cycle is considered to be the best possible performance expected from a refrigeration cycle between the hot and cold thermal reservoirs. The *T*-*s* diagram for the Carnot cycle is specified in Figure 3-3. Note that the hot and cold reservoir temperatures are specified at T_h and T_c, respectively.

The refrigerant enters the condenser at State 4 as a two-phase liquid-vapor mixture. Some of the refrigerant changes phase from liquid to vapor as a result of the heat transfer from the cold region at T_c. The temperature and pressure of the refrigerant remains constant throughout the evaporator (from State 4 to State 1). The refrigerant is compressed adiabatically from the two-phase liquid-vapor mixture at State 1 to a saturated vapor at State 2. During this process, the temperature of the refrigerant is increased from T_c to T_h and the pressure increases. The refrigerant then passes through the evaporator where it changes from a saturated vapor to a saturated liquid due to heat transfer to the region T_h. The temperature and pressure remain constant from State 2 to State 3. The refrigerant finally expands adiabatically through the turbine to return to the conditions at State 1.

The coefficient of performance β for the Carnot refrigeration cycle is represented as β_{max} because the Carnot cycle represents the maximum theoretical COP of any refrigeration cycle operating between T_c and T_h [13]. For the Carnot cycle shown in Figure 3-3, the COP β_{max} is:

$$\begin{aligned}\beta_{max} &= \frac{\dot{Q}_{in}/\dot{m}}{\dot{W}_c/\dot{m} - \dot{W}_t/\dot{m}} = \frac{T_c(s_1 - s_4)}{(T_h - T_c)(s_1 - s_4)} \\ &= \frac{T_c}{T_h - T_c}\end{aligned} \quad 3.3$$

Similarly, the COP γ_{max} for the heat pump cycle can also be defined as:

$$\gamma_{max} = \frac{T_h}{T_h - T_c} \quad 3.4$$

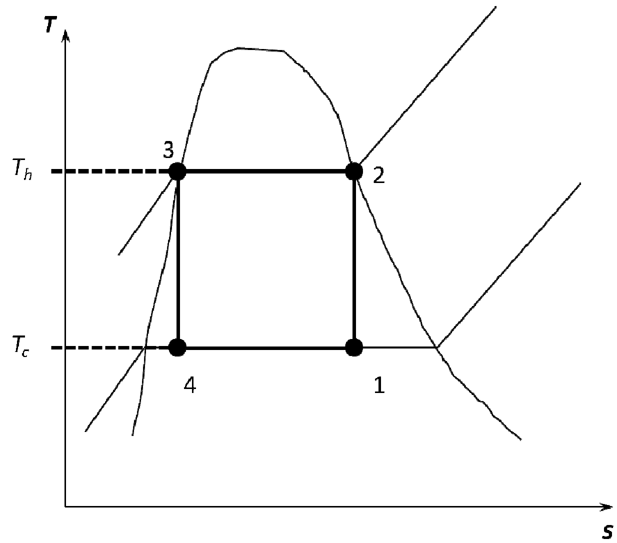

Figure 3-3: T-s diagram for the Carnot cycle.

3.1.3. Ideal vapor compression cycle

Actual vapor compression cycles, including the "ideal" model, depart significantly from the Carnot cycle and thus have lower COPs than those calculated using Equation 3.3 and Equation 3.4. The components of an ideal vapor compression refrigeration cycle are shown in Figure 3-4 and the states of the ideal vapor compression cycle are illustrated in the *T*-s diagram of Figure 3-5. There are several key differences from the Carnot cycle. One of the most significant departures is related to the heat transfers between the refrigerant and the two regions [13]. Actual systems have irreversibilities due to heat transfer and thus are not reversible as in a Carnot cycle. In order to maintain isothermal conditions for the refrigerant in the evaporator, the refrigerant must actually be several degrees below T_c. Likewise the temperature in the condenser must be several degrees higher than T_h. The Carnot cycle compresses a liquid-vapor mixture to a saturated vapor, which is known as wet compression, and is particularly damaging to the compressor unit. To correct this, the ideal cycle compresses a saturated vapor to a superheated vapor to avoid handling a liquid-vapor mixture known as dry compression. Finally, there is a relatively small amount of work generated by the expansion process from State 3 to State 4 compared to the compressor work input. In addition, turbines operating under these conditions typically have low efficiencies and high costs associated with operation. As a result, the turbine is often discarded and replaced with a simple throttling (or expansion) valve.

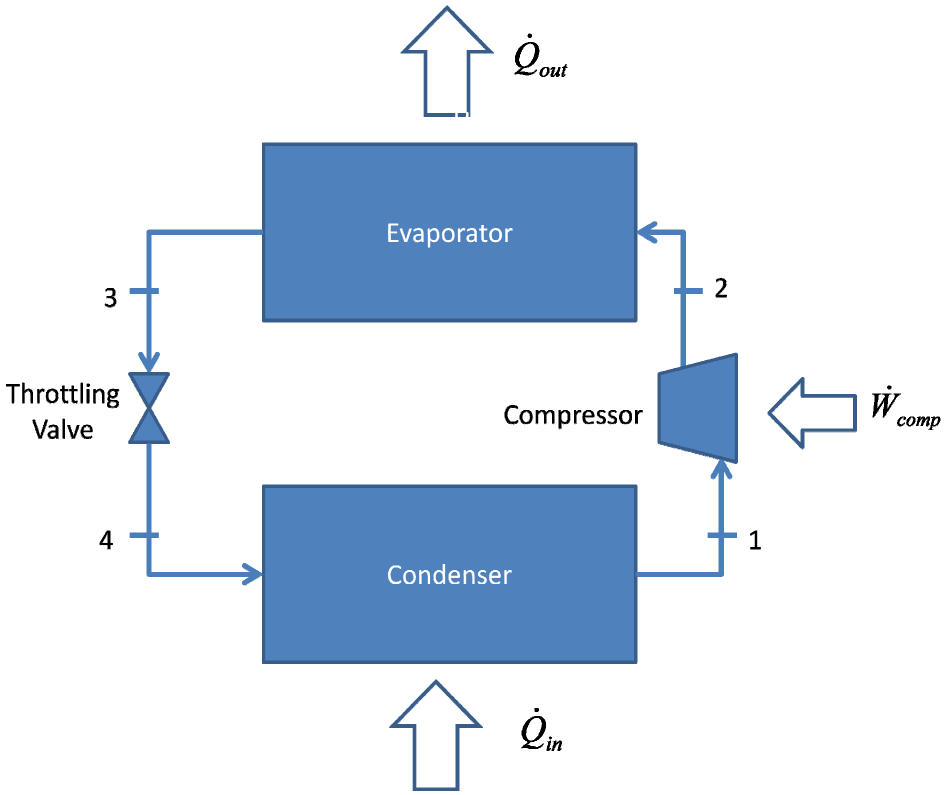

Figure 3-4: Components of the ideal vapor compression refrigeration cycle.

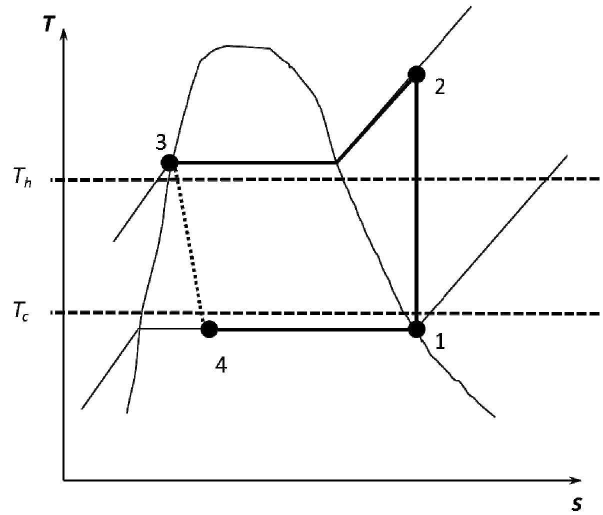

Figure 3-5: T-s diagram for ideal vapor compression cycle.

To calculate the principal work and heat transfers for the cycle shown in Figure 3-4, there are several simplifying assumptions that must be made: (1) the system is operating under steady-state conditions, (2) kinetic and potential energy changes are neglected, and (3) there is a single stream of refrigerant mass flowing through the system.

The analysis starts at the evaporator where the desired refrigeration effect is needed to condition the space. As the refrigerant passes through the heat exchanger the heat transferred to the system causes the refrigerant to vaporize. Reduction of the conservation of energy equations related to the evaporator gives:

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4 \qquad 3.5$$

where $\dot{m}$ is the mass flow rate of the refrigerant. Considering the compressor and assuming adiabatic compression, the work required per mass flow refrigerant is:

$$\frac{\dot{W}_{net}}{\dot{m}} = h_2 - h_3 \tag{3.6}$$

For the condenser, the heat transfer per unit mass flow of refrigerant is:

$$\frac{\dot{Q}_{out}}{\dot{m}} = h_2 - h_3 \tag{3.7}$$

The refrigerant enters the expansion valve, which is modeled as a throttling process, and:

$$h_4 = h_3 \tag{3.8}$$

The coefficient of performance for the system based on Equation 3.1 is:

$$\beta = \frac{\dot{Q}_{in}/\dot{m}}{\dot{W}_{net}/\dot{m}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{3.9}$$

For a heat pump cycle, the transfer of heat to the conditioned space is the desired effect. The COP can be derived based on Equation 3.2 as:

$$\gamma = \frac{\dot{Q}_{out}/\dot{m}}{\dot{W}_{net}/\dot{m}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{3.10}$$

3.1.4. Actual vapor compression cycle

The typical vapor compression cycle has four components: the evaporator, compressor, condenser, and expansion valve. The evaporator heats the refrigerant, which enters as a cold liquid-vapor mixture, until the refrigerant is completely vaporized. The refrigerant vapor then enters the compressor, where it is compressed to a high temperature, high pressure vapor. Next, superheated vapor next travels through the condenser, where heat is rejected to the "hot" body, which causes the vapor to condense to a high pressure liquid-vapor mixture. The liquid-vapor mixture then passes through the expansion valve, where the pressure and temperature are reduced. The liquid-vapor mixture then enters the evaporator and the cycle repeats.

Actual vapor compression cycles feature a few key departures from the ideal cycle discussed previously. First, the heat transfers between the refrigerant and the hot and cold regions are not accomplished reversibly. To account for irreversabilities, the refrigerant temperature in the evaporator must be less than the cold region temperature T_c, and the refrigerant temperature in the condenser must be greater than the warm region temperature T_h [13]. Next, the compression process from State 1 to State 2 is not isentropic. If the refrigeration capacity is to remain the same for both the actual and ideal vapor compression cycles, the work input will be greater for the actual cycle to account for the irreversible compression. Finally, the refrigerant is slightly superheated at State 1 to avoid wet compression and slightly subcooled at State 3 to ensure maximum heat transfer. Figure 3-6 illustrates the differences between the actual and ideal vapor compression cycles via an overlay of each system's T-s diagram. The states for the ideal vapor compression cycle are labeled as 1, 2, 3, and 4. The corresponding states for actual vapor compression are labeled as 1', 2', 3', and 4', respectively.

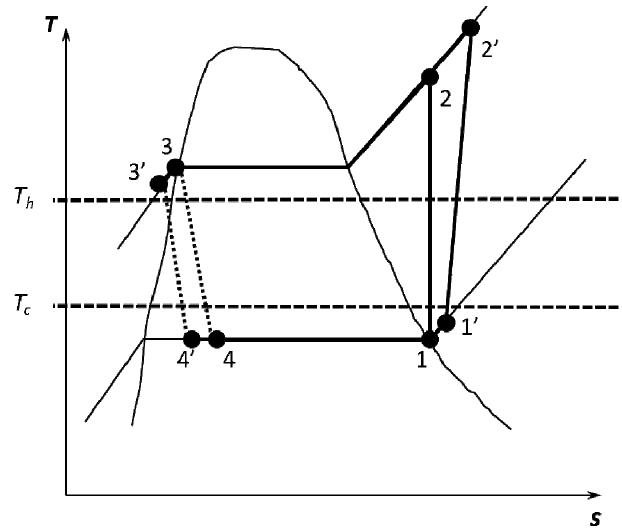

Figure 3-6: T-s diagram showing both ideal and actual vapor compression cycles.

The effect of an irreversible compression process can be accounted for by using the isentropic compressor efficiency for the states in Figure 3-6, which gives:

$$\eta_c = \frac{\left(\dot{W}_{net}/\dot{m}\right)_s}{\left(\dot{W}_{net}/\dot{m}\right)} = \frac{(h_2 - h_1)_{int\,rev}}{h_2 - h_1} = \frac{h_2 - h_1}{h_{2'} - h_{1'}} \qquad 3.11$$

where η_c is the isentropic compressor efficiency and the subscript $int\,rev$ refers to the irreversible process.

3.1.5. Relationship of EER and COP

The energy efficiency ratio (EER) is a measure of a heat pump or air conditioner performance in cooling operations. The EER is a unit ratio defined as:

$$EER = \frac{\text{energy removed}}{\text{work required}} = \frac{\dot{Q}_{in}}{\dot{W}_{net}} \left[\frac{\text{Btu/hr}}{\text{W}}\right] \qquad 3.12$$

The EER is similar to the COP as both are a measure of system output versus system input. However, unlike the COP (which is unit-less), the EER has units of Btu/hr per Watt (W). The EER is an important ratio commonly used to measure and compare efficiencies of air-conditioning systems.

3.1.6. Impacts of a desuperheater

A desuperheater allows a heat pump to take the heat removed during refrigeration operations and transfer it to the domestic hot water system. This reduces the heating demand on the hot water system and improves the efficiency of the system by allowing for the recovery of waste heat that would normally be rejected to the ground-source loop. Normal systems utilize the hot water heater storage tank to dump the recovered heat; however, for the system that will be presented this hot water tank has been eliminated to further improve efficiency by supplying hot water only on demand. An on-demand system removes the hot water tank and circulation pumps in order to reduce energy demand from the system. The effects of a desuperheater, particularly with regard to these applications, will be presented and discussed later.

3.2. Energy recovery ventilators

Energy recovery ventilation (ERV) systems provide a controlled way of ventilating a home while minimizing energy losses [1]. In the winter, an ERV reduces the cost of heating by transferring heat from the warm conditioned air being exhausted from the house to the fresh yet cold supply air. In the summer, the system works in the opposite manner by cooling the warmer supply air via the cool exhausted air from the house. Note that there is a key difference between energy recovery ventilators and heat recovery ventilators. ERVs allow a certain amount of water vapor to transfer between the streams of air in addition to the simple transfer of heat. The transfer of water vapor results in the transfer of both sensible and latent thermal energy, the latter due to the difference in water vapor pressures between the airstreams or between an airstream and a solid surface [5]. The transfer of latent heat allows an ERV to provide a means of humidity control to a space. Allowing the transfer of water vapor also keeps the heat exchanger core warmer, which helps to minimize freezing problems [1].

3.2.1. Types of Energy Recovery Ventilators

The two types of common energy recovery ventilators are membrane plate assemblies and energy wheel assemblies. A membrane plate assembly is a type of fixed-plate heat exchanger that contains a permeable microporous membrane designed to maximize moisture and energy transfer between airstreams while minimizing air transfer [5]. Membrane plate ERVs have no moving parts, a low pressure drop across the exchanger, and relatively low air leakage. They are also compact in design and relatively simple to install. Figure 3-7 shows the general setup for a membrane fixed-plate cross-flow heat exchanger.

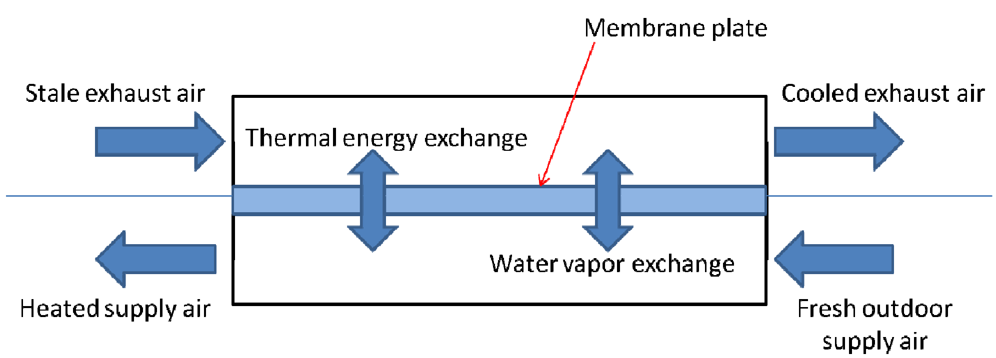

Figure 3-7: Membrane fixed-plate energy exchanger.

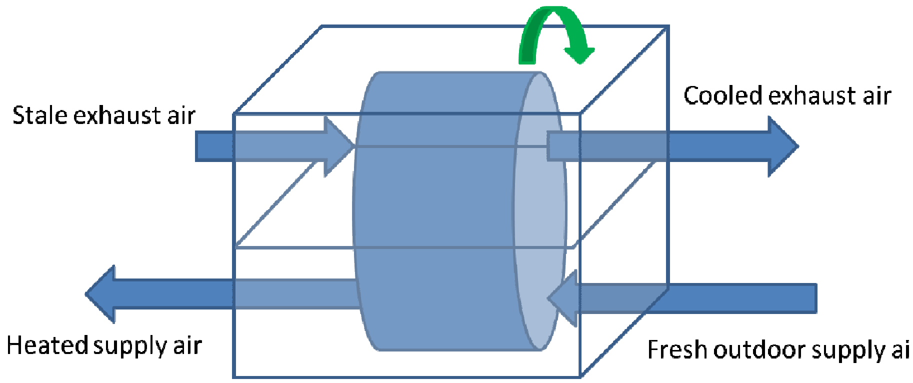

Figure 3-8: Rotary enthalpy wheel energy exchanger.

Energy wheel assemblies (also known as rotary enthalpy wheels) are a type of rotary air-to-air heat exchanger that contains a revolving cylinder with an air-permeable medium having a large internal surface area [5]. Sensible heat is exchanged as the cylinder medium stores heat from the hot airstream and transfers it to the cool one, while latent heat is transferred as the medium absorbs water from the high-humidity stream and desorbs moisture into the lower humidity airstream [5]. Figure 3-8 shows the general setup for a rotary air-to-air energy exchanger. Rotary enthalpy wheels are compact and have a low pressure drop across the exchanger, in addition to having designs that will fit almost any ventilation platform. However, the supply air from an energy wheel system may require additional heating or cooling compared to other systems.

3.2.2. Relations regarding the study of a combined GSHP-ERV system

An ERV system is first and foremost a ventilation device. The incorporation of an ERV allows for the recovery of energy from air that must be exhausted from the house in order to maintain healthy indoor air quality. The ability to consider the ERV part of the ventilation system allows for a whole house analysis approach that deals with heat loss and gain based on insulation values. Chapter 4 will discuss the assumptions and equations used to analyze a residence based on the whole-house approach, including the representation of the ERV system.

3.3. Hot Water Cogeneration/Waste Heat Recovery

The addition of a desuperheater, as mentioned previously, allows for the waste heat from air conditioning processes to be transferred to the domestic hot water system [1]. The recovery of thermal energy reduces the amount of energy required to heat the domestic hot water. Figure 3-9 displays the necessary system setup for the heat pump with the incorporation of a desuperheater, which acts as the condenser for the cycle. The amount of heat transferred is limited by the temperature from the air entering the house. The limited temperature portrays what may be seen as a disadvantage for the combined ground-source ERV system because the ERV is limiting the temperature supplied to the space. However, it is important to remember that the goal of the system is to lower the overall energy demand for space conditioning; a higher temperature would require more work from the heat pump, which would increase the amount of energy needed for the heat pump to maintain the required space temperature.

The cogeneration of hot water could be utilized for other applications besides domestic hot water supply. For instance, if the residence has a pool the waste heat from air conditioning could be dumped into the pool during the summer [1]. The hot water could also be used for outdoor applications where warm water may be desirable, such as bathing the household pet.

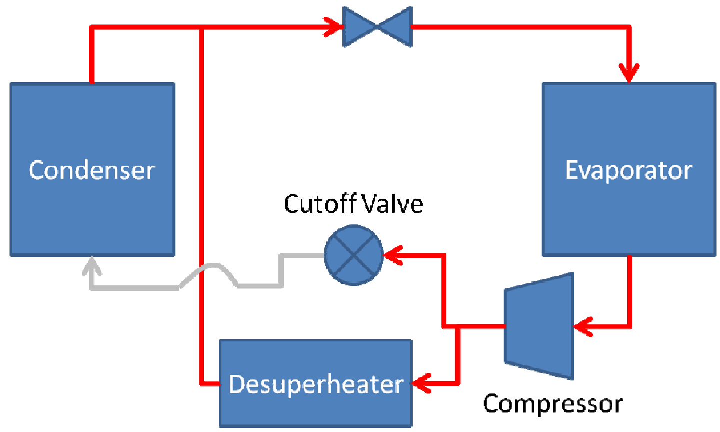

Figure 3-9: Diagram for heat pump unit with the incorporation of a desuperheater.

Chapter 4 Proposed Design Concepts

The theory for sizing a heat pump and the proposed design stipulations for the coupled energy recovery ground-source heat pump will be presented in this chapter. Several benefits and drawbacks to the proposed design are also presented.

4.1. Assumptions and code stipulations

There are several large assumptions that must be made before the heat pump system can be sized. In order to provide maximum efficiency, a space conditioning system must be sized to the specific structure and location. For example, a system designed for a tropical region such as Key West, Florida will not perform well in Minneapolis, Minnesota and vice versa. Thus, it is important to first determine the climate of the region where the system will be implemented in order to design for maximum efficiency. The location of Blacksburg, Virginia is considered for this study due to both familiarity with the region and the range of conditions experienced during the course of a year. Blacksburg experiences warm, humid summers with an average relative humidity of 77% and cold winters with an average temperature of 39°F [14]. The Town of Blacksburg is a terrific example of a region where ground-source heat pumps are ideal because each house requires a large amount of both heating and cooling throughout the year.

Next, the residence itself must be considered. Obviously the amount of heating and cooling will change depending on the size, orientation, and construction of the residence. A house with a large amount of windows on the north face will lose considerably more heat that a house designed for maximum passive solar gains. The assumptions made about the residence have the largest impact on space conditioning demands and are the most important to consider. In light of new building standards and ongoing efforts to reduce energy demands, buildings are becoming more tightly sealed in order to reduce the amount of infiltration [12]. Insulation values are also increasing as a result of both tighter space seals and more advanced materials such as spray foam insulation. In light of new insulation trends the assumptions for this study will assume a well insulated house with an effective distribution system. There are also no attached unconditioned spaces such as a garage or greenhouse. The house that will be used for analysis is based on an efficient home in the Blacksburg area. It will be assumed that the energy efficient home has double pane low-e windows and a well sealed, tightly insulated envelope [6]. Table 4-1 lists the resistance ratings R for each material associated with a specific building component. R-values are a summation of the thermal resistance ratings of each material associated with a specific building component. Manufacturers list the R-value for a material as part of the product description. Also shown is the UA-value for each component, which is a measure of the overall heat transfer coefficient (U) multiplied by the area A of each material. U-values are calculated by taking the inverse of the R-value for each component. By measuring the overall heat transfer coefficient for each area of the home, the rate of heat loss through the structure can be determined.

It is also important to account for infiltration and ventilation for both heating and cooling. Heating conditions deal with the loss of heat to the ambient conditions via infiltration. Table 4-1 also tabulates the effects of infiltration and ventilation on the overall UA-value of the house. Cooling calculations must deal with both latent and sensible heat gains from the ambient air due to the higher humidity conditions. Cooling must also account for solar gains during the hot summer months and internal gains due to occupants. The methods used to calculate heating and cooling loads for the residence are described in the following sections.

Table 4-1: House $\boldsymbol{UA}$-values and associated components used for analysis.

Component	Area (ft^2)	Insulation	R-Value $\left(\frac{ft^2 hr\ °F}{Btu}\right)$	UA-value $\left(\frac{Btu}{hr\ °F}\right)$
Ceiling	1,500	R-47.6 #13	47.6	31.5
Windows	250	Dbl low-e Arg #23	3	83.3
Doors	60	Storm #19	3.8	15.8
Walls	970	24"-o-c 2" ISO #4	34.5	28.1
Floors	1,500	R-21 + 2" ISO #7	37	40.5
	ACH	**Volume**	**Efficiency**	
Infiltration	0.3	12,000	0	64.8
Ventilation	0.3	12,000	70%	19.4
			Total UA-value	283.4

4.1.1. Calculating Heating Loads

Heating mode calculations are easier to determine than cooling loads because heating often deals only with the sensible loads of the structure [12]. The sensible effects of ventilation and infiltration are important to consider and can be determined from blower door tests and manufacturer specifications of the ERV. The sensible effects can be incorporated into the overall UA-value calculation for heating [6]. The calculation of the overall UA-value accounts for the calculation of the maximum system load as a function of interior set point, the 99% design temperature, the distribution efficiency, and the furnace pickup factor. The interior set point T_{set} is the temperature that the heat pump unit must maintain for comfortable temperature levels in the home. The 99% design temperature T_{design} is specified as the coldest ambient temperature for a region where the heat pump must be able to maintain the expected load for the home for 99% of the operating time [6]. The distribution efficiency η_{dist} is used to account for losses associated with moving the conditioned air from the unit to a room within the home via air ducts, plenum space, or another type of distribution system [5]. The furnace pickup factor, PU, is used to slightly oversize a system in order to ensure adequate heat supply and give an extra boost to help heat the space quickly when the unit is initially activated [6]. Note that the maximum system load is not the typical load for the system but is used to account for worst-case scenario operating conditions.

For Blacksburg, the 99% design temperature T_{design} is -5°F and the distribution efficiency η_{dist} for the residence is 90% based on the earlier assumptions of a well insulated home [6]. In addition, the furnace pickup factor PU is defined as 40% above the design load in order to efficiently bring the house to the set point temperature quickly [6]. The interior set point T_{set} is assumed to be 73°F in order to stay within the range of 72-76°F for strict thermal comfort levels [15]. To determine the maximum output for a heat pump system, the required heat transfer $\dot{Q}_{max}$ is [6].

$$\dot{Q}_{max} = \frac{UA_{total}\left(T_{set} - T_{design}\right) \times PU}{\eta_{dist}} \qquad 4.1$$

Inserting the corresponding values from Table 4-1 and the design parameters listed above into Equation 4.1 yields a required output $\dot{Q}_{max}$ of 29,540 Btu/hr. In order for the heat pump to meet the design conditions of Blacksburg, Virginia, the unit must be capable of transferring 29,540 Btu/hr on the coldest of winter days.

On a typical winter day in Blacksburg the ambient outdoor temperature will not reach the worst-case scenario design point of -5°F. Due to the high insulation values, the tight seal on the home, and the use of an energy recovery ventilator, the return air temperature can be used as the reference temperature for the supply of air to the unit [12]. The volumetric flow rate of air is set at a maximum value of 900 ft^3/min in order to ensure adequate air movement. The geothermal airflow rates and temperatures are known from the maximum load conditions, thus a simple energy balance can be used at the condenser (where the supply air is being fed to the house) in order to calculate the amount of heat the system must deliver. An energy balance of the condenser via the First Law of Thermodynamics yields [13]:

$$\frac{dE}{dt} = \dot{Q}_{norm} - \dot{W} + \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} \qquad 4.2$$

The mass flow rates $\dot{m}_{in}$ and $\dot{m}_{out}$ are assumed to be equal and constant because there are no other mass flows associated with the condenser. Losses due to friction and other factors are also neglected. The change in energy over time dE/dt is assumed to be zero because the heat pump system is operating at steady state. The work input $\dot{W}$ is neglected because there is no work associated with the heat exchanger. Air can be assumed to follow an ideal gas model with a constant specific heat capacity, defined as $c_{p,air}$ because the mass flow for the system is equal (i.e., $\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$). Equation 4.2 can be rewritten to determine the system output based on return air temperature T_{return}, supply air temperature T_{supply}, the mass flow of air $\dot{m}_{air}$, and the constant pressure specific heat of air $c_{p,air}$:

$$\dot{Q}_{norm} = \dot{m}_{air}c_{p,air}\left(T_{supply} - T_{return}\right) \qquad 4.3$$

The supply air temperature must account for the distribution system and is thus typically higher than the set point temperature [5]. The supply air temperature is assumed to be 90°F and the return air temperature is assumed to be 70°F. Air can be assumed to follow an ideal gas model for this analysis, thus $c_{p,air}$ is 0.24 Btu/lb°R [13]. The mass flow of air can be determined from:

$$\dot{m}_{air} = \frac{P\dot{V}}{RT_{supply}} \tag{4.4}$$

where P is the pressure (14.7 lbf/in^2 for atmospheric pressure), $\dot{V}$ is the volumetric flow rate in ft^3/min, R is the specific gas constant (approximately 53.3 ft·lbf/lb°R), and T_{supply} is the supply air temperature in °R. Using Equation 4.4, the mass flow rate for air is 1.084 lb/s. Applying Equation 4.3, the furnace output $\dot{Q}_{norm}$ is 18,730 Btu/hr; in other words the heat pump must be capable of delivering 18,730 Btu/hr under normal winter operating conditions in Blacksburg. The normal operating conditions differ from the maximum conditions by approximately 37%, which signifies that the heat pump will be running at approximately 63% capacity when activated during the winter.

The annual heating load and the associated annual costs are functions of the interior set point temperature, the total building UA-value, internal gains from activity and appliances, the number of heating degree days (HDD), the distribution efficiency, the heat pump efficiency, and the cost of electricity. Heating degree days are a measure of how cold and the duration of the heating season is for a given location [6]. First, the balance point temperature must be defined to account for the interior set point and the internal gains:

$$T_{bal} = T_{set} - \frac{\dot{Q}_{int}}{UA_{total}} \tag{4.5}$$

where $\dot{Q}_{int}$ is the internal gain in Btu/hr, either $\dot{Q}_{max}$ or $\dot{Q}_{norm}$. The number of heating degree days must be adjusted to reflect the change in interior set point due to the internal gains [6]:

$$HDD_{T_{bal}} = HDD_{65} - (0.021 \cdot HDD_{65} + 114)(65 - T_{bal}) \tag{4.6}$$

where HDD_{65} corresponds to a heating degree day where the temperature is greater than the reference temperature of 65°F. A temperature lower than 65°F corresponds to a HDD_{65} of 0. The amount of thermal energy annually delivered to the house, $\dot{Q}_{del}$, can then be determined by [6]:

$$\dot{Q}_{del} = 24(UA_{total})\left(HDD_{T_{bal}}\right) \tag{4.7}$$

The amount of fuel energy annually provided, $\dot{Q}_{fuel}$, necessary to generate the thermal energy $\dot{Q}_{del}$ depends on the heat pump efficiency and the distribution efficiency:

$$\dot{Q}_{fuel} = \frac{\dot{Q}_{del}}{\eta_{dist} \times \eta_{pump}} \tag{4.8}$$

where η_{pump} is the efficiency of the heat pump. The annual fuel bill can thus be estimated as [6]:

$$\frac{\$}{yr} = \dot{Q}_{fuel} \times \text{Price} \tag{4.9}$$

The price of fuel is dependent on region and is not a set value. However, electricity is typically cost competitive with natural gas prices and can be driven down to essentially nothing if the power is provided from independent alternative energy sources.

4.1.2. Calculating Cooling Loads

Cooling loads are more complex to calculate than heating loads due to latent effects from humid air [6]. As a result, the cooling load must account for infiltration and ventilation, solar gains, shading, and fenestration effects in addition to the overall building envelope [12]. The residential load factor method, outlined in Chapter 17 of the ASHRAE Fundamentals handbook, can be used to calculate residential cooling loads [12]. The regional design criterion for the assumed house design is outlined in Table 4-2.

It is important to determine the effects of building insulation and ventilation when incorporating an energy recovery ventilator. Mechanical ventilation modifies the infiltration leakage rate for the home [12]. In order to model the effect of mechanical ventilation, the overall supply and exhaust flow rates must be determined and divided into "balanced" and "unbalanced" components [12]. The balanced volume flow rate $\dot{V}_{sup}$ of the air supplied is defined as:

$$\dot{V}_{sup} = \frac{ACH(V)}{60} \qquad 4.10$$

where ACH is the number of air changes per hour and V is the building volume [12]. Inserting the values from Table 4-1 into Equation 4.10 yields a supply flow rate $\dot{V}_{sup}$ of 60 cubic feet per minute (cfm).

Table 4-2: Regional design criteria used for house analysis.

Item	Value	Notes
Latitude	37.24°N	Taken from the National Oceanic and Atmospheric Association
Elevation (ft)	2099	
Indoor temp. (°F)	73	Interior set point temperature
Indoor RH (%)	50	No humidification
Outdoor temp. (°F)	92	Cooling 1% value (12)
Daily Range DR (°F)	17	Collected from ASHRAE Fundamentals handbook (12)
Outdoor wet bulb (°F)	75	
Wind speed (mph)	15	Taken from the National Oceanic and Atmospheric Association
Design ΔT (°F)	19	Determined from indoor temperature and outdoor temperature data.

Similarly, the exhaust flow rate $\dot{V}_{exh}$ is found to be 120 cfm (the ACH value for exhaust is 0.6 from Table 4-1 and the formula for $\dot{V}_{exh}$ is the same as that for $\dot{V}_{sup}$). Both the supply and exhaust flow rates can be used to find the balanced flow rate $\dot{V}_{bal}$ [12]:

$$\dot{V}_{bal} = \min\left(\dot{V}_{sup}, \dot{V}_{exh}\right) \qquad 4.11$$

The exhaust airflow rate and supply airflow rate can also be used to find the unbalance airflow $\dot{V}_{unbal}$ [12]:

$$\dot{V}_{unbal} = \max\left(\dot{V}_{sup}, \dot{V}_{exh}\right) - \dot{V}_{bal} \qquad 4.12$$

Airflow components can be combined with leakage rates to find the combined infiltration/ventilation flow rate $\dot{V}_{vi}$ (12):

$$\dot{V}_{vi} = \max\left(\dot{V}_{unbal}, \dot{V}_{i} + 0.5\dot{V}_{unbal}\right) \qquad 4.13$$

where $\dot{V}_{i}$ is the infiltration leakage rate (in cfm) assuming no mechanical pressurization. The cooling and heating load for the incorporation of an energy recovery ventilator is calculated as [12]:

$$\dot{Q}_{vi,s} = C_s\left[\dot{V}_{vi} + (1-\varepsilon_s)\dot{V}_{bal}\right]\Delta T \qquad 4.14$$

$$\dot{Q}_{vi,t} = C_t\left[\dot{V}_{vi} + (1-\varepsilon_t)\dot{V}_{bal}\right]\Delta h \qquad 4.15$$

$$\dot{Q}_{vi,l} = \dot{Q}_{vi,t} - \dot{Q}_{vi,s} \qquad 4.16$$

where $\dot{Q}_{vi,s}$ is the sensible ventilation/infiltration load, $\dot{Q}_{vi,t}$ is the total ventilation/infiltration load, $\dot{Q}_{vi,l}$ is the latent ventilation/infiltration load, ΔT is the indoor/outdoor temperature difference, and Δh is the indoor/outdoor enthalpy difference. ε_s is the ERV sensible efficiency and ε_t is the ERV total efficiency. C_s is the air sensible factor in Btu/ hr·°F·cfm (1.1 at sea level) and C_t is the air total heat factor in Btu/hr·cfm per Btu/lb enthalpy (4.5 at sea level) and these values must be adjusted for elevation [12]:

$$C_s = C_s[1 - \text{Elevation}(6.8754 \times 10^{-6})]^{5.2559} \qquad 4.17$$

$$C_t = C_t[1 - \text{Elevation}(6.8754 \times 10^{-6})]^{5.2559} \qquad 4.18$$

ε_s is assumed to be 70% and ε_t is assumed to be 60% based on a practical range of current manufacturer specifications [5]. The results of Equation 4.14 through Equation 4.16 yield the cooling load due to ventilation/infiltration $\dot{Q}_{vi,t}$ to be 5,870 Btu/hr for the home used in this study.

Next, the heat load through the walls, floors, and ceiling that must be removed via cooling is determined based on the building opaque surfaces. Opaque surfaces are differentiated from fenestration structures (i.e., windows) due to the materials used for construction. The heating load for an individual opaque surface that must be removed via cooling is:

$$\dot{Q}_{opq} = UA[OF_t \Delta T + OF_b + OF_r(DR)] \qquad 4.19$$

where UA is the UA-value for the building component, DR is the daily temperature range (in Fahrenheit), ΔT is the indoor/outdoor temperature difference, and OF_t, OF_b, and OF_r are opaque-surface cooling factors taken from Table 7 of Chapter 17 of the ASHRAE Fundamentals handbook [12]. OF_t captures the buffering effects of attics and crawlspaces, OF_b represents incident solar gain on a surface, and OF_r captures heat storage effects by reducing the effective temperature difference [12]. The surface cooling factors must be assumed for the heat pump design scenario because the house utilized in this study is theoretical. Surface cooling factors depend on the orientation and surrounding terrain of the house. For instance, if the west side is completely shaded by tall trees and bushes the load factor will be greatly reduced. Conversely, if the house is located in the middle of a field with no surrounding cover the load factors will be very large. OF_t and OF_r values are given in the 2009 ASHRAE Fundamentals handbook [12]. The study assumes the roof is constructed using asphalt shingles, which has an impact on the OF_b value used for calculations. OF_b is calculated from:

$$OF_b = 25.7\alpha - 8.1 \quad 4.20$$

where α is determined based on the surface material (given as 0.85 for asphalt shingles)(12).

Table 4-3 gives an overview of the assumed load factors for the house used in this analysis. The UA-values are taken from the values given in Table 4-1. A key assumption must be made at this point concerning the directional orientation of the house. A southern facing wall will experience more solar gain than the other surfaces, however some buildings are typically oriented to face south in the northern hemisphere in order to increase the effects of solar gains in the winter [3]. Solar gains are desirable in the winter because they can provide a significant source of heating for the home [6]. For this study, it is assumed that 25% of the house walls face south.

Table 4-3: Opaque load factors for cooling calculations.

	OF_t	OF_b	OF_r	UA (Btu/hr°F)	$\dot{Q}_{opq}$ (Btu/hr)
Ceiling	0.62	13.75	-0.19	31.5	702.29
Door, Wall w/ Solar Exposure	1	14.8	-0.36	14.92	413.05
Door, Wall (Shaded)	1	0	-0.36	28.99	373.36
Floor (Crawlspace)	0.33	0	-0.28	40.5	61.16

The fenestration loads (loads through transparent surfaces) can be calculated as a result of several key assumptions: first, 70% of the window area is located on the southern wall of the house due to design criteria based on the amount of southern facing windows being 5-12% of the total floor area [16]; second, all windows except those facing south have some sort of shading characteristic [12]; and third, the windows used for the home are low-e, energy efficient windows based on Table 4-1 with a solar heat gain coefficient (SHGC) of 0.33 [12]. The cooling load rate from fenestration effects on a surface $\dot{Q}_{fen}$ can be calculated as:

$$\dot{Q}_{fen} = UA(\Delta T - 0.46\,DR) + A_{surf}(PXI)(SHGC)(IAC)(FF_s) \qquad 4.21$$

where A_{surf} is the surface area, PXI is the peak exterior irradiance, IAC is interior attenuation coefficient, and FF_s is the fenestration solar load factor [12]. PXI is dependent on the orientation of each surface and is defined by:

$$PXI = T_x E_t \text{ (unshaded)} \qquad 4.22$$

$$PXI = T_x[E_d + (1 - F_{shd})E_D] \text{ (shaded)} \qquad 4.23$$

where T_x is the transmission of the exterior attachments such as screens (assumed to be 1 for the study), F_{shd} is the fraction of fenestration shaded by permanent overhangs, fins or environmental obsticles (assumed to be 0.73) and E_d, E_t, and E_D are the peak total, diffuse, and direct irradiance values taken from the ASHRAE handbook [12].

Additionally, based on handbook examples and information IAC is assumed to be 0.8 as a large fraction of windows have some type of interior shading and FF_s can be taken from given tables based on fenestration orientation [12]. Table 4-4 gives an overview of the factors and cooling loads associated with the fenestration characteristics of the example house used for this study. It is readily apparent that the fenestration loads are greater than the loads due to opaque surfaces, which is to be expected as windows are not as effective from an insulation standpoint compared to walls.

Internal gains, both sensible and latent, will have an effect on the cooling load in the summer. Internal gains are often desirable during the heating season because they reduce the demand for energy by helping to heat the home, but in the summer months the house must be kept cool not hot. The internal sensible and latent gains $\dot{Q}_{ig,s}$ and $\dot{Q}_{ig,l}$ can be calculated as (12):

$$\dot{Q}_{ig,s} = 464 + 0.7\, A_{cf} + 75\, N_{oc} \qquad 4.24$$

$$\dot{Q}_{ig,l} = 68 + 0.07\, A_{cf} + 41\, N_{oc} \qquad 4.25$$

where A_{cf} is the floor area of the home, N_{oc} is the number of occupants (estimated as 4 people), $\dot{Q}_{ig,s}$ is the internal sensible gain, and $\dot{Q}_{ig,l}$ is the internal latent gain.

Table 4-4: Factors and cooling loads associated with fenestration characteristics for example home.

Orientation	*E*		*PXI*	FF_s	$\dot{Q}_{fen}$ (Btu/hr)
North	E_D	26	34.1	0.44	192.1
	E_d	30			
	E_t	56			
South	E_D	68	131	0.47	3,496.1
	E_d	63			
	E_t	131			
East	E_D	177	193.2	0.31	488.4
	E_d	60			
	E_t	237			
West	E_D	177	193.2	0.56	807.1
	E_d	60			
	E_t	237			

The effects of distribution losses and gains must also be examined with regard to cooling load. No duct system can be perfectly sealed and insulated from the elements and the placement of the ducts can have a large impact on the distribution efficiency. If, for example, the duct system is run through the attic, during the summer, heat from the unconditioned space in the attic will warm the conditioned air as the air travels between the heat pump unit and the conditioned room of the house. The distribution losses can be calculated as a function of the total sensible load for the residence [12]:

$$\dot{Q}_{dist} = F_{dl} \sum \dot{Q}_s \qquad 4.26$$

where $\sum \dot{Q}_s$ is the summation of all previously calculated sensible loads and F_{dl} is the duct loss/gain factor. The example house used for this study is assumed to have a highly efficient distribution system, thus F_{dl} for this study is assumed to be 0.05 based on a well-insulated and sealed duct system in a crawlspace [12].

The total cooling load for the system can be calculated as a function of all previous loads. Note that a change to any factor of the house, even simple modifications such as upgrading the insulation or using more efficient appliances that do not radiate a large amount of heat, can have a large impact on the total system load. The total cooling load for the residence is calculated as:

$$\dot{Q}_{tot} = \sum \dot{Q}_s + \sum \dot{Q}_l \qquad 4.27$$

where $\dot{Q}_s$ is the total sensible load including distribution losses and $\dot{Q}_l$ is the total latent load. Separate calculations for the ERV loads are not necessary due to the incorporation of the ERV effects on the total ventilation loads.

4.1.3. Calculation Specifications

The calculations specified in the previous section have several parameters that vary with the design of the home and the surrounding climate specifications. The tables presented in this section are a summary of the values used for this study and serve to provide the vital loads necessary for system sizing.

The calculation specifications for the annual heating load and estimated costs are made based on the previous heat load equations. For the region of Blacksburg, Virginia, the results for annual heating load and estimated costs are summarized in Table 4-5. From the analysis the amount of energy needed for annual heating is 5.2 million Btu/year with an annual cost of $294.

The calculations for cooling load for the home used in this study can be made based on the previously presented equations for cooling load. Table 4-6 gives an overview of the individual and total loads associated with cooling for the home. From the analysis the maximum designed cooling load that the home experiences will be 15,076 Btu/hr. Note that this is not a yearly average but a maximum rate of thermal energy that must be removed from the home at a given time. The maximum load capacity for cooling is lower than the average load capacity for heating, which is an indication that the heating load is the dominant factor for design in the Blacksburg region. The conclusion of a dominant heating load is further justified by the number of heating degree days (5052) compared to the number of cooling degree days (737).

Table 4-5: Specifications and associated calculations for annual heating loads and costs.

Parameter	Value
$\boldsymbol{T_{set}}$	73°F
$\boldsymbol{\dot{Q}_{int}}$	3000 Btu/hr
$\boldsymbol{HDD_{65}}$	5,052 °F-d/yr (6)
$\boldsymbol{HDD_{T_{bal}}}$	4,483 °F-d/yr
$\boldsymbol{\eta_{dist}}$	0.9
$\boldsymbol{\eta_{pump}}$	6.5 (corresponds to γ of 6.5)
$\boldsymbol{\dot{Q}_{del}}$	30.5 million Btu/yr
$\boldsymbol{\dot{Q}_{fuel}}$	**5.2 million Btu/yr**
Cost	**$294/yr**

Table 4-6: Overview of individual and total calculated cooling loads for the home.

Parameter	Value	Parameter	Value
$\boldsymbol{\dot{V}_{bal}}$	60 cfm	$\boldsymbol{\dot{Q}_{opq}}$	1,550 Btu/hr
$\boldsymbol{\dot{V}_{unbal}}$	60 cfm	$\boldsymbol{\dot{Q}_{fen}}$	4,984 Btu/hr
$\boldsymbol{\dot{V}_{vi}}$	90 cfm	$\boldsymbol{\dot{Q}_{ig,s}}$	1,814 Btu/hr
$\boldsymbol{\dot{Q}_{vi,t}}$	5,870 Btu/hr	$\boldsymbol{\dot{Q}_{ig,l}}$	337 Btu/hr
$\boldsymbol{\dot{Q}_{vi,s}}$	2,091 Btu/hr	$\boldsymbol{\dot{Q}_{dist}}$	522 Btu/hr
$\boldsymbol{\dot{Q}_{vi,l}}$	3,778 Btu/hr	$\boldsymbol{\dot{Q}_{tot}}$	**15,076 Btu/hr**

As with the comparison of the maximum heating capacity to the normal operating heating capacity, the normal cooling capacity is less than what is expected during maximum design conditions. Using the same principles outlined in 4.2 to 4.4, and assuming the indoor air return temperature is 78°F and the supply temperature is 68°F, the normal cooling capacity is determined to be 9,366 Btu/hr.

4.2. Proposed design

The proposed system design schematic and the benefits and drawbacks are discussed in this section. The purpose is to define a physical system diagram that illustrates how the proposed system will be assembled for installation. Recall that the goal of this study is to provide a single unit (i.e., one “big box”) that contains the necessary components for energy recovery ventilation, geothermal heating and cooling operation, and hot water generation.

The combination of a ground-source heat pump and energy recovery ventilation was used by the Virginia Tech Solar Decathlon team and was shown to be one of the best ways to condition a residence while maintaining low energy consumption [8]. However, the system used for the solar house was rather large and complex. Figure 4-1 is a picture of the system used for the lumenHAUS® during competition. The system is split into separate units for the water-to-air and water-to-water heat pumps, and the energy recovery ventilator is completely separate from both heat pumps. In addition, there is a hot water storage tank and several circulation pumps that help produce hot water for domestic use.

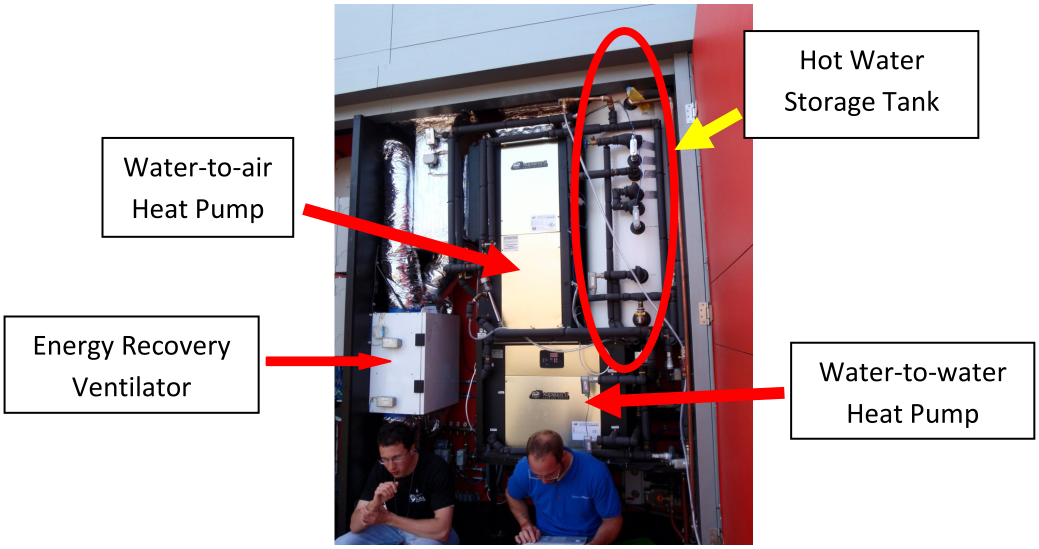

Figure 4-1: Ground-source heat pumps and energy recovery ventilation used by the Virginia Tech Solar Decathlon team.

4.2.1. Overall system diagram and setup

The overall system diagram is shown in Figure 4-2. The figure shows the internal diagram for the system. In other words, the system shown would be housed in a single unit with hookups for the necessary inputs (geothermal supply in, geothermal return, air ducts, etc.). A single unit allows for quick installation and reduces the amount of space required for necessary components. In addition, the system design of Figure 4-2 allows for direct hot water generation which removes the need for a hot water storage tank and associated circulation pumps.

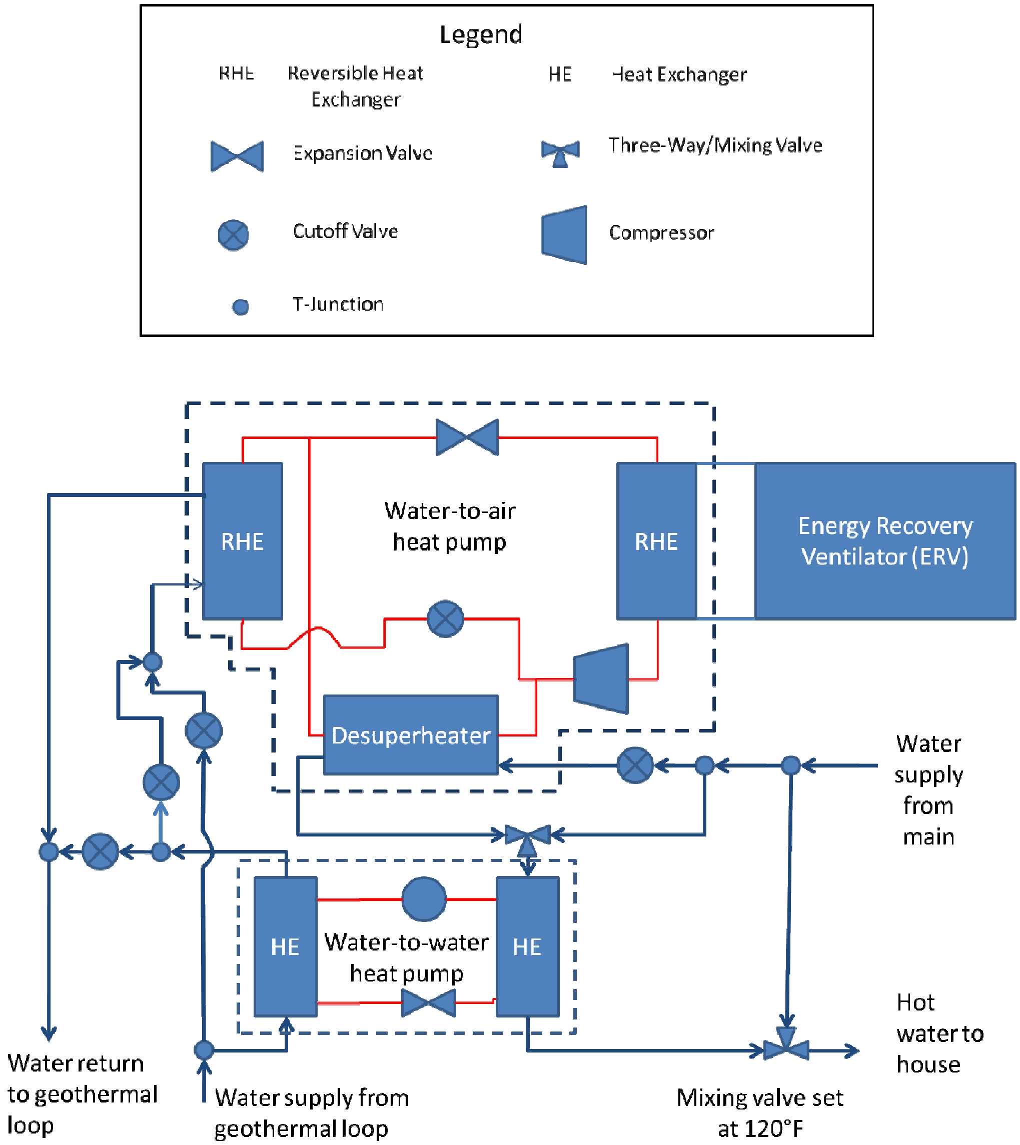

Figure 4-2: System diagram for proposed design of coupled ground-source heat pump and ERV unit.

4.2.2. Benefits

As previously stated, the benefits of the system are the reduced size of the unit and the elimination of the domestic hot water storage system. The reduction in size allows for easy installation and helps the unit to fit into areas where space is critical. Instead of having to occupy all the space in a utility closet, the unit could potentially be placed beside a washer and dryer which would allow for more space in the residence. The simple, one unit design helps with maintenance issues because everything is located within the unit and makes it easier to troubleshoot any errors or potential leaks. The excess polyethelene cross-linked pipe (PEX) and copper piping found in the complex lumenHAUS system is eliminated, with only the minimum amount utilized internally to interconnect the specific heat pump and air conditioning components.

The proposed design is capable of producing hot water on-demand, which reduces the need for a hot water storage tank system. Hot water storage tank systems require circulation pumps, resistance heating within the tank for certain situations, insulation, and thermostats to help keep the water residing in the tank at an acceptable level [1]. If a large amount of hot water is needed (i.e., several residents are taking morning showers, washing hands, doing laundry, etc.) the hot water supply in the tank will quickly be exhausted. In a typical hot water storage tank setup, once the hot water in the tank is exhausted the residents must either wait for the tank to refill and reheat or deal with a lukewarm, and often cold, water supply. In the case of clothes washers, the loss of hot water is unacceptable especially if the unit is attempting to perform a sanitizing cycle. In the case of a loss of hot water, some laundry units are programmed to shut down until hot water can be restored in order to ensure proper clothes treatment.

The proposed unit is capable of providing direct hot water on-demand, which means the hot water supply will theoretically never expire. As long as the water-to-water heat pump (WWHP) has power and is capable of running, the unit will produce hot water for an indefinite amount of time. Residents will not have to worry about losing hot water after extended periods of use. In addition, direct hot water reduces the electrical load for the system by eliminating the need for circulation pumps and the losses incurred from trying to keep a large amount of water at a hot temperature.

The proposed design is also capable of hot water cogeneration, meaning the water-to-air heat pump (WAHP) unit can help provide domestic hot water heating when the WAHP is in cooling mode. When cooling, the heat pump is drawing heat energy from the intake air and rejecting it to the geothermal loop. However, through the use of a desuperheater (a separate heat exchanger within the water-to-air unit) the WAHP can reject the extracted energy to the domestic hot water supply. While the amount of heat rejected from the space conditioning unit to the domestic line is typically not enough to satisfy hot water code requirements, the extra energy does provide a boost in the temperature of the water and reduces the amount of work required by the WWHP. If the residents owned a pool, or for some reason required a warm supply of water to an outside faucet, a separate valve could be installed to pipe the water from the desuperheater to the pool/other supply. A separate valve setup would allow for the recovered hot water energy to be diverted for these uses where hot water requirements are not mandatory. The pool could be heated from the rejected heat due to cooling the house.

The use of an energy recovery ventilator (ERV) lowers the demand on the WAHP by recovering energy from the exhausted air leaving the home. As discussed in Chapter 3, the use of an energy recovery ventilator can reduce the load on the WAHP by pre-conditioning the air entering the system, which lowers the amount of work required by the WAHP to heat or cool the air supplied to the home. In certain situations, the ERV may be capable of providing the space conditioning needs and the WAHP may not even need to operate. Air would simply be conditioned to appropriate levels when entering the system through the ERV.

4.2.3. Drawbacks

One of the drawbacks associated with the proposed combined system is the amount of water required from the geothermal loops in order to provide a sufficient exchange of heat energy. The ability to heat water from supply conditions to 120°F for domestic hot water supply at a volumetric flow rate of 3 gal/min requires a large amount of water from the geothermal wells [17]. While a heat pump is the most efficient way to add or remove heat from the supply water, the required flow rate is still large. The volumetric flow of water from the geothermal loop and the associated effects on the system will be discussed further in Chapter 5.

The geothermal loops require a sufficient amount of land area for installation and can prove quite costly to install. The type of underground setup, i.e., horizontal field loops or vertical field loops, depends on the amount of available land area a resident has for installation. If living in a cramped suburb, a resident is almost forced to go with a vertical installation as the surrounds do not allow for a horizontal field loop setup. Conversely, if the resident is living in a rural area with a large amount of land he/she can elect to go with either a horizontal setup which can drastically reduce the capital costs. Underground obstructions, such as rocks and other obstacles, may add a significant amount of time and capital cost to the installation. If the installation is not done properly the entire system performance will be compromised, as the ability to exchange heat with the earth at a specific depth is the backbone of the geothermal system.

The proposed system operates on a central heating and distribution basis, meaning the heat pump units heat the conditioned air and domestic hot water at a central location in the home and the fluids are then distributed throughout the residence. Central heating and cooling methods may not be as efficient in a large home, where the distance between the central units and the outermost points of demand may be enough to affect the temperature of the disbursed fluid. For example, if the proposed unit were installed in a large home care would have to be taken to ensure that the water lines to the furthest bathroom in the home were run as straight as possible in order to reduce the losses incurred from moving the hot water from the WWHP to the faucet. If the distance is too large, or there are too many twists and turns, the hot water may not be 120°F once it reaches the tap. The lower temperature could be a problem if hot water is required to be 120°F at the point of use. The issue of heat loss through the domestic supply is a separate research issue entirely and thus is not associated with this study.

Chapter 5 Analysis of Design

The analysis and an explanation of the theoretical results for the ground-source heat pump design are presented in this chapter. The energy usage and operating costs are compared to both an air-source heat pump operating under the same conditions and a natural gas furnace for heating.

5.1. Experimental results

The theoretical analysis for the study of the proposed ground-source heat pump system in this study was conducted with the use of Engineering Equation Solver (EES) as well as Microsoft Excel. The proposed design must be able to meet both the maximum capacity operations and normal operation parameters as specified in Chapter 4. The theoretical analysis will demonstrate that the incorporation of a coupled GSHP-ERV can significantly reduce the amount of energy required to heat and cool the example home as well as significantly lower the operating costs compared to other heating and cooling devices.

5.1.1. Maximum capacity

The maximum heating capacity refers to the amount of heat that the geothermal heat pump must be able to output at any given time in order to maintain a comfortable temperature in the residence (e.g., emergency heating mode). From Chapter 4, the maximum heating capacity needed for the home is 29,540 Btu/hr. The maximum capacity conditions determine the volumetric flow rate of water needed from the geothermal wells to the heat pump in order to maintain the design conditions. Without the appropriate volumetric flow rate of water through the geothermal exchange, the heat pump will not be able to transfer enough heat to the water (or draw enough heat from the water) on the geothermal side, which will cause both the space conditioning and hot water supply to fall outside the desired temperature ranges.

The volumetric flow rate from the geothermal loop is used to determine the maximum amount of water demand for the system at a given time, since the operation of heating both the home and the domestic hot water requires geothermal water be supplied at the warmest temperature possible to both units. If the water were to first pass through the WWHP, as it does when the WAHP is in cooling mode, the water supplied to the WAHP would be cold (around 40°F) and thus would not contain a lot of energy for heat exchange. Figure 5-1 shows a schematic for the design scenario of combined space heating and domestic hot water heating. The maximum design condition only applies when both units are running at maximum capacity simultaneously. A typical scenario for this condition would be the WAHP operating in an emergency heat condition (for instance, after the family returned from an evening out and left the heat turned down) while the WWHP is operating at maximum capacity (i.e. washing a load of laundry on sanitize).

The maximum volumetric flow rate of water is highly dependent on the temperature of the water coming from the geothermal wells as shown in Figure 5-2. The temperature change has a greater effect on the necessary flow for the WWHP unit because the WWHP is attempting to deliver a constant flow of water at 3 gal/min (120°F) water to the residence in order to satisfy water supply demands [17]. From Figure 5-2 it is clear that higher water supply temperatures require a lower volumetric flow rate of water from the geothermal wells. It is important to realize that there are practical limits to the maximum temperature. For instance, water can be supplied from a geothermal well at 70°F but the costs of installation from drilling to a sufficient depth (hundreds of feet below the surface) to produce this temperature would be rather large.

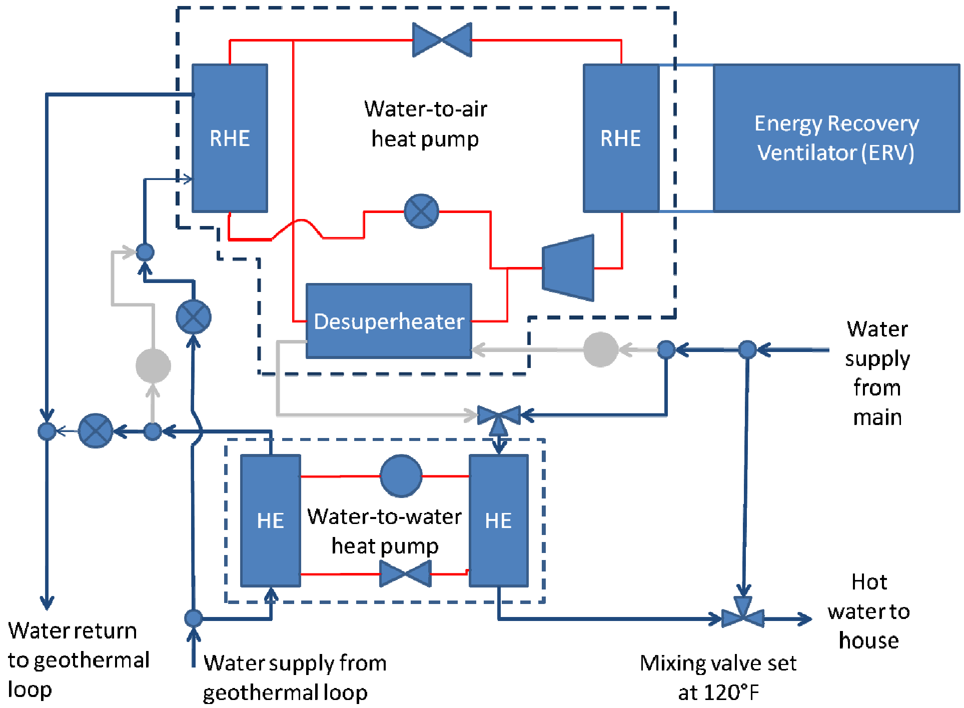

Figure 5-1: Combined system in heating mode. Note the lines in grey are inactive for this schematic (see Figure 4-2 for legend).

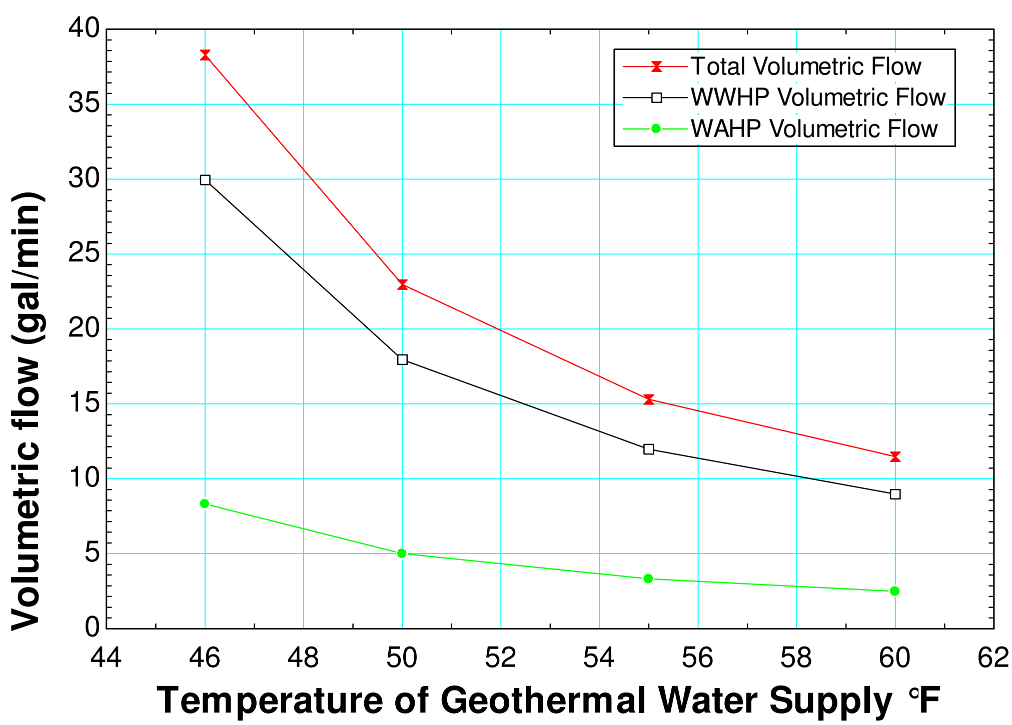

Figure 5-2: Effect of geothermal water supply temperature on the volumetric flow.

Table 5-1 gives an overview of the system performance characteristics found for the proposed design with a geothermal supply temperature of 60°F. The specific temperature of 60°F was chosen because the temperature is still practical to obtain and yields the lowest total flow requirement (approximately 11.5 gal/min). The maximum flow of 11.5 gal/min is large for ¾" inch pipe with a velocity of 8 ft/s, however it is well within acceptable bounds for 1" residential piping with a velocity of 8 ft/s [17]. The total COP, which is a ratio of the heat energy output from both units during maximum heating conditions to the work input for both units per definition in Chapter 3, is approximately 6.8. For the individual heat pump units, the WAHP has a COP of 6.5 and the WWHP has a COP of 6.9. The high COP measurements indicate the incredible efficiency of the system even under the most extreme conditions.

Table 5-1: Results for proposed design for combined maximum heating scenario.

Parameter	WAHP	WWHP
COP	6.5	6.9
Compressor power $\dot{W}$ (kW)	1.32	4.48
Heat output $\dot{Q}$ (Btu/hr)	29,540	105,100
Geothermal Volumetric Flow $\dot{V}$ (gal/min)	2.5	8.8

The maximum cooling capacity is 15,076 Btu/hr as previously calculated in Section 4.1.2. 4.1.1. of Chapter 4. For the maximum cooling scenario, the assumption is made that the WAHP is in cooling mode and the WWHP is turned off. Figure 5-3 is a depiction of the system diagram for this scenario. By turning of the WWHP, the water supplied to the WAHP comes directly from the geothermal wells, which based on previous analysis, stipulates that the temperature of the geothermal water is 60°F. Note that the system diagram for this scenario is the same for WAHP-only operation in heating mode. For both scenarios the WAHP is the only device receiving water from the geothermal wells and transferring water back to the geothermal loops as the WWHP is not in operation. The WWHP switched off corresponds to the normal living conditions when the house needs to be conditioned but there is no hot water needed for domestic tasks.

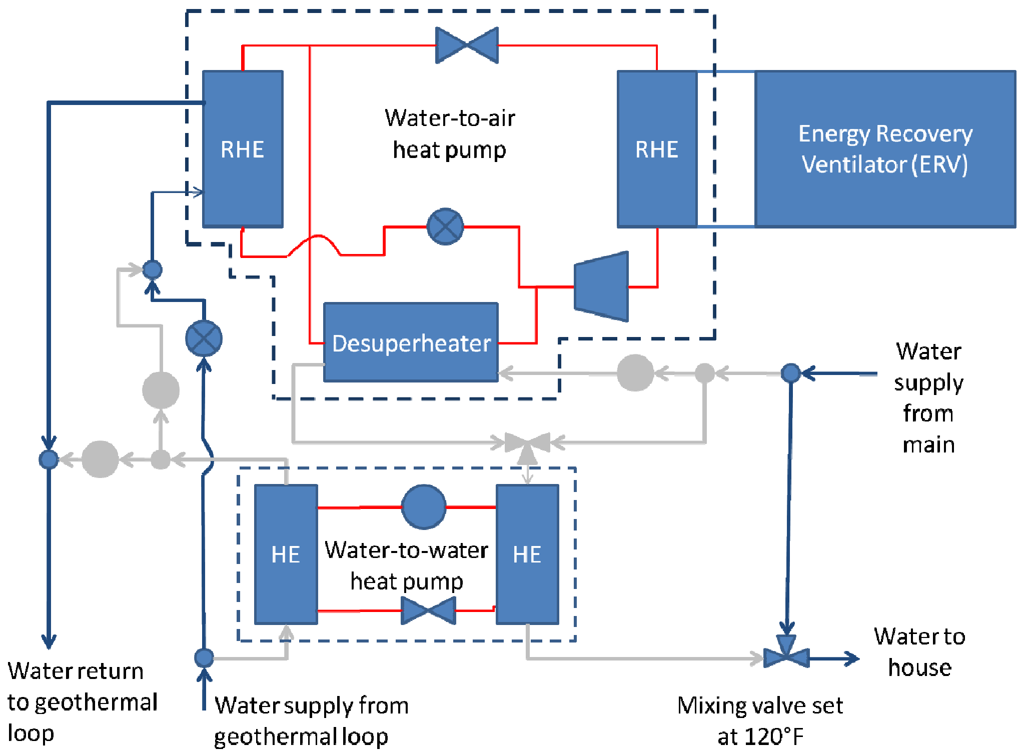

Figure 5-3: System diagram for WAHP only operation with the WWHP switched off. The system setup is the same for both heating and cooling scenarios with the WWHP not in operation (see Figure 4-2 for legend).

Table 5-2 gives an overview of the performance characteristics for the proposed design in the maximum cooling scenario. The proposed system has an energy efficiency ratio (EER) of 25, meaning for every 25 Btu/hr of energy removed from a space the system requires 1 Watt of power. It is important to realize that the required flow from the geothermal wells is equal to the total flow of the system at maximum capacity. The system is sized to provide a maximum of

11.5 gal/min when both units are switched on, thus when one unit is switched off the mass flow rate must be conserved which forces all the water to pass through the WAHP. If two separate geothermal wells were used, one well could be sized to accommodate the WWHP and one well could be sized to accommodate the WAHP. It is rare that a system would use two separate wells because the installation of one simple loop is often more practical and requires less calculation and work (and thus less cost).

Table 5-2: Proposed system performance for maximum cooling scenario with WWHP turned off.

Parameter	WAHP
EER (Btu/hr/W)	25
Compressor power $\dot{W}$ (kW)	0.60
Heat removal $\dot{Q}$ (Btu/hr)	15,076
Geothermal Volumetric Flow $\dot{V}$ (gal/min)	11.5

If the WWHP is switched on while the WAHP is in use, the system simply routes the domestic water supply through the desuperheater and deactivates the controls for the geothermal exchange on the WAHP. An example of a scenario when the WAHP is in cooling mode while the WWHP is on correlates to a resident taking a shower in the middle of the day during the summer, when the air in the home must be kept cool. Figure 5-4 shows the system diagram for combined operation when the WAHP is in cooling mode and the WWHP is operating. The use of the desuperheater allows for the energy rejected from the cooling process to precondition the water for domestic heating which reduces the demand on the WWHP.

Table 5-3 gives the results for the combined system with the WAHP in cooling mode. It should be documented that the use of the desuperheater, in combination with the total flow of 11.5 gal/min flowing through the WWHP, allows the water exiting the WWHP in the geothermal loop to have a temperature of 47°F. The rise in temperature was anticipated since the system is attempting to perform at an optimum level; if the outlet temperature on the evaporator were fixed for this analysis the system would be over-constrained. Considering the compressor for the WWHP needs to have 5 kW of capacity, the resulting 3.8 kW of demand for the combined system is practical and within the limits of the maximum design. When the system is operating only the WWHP, which corresponds to the air unit being switched off while the hot water is in use, is considered a normal operating mode for the WWHP and will be discussed at length in the following section.

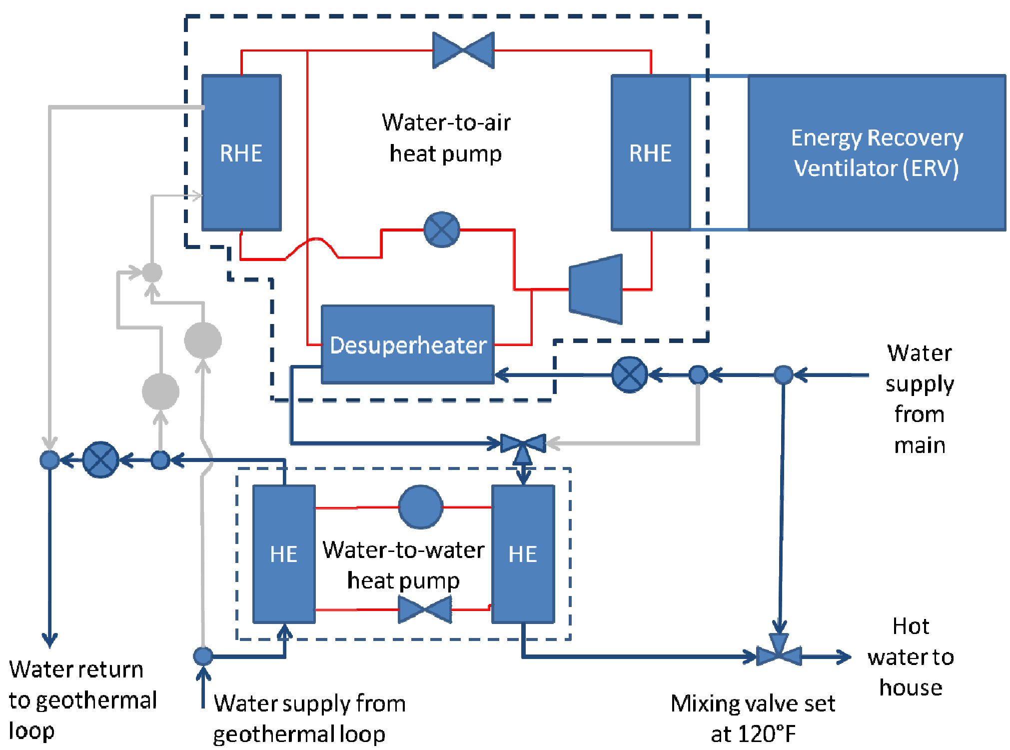

Figure 5-4: System diagram representing the combined system in cooling mode. The desuperheater is used to divert energy from the WAHP to the WWHP (see Figure 4-2 for legend).

Table 5-3: Proposed system results for combined operations with the WAHP in cooling mode under maximum capacity.

Parameter	WAHP	WWHP
COP	-	6.7
EER	25	-
Compressor power $\dot{W}$ (kW)	0.6	3.83
Heat output $\dot{Q}$ (Btu/hr)	-	87,900
Heat removal $\dot{Q}$ (Btu/hr)	15,076	-
Geothermal heat exchanger outlet temperature (°F)	-	47

5.1.2. Normal operation capacity

As previously stated in Section 4.1.1. , the system will only operate at maximum capacity in emergency situations. For the majority of annual operations the system will need to satisfy the demands for normal heating and cooling capacity calculated in Section 4.1.1. and 4.1.2. First, it is important to consider the operation of the WWHP while the WAHP is not in use. An example of the scenario would be a resident taking a shower while the space conditioning is turned off. Figure 5-5 shows the system diagram for the operation of the WWHP while the WAHP is not in operation. The impact of having the WAHP unit switched off will lower the amount of preheated water on the domestic side but will increase the flow to the WWHP geothermal exchanger on the ground-source side of the heat pump. Table 5-4 tabulates the results for the WWHP normal operation scenario. The geothermal heat exchanger outlet for the WWHP has risen to approximately 44.4°F as expected. The WWHP is able to maintain impressive performance statistics, with a COP of approximately 6.9 and a required compressor power of 4.46 kW (roughly the same specifications as for the combined performance for the WAHP-WWHP heating scenario).

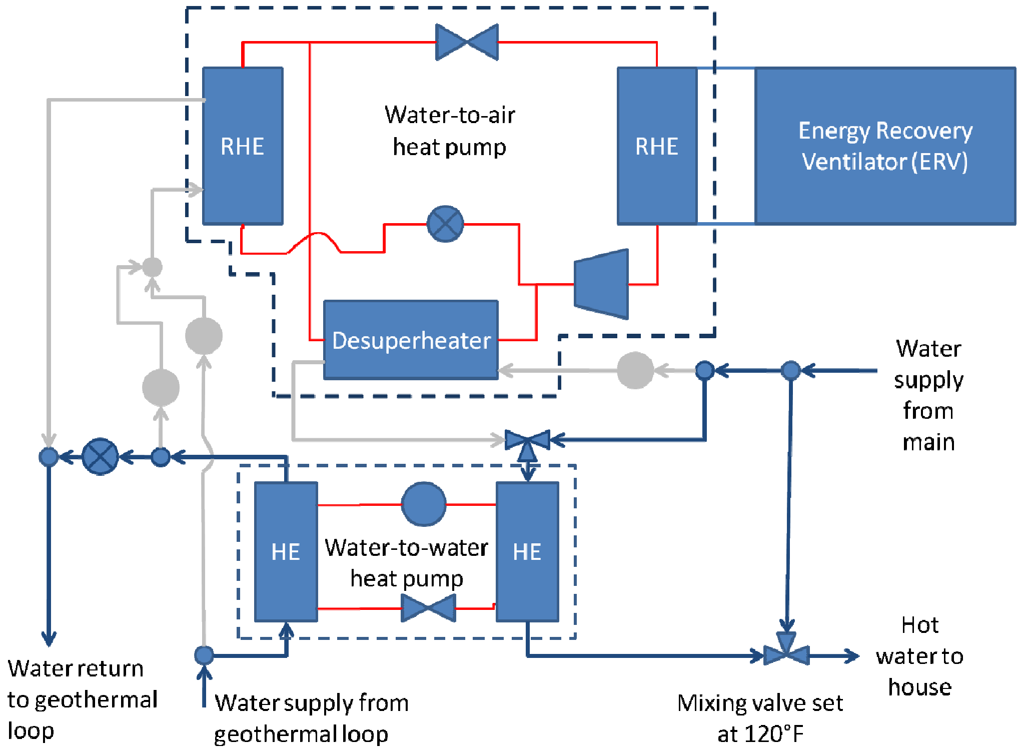

Figure 5-5: System diagram showing the operation of the WWHP with no other components in use.

Table 5-4: Performance characteristics for WWHP under normal operating conditions.

Parameter	WWHP
COP	6.9
Compressor power $\dot{W}$ (kW)	4.46
Heat supplied $\dot{Q}$ (Btu/hr)	105,100
Geothermal Volumetric Flow $\dot{V}$ (gal/min)	11.5
Geothermal heat exchanger outlet temperature (°F)	44.4

From the previous results outlined in Section 4.1.1. , the normal operating capacity for heating is 18,730 Btu/hr. The inputs to the proposed system model are adjusted to reflect this updated capacity of 18,730 Btu/hr and the analysis for the combined system is run to determine the performance based on normal conditions. The schematic for combined WAHP and WWHP heating under normal conditions is identical to Figure 5-1. Table 5-5 presents results for the system performance regarding combined operations in heating mode for normal conditions. The volumetric flow rates for the geothermal lines will be the same as those used for the maximum heating calculations of the combined system. While the geothermal heat exchanger for the WWHP has an outlet temperature close to the original value for the maximum heating mode (40°F), the temperature for the outlet of the WAHP geothermal heat exchanger has risen from 40°F to approximately 47.5°F due to the reduced demand on the system. The rise in temperature corresponds to the earlier assumption that fixing the temperature at this point would over-constrain the design.

Table 5-5: Performance characteristics for the proposed system in heating mode for normal conditions.

Parameter	WAHP	WWHP
COP	6.0	6.9
Compressor power $\dot{W}$ (kW)	0.9	4.43
Heat output $\dot{Q}$ (Btu/hr)	18,740	105,100
Geothermal heat exchanger outlet temperature (°F)	47.5	40

The normal operating capacities for cooling will also have an effect on the energy demands and the geothermal return temperatures for the system. The normal cooling capacity is determined to be 9,366 Btu/hr from the analysis conducted in Chapter 4. It is important to again emphasize that the maximum cooling load capacity is less than the normal operating capacity for heating by approximately 9,400 Btu/hr. Following the same approach outlined for analysis of the combined system as shown in Figure 5-3, the performance characteristics for the WAHP can be determined for the normal load conditions while the WWHP is not in use. Table 5-6 shows the performance results for the proposed system with the WAHP operating under normal cooling conditions. The required compressor power has dropped from 0.6 kW to 0.48 kW, demonstrating the minimal demand that the cooling scenario has on the energy effects of the system.

Table 5-6: System performance characteristics for normal cooling loads with the WWHP not in use.

Parameter	WAHP
EER (Btu/hr/W)	20
Compressor power $\dot{W}$ (kW)	0.48
Heat removal $\dot{Q}$ (Btu/hr)	9,400
Geothermal Volumetric Flow $\dot{V}$ (gal/min)	11.5

The normal operating capacity will also affect the combined system performance for cooling (see Figure 5-4). The lower amount of heat removal for normal operations will not transfer as much energy to the domestic hot water via the desuperheater, thus the WWHP will have to increase the compressor work in order to maintain a high COP. Table 5-7 shows the results for the proposed system for the combined mode with the WAHP in cooling mode under normal conditions. Both the WAHP and the WWHP are able to maintain high efficiencies while the compressor input for the WWHP increases to 4.13 kW, as compared to the compressor input of 3.83 kW for the combined system in cooling mode under maximum conditions. The resulting compressor power is still lower than the required power for the normal operation of the WWHP (4.46 kW) without the contributions from the desuperheater, showing that even the smallest cooling load for the WAHP will have an impact on the performance of the WWHP. Note also that the geothermal heat exchanger outlet temperature has dropped to 46°F, which indicates that more energy was removed from the water in the geothermal exchange than under maximum cooling load conditions (where the temperature only dropped to 47°F).

Table 5-7: System performance for the combined operation with the WAHP in cooling mode under normal conditions.

Parameter	WAHP	WWHP
COP	-	6.7
EER	25	-
Compressor power $\dot{W}$ (kW)	0.37	4.13
Heat output $\dot{Q}$ (Btu/hr)	-	94,400
Heat removal $\dot{Q}$ (Btu/hr)	9,366	-
Geothermal heat exchanger outlet temperature (°F)	-	46.1

5.1.3. Justification for neglecting ERV effects on COP and EER analysis

The analysis conducted for the WAHP and WWHP excluded the effects of the ERV, as the ERV was previously incorporated into the calculations for the load demand. However, the ERV does require power in order to operate the ventilation fans and maintain a balanced flow. Several design standards indicate that the power required to heat a home are minimal when only an ERV is used [3]. Additionally, more and more designs are turning to energy recovery ventilation as the units are simple to operate yet have a large return with regard to the reduction of energy used for space conditioning [18]. For the purposes of this study, the effects of the ERV on energy demand can be neglected for the system COP and EER studies.

5.1.4. Summary of maximum and normal operating conditions

The results for the analysis concerning the heating and cooling performance for the proposed coupled GSHP-ERV system are summarized in this section. The goal is to provide a quick reference table that can be used to analyze and compare all the studied scenarios. Table 5-8 presents a summary of the results found for each scenario outlined in the previous sections of this study. The results clearly show that the system is well designed and capable of meeting the heating and cooling demands for the reference residence. The WWHP has a COP of approximately 7 for each scenario, and the WAHP has a COP of approximately 6.5 for heating scenarios and an EER of approximately 20-25 for cooling scenarios. For most operations, the EER of the WAHP in cooling mode is approximately 20, corresponding to the reduced cooling load under normal operation. The combined system in cooling mode under normal conditions experiences an increase in the EER, up to 25, due to the change in the water conditions entering the desuperheater (which is acting as the condenser for this scenario). The volumetric flow and temperature for the desuperheater under normal conditions is 3 gal/min and 50°F, respectively, which is a significant change from the geothermal conditions of 11.5 gal/min and 60°F.

The maximum capacity scenarios with regard to heating are a reflection of just how well the system can perform under a heavy load. As long as a residence has electricity, the system will be capable of performing as efficiently as it would under normal cooling day conditions. The performance is drastically different from an air-source heat pump, which on the coldest winter days suffers from a decrease in the COP due to the environmental conditions [9]. It is important to note that the heat pump system is entirely dependent on the availability of electricity. If power loss is a frequent and unavoidable scenario for a specific region, i.e., a home that is surrounded by a heavily wooded area and experiences frequent and intense storms, a homeowner should consider having other forms of emergency heat to condition the home in the event of a power outage. Blacksburg is a region that is subject to inclement weather, particularly snow, during certain times of the year thus it would be a prudent measure for a residence equipped with the proposed system to consider a form of backup heat or power generation to account for periods with a loss of power.

Table 5-8: Summary table for analyzed conditions regarding proposed design.

Scenario	Parameter	WAHP	WWHP
Maximum capacity, combined system heating mode	COP	6.5	6.9
	Compressor power $\dot{W}$ (kW)	1.32	4.48
	Heat output $\dot{Q}$ (Btu/hr)	29,540	105,100
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	2.5	8.8
Maximum capacity, WAHP-only cooling mode	EER (Btu/hr/W)	25	-
	Compressor power $\dot{W}$ (kW)	0.60	-
	Heat removal $\dot{Q}$ (Btu/hr)	15,076	-
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	11.5	-
Maximum capacity, combined system cooling mode	COP	-	6.7
	EER	25	-
	Compressor power $\dot{W}$ (kW)	0.6	3.83
	Heat output $\dot{Q}$ (Btu/hr)	-	87,900
	Heat removal $\dot{Q}$ (Btu/hr)	15,076	-
	Geothermal heat exchanger outlet temperature (°F)	-	47
Normal capacity, combined system heating mode	COP	6.0	6.9
	Compressor power $\dot{W}$ (kW)	0.9	4.43
	Heat output $\dot{Q}$ (Btu/hr)	18,740	105,100
	Geothermal heat exchanger outlet temperature (°F)	47.5	40
Normal capacity, WAHP-only cooling mode	EER (Btu/hr/W)	20	-
	Compressor power $\dot{W}$ (kW)	0.48	-
	Heat removal $\dot{Q}$ (Btu/hr)	9,400	-
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	11.5	-
Normal capacity, combined system cooling mode	COP	-	6.7
	EER	25	-
	Compressor power $\dot{W}$ (kW)	0.37	4.13
	Heat output $\dot{Q}$ (Btu/hr)	-	94,400
	Heat removal $\dot{Q}$ (Btu/hr)	9,366	-
	Geothermal heat exchanger outlet temperature (°F)	-	46.1
Normal capacity, WWHP only heating mode	COP	-	6.9
	Compressor power $\dot{W}$ (kW)	-	4.46
	Heat supplied $\dot{Q}$ (Btu/hr)	-	105,100
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	-	11.5
	Geothermal heat exchanger outlet temperature (°F)	-	44.4

5.2. Estimated cost and energy savings

The estimated cost and energy savings are covered in this section. As previously shown in Section 5.1.2. , the maximum capacity load for cooling is approximately 9,400 Btu/hr less than the normal capacity load for heating. For Blacksburg, space heating demands also account for 47% of the total energy consumed by a home in one year, whereas air-conditioning accounts for roughly 6% of the total energy consumed by a home in one year [19]. Based on the low percentage of cooling demand throughout the year for Blacksburg, cost and energy savings analysis will only be presented for heating demand.

Energy and cost savings between the WAHP of the proposed system and air-source heat pumps will be presented, as will energy and cost savings between the WAHP of the proposed system and natural gas furnaces. Current analysis from the Blacksburg Climate Action Plan will be used to determine the annual heating energy and costs related to air-source heat pumps and natural gas furnaces [19].

5.2.1. Energy and cost savings related to air-source heat pumps

Air-source heat pumps represent an efficient way to heat and cool a home in one singular unit. The Blacksburg Climate Action Plan numbers regarding air-source heat pump performance for Blacksburg must be updated to reflect the energy efficiency of the home characteristics used for this analysis. The equations presented in Chapter 4 (particularly Equation 4.8) can be updated to reflect the performance of an air-source heat pump by adjusting the efficiency rating for the heat pump to correspond with the heating seasonal performance factor (HSPF) of a listed air-source heat pump. The HSPF is a measure of an air-source heat pump efficiency rated over the entire heating season to account for fluctuations in minimum temperature. For the Climate Action Plan, the HSPF for a normal heat pump was assumed to be 6.5 and the HSPF for a highly-efficient air-source heat pump was assumed to be 9.75 [19]. The HSPF correspond to an efficiency of 190% (or a COP of 1.9) for the normal air-source heat pump and 285% (or a COP of 2.85) for the highly-efficient air-source heat pump. The cost of electricity was given to be \$0.12/kW·h [19]. The annual energy and cost related to the operation of a heat pump are determined using Equation 4.5 through Equation 4.9 from Chapter 4.

Table 5-9 presents data for the annual energy and cost savings between the proposed system, the maximum air-source efficiency rating, and a normal air-source heat pump for Blacksburg, Virginia. The proposed system can reduce the energy consumption for the home used in this study by 56% and can reduce the annual cost by 45% when compared to a maximum efficiency air-source heat pump used for the same residence. The proposed system provides a 71% energy reduction and a 61% cost reduction when compared to a normal air-source heat pump for the residence. Note that this analysis is used to reflect annual performance and does not account for individual day-to-day performance of a system. As previously discussed in Section 2.1, air-source heat pumps suffer greatly during cold weather conditions whereas geothermal systems performance remains relatively constant [2].

Table 5-9: Energy and cost comparison for the proposed system, the maximum efficiency air-source heat pump, and a normal air-source heat pump.

Type	Energy Used (million Btu/yr)	Energy Used (kW·h)	Annual Cost
Geothermal	5.2	1,526.7	$294
Air Max	11.9	3,481.9	$531
Air Normal	17.8	5,222.6	$743

5.2.2. Energy and cost savings related to other forms of space heating

The information from the proposed Blacksburg Climate Action plan can also be adapted to show the effects of installing the proposed system over choosing to install a natural gas furnace [19]. For the study, a normal natural gas furnace was assumed to be 80% efficient, while a maximum efficiency unit was assumed to be 95% efficient, and the cost of natural gas was assumed to be $14 per million Btu [19]. The calculations were conducted under the same principles used to calculate the cost and energy savings of the proposed system related to air-source heat pumps.

Table 5-10 provides the cost and energy savings of the proposed system related to using a natural gas furnace to meet heating demands for the home characteristics of the study in Blacksburg, Virginia. The proposed design can save 85% of the energy used for a natural gas furnace with maximum efficiency and 88% of the energy used by a normal natural gas furnace. The proposed system can save 52% of the annual cost compared to a furnace with maximum efficiency and 58% of the annual costs compared to a normal natural gas furnace. In addition, the total energy consumption for the WAHP for the proposed design is estimated to be approximately 1,530 kW·h for the entire heating season.

Table 5-10: Energy and cost savings for the proposed system compared to natural gas heating.

Type	Energy Used (million Btu/yr)	Energy Used (kW·h)	Annual Cost
Geothermal	5.2	1,526.7	$294
Natural Gas Max	35.7	10,445.8	$607
Natural Gas Normal	42.3	12,404.2	$701

5.2.3. Summary of energy and cost savings related to space conditioning

For the proposed coupled ERV-GSHP system, the amount of energy predicted for an annual heating season is 1,530 kW·h, which at a cost of \$0.12/kW·h results in an annual fuel bill of \$294. The proposed system can reduce annual costs by 45-61% and can reduce annual energy consumption by 56-88%. Figure 5-6 shows the energy savings, in kWh, for each analyzed system as a side-by-side comparison and Figure 5-7 shows the cost savings in a side-by-side comparison for each analyzed system for further reference.

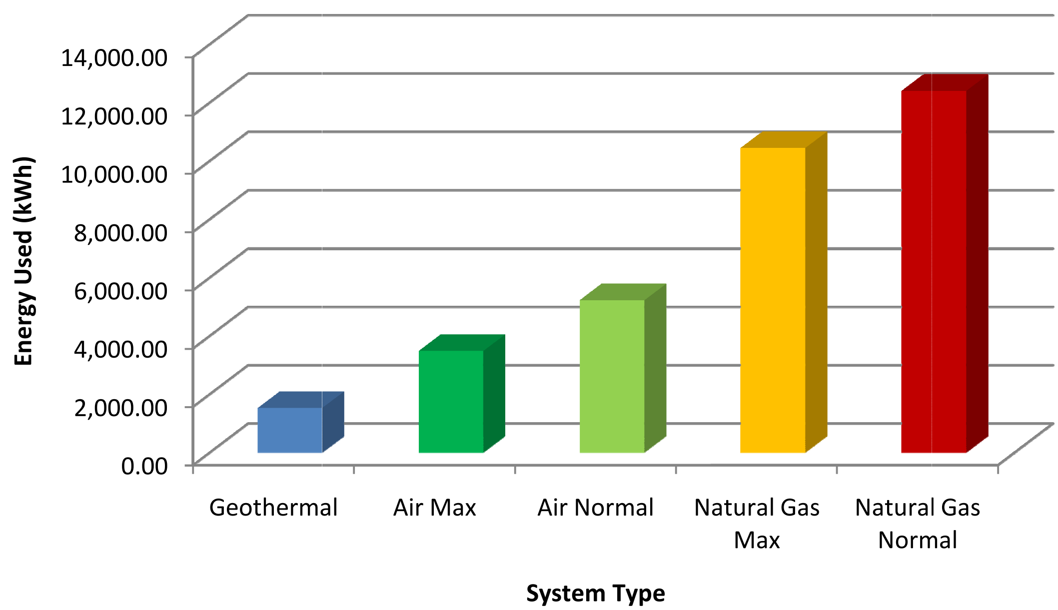

Figure 5-6: Comparison of annual energy used by the proposed system and the systems outlined in the proposed Blacksburg Climate Action Plan (adjusted to the home used in this study).

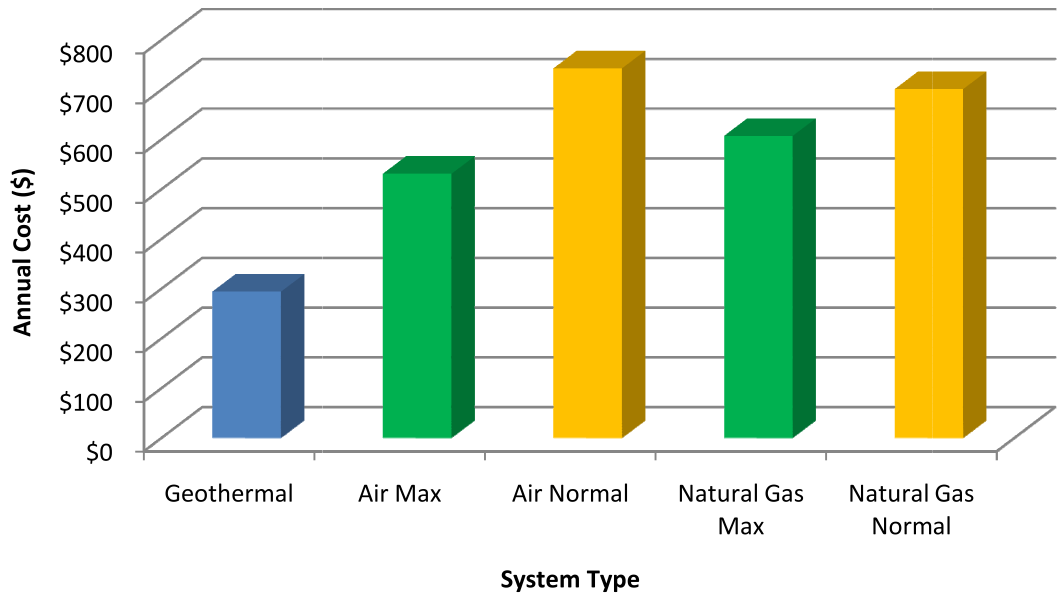

Figure 5-7: Comparison of annual costs of fuel for the proposed system and the systems outlined in the proposed Blacksburg Climate Action Plan (adjusted to the home used in this study).

It is important to notice that the proposed heat pump system runs on electricity, which opens up the opportunity for the power supplied to come from renewable resources. The installation of a PV system or a residential wind turbine could lower operating costs even further, or eliminate the cost altogether. Residential renewable energy is dependent on the resident capability to afford such a system and thus is not widely adapted [6]. However, more financial incentives are becoming available to help reduce capital costs for renewable systems. For example, the proposed Climate Action Plan for Blacksburg includes an outline for a low-interest loan system that could help offset the initial cost and make renewable energy more affordable [19]. It is important for future studies to account for the adaptation of residential renewable energy and the associated impacts on annual fuel costs.

Chapter 6 Conclusions

6.1. Summary

The work presented in this thesis focused on reducing the energy demand of a residential building by using a coupled GSHP-ERV system. The proposed GSHP-ERV system was based on previous work with the Virginia Tech lumenHAUS systems and presented a novel approach to space condition and domestic hot water supply for a residence. Unlike most current geothermal systems, which focus on space conditioning, the proposed system is capable of providing hot water on-demand, thus eliminating the need for a hot water storage tank and circulation system.

Basic heat pump theory was presented alongside the basic principles for energy recovery ventilation in order to provide background for the concepts that inspired the proposed GSHP-ERV system. An energy efficient home located in Blacksburg, Virginia was used to predict the necessary size of the proposed system. The maximum and normal heating and cooling loads for the energy efficient home were calculated based on the climate characteristics of Blacksburg, Virginia and the construction specifications for the assumed home.

The results from the load analysis of the proposed GSHP-ERV system were used to predict energy consumption and costs associated with annual operations. The results for the predicted heating annual energy consumption and costs for the GSHP-ERV system were compared to an air-source heat pump and a natural gas furnace. On average, it was determined that the proposed system was capable of reducing annual energy consumption by 56-78% over air-source heat pumps and 85-88% over a natural gas furnace during the heating season. The proposed GSHP-ERV system reduced costs by 45-61% over air-source heat pump systems and 52-58% over natural gas furnaces during the heating season. As discussed in Section 5.2, the annual energy consumption and costs associated with cooling were not calculated as cooling accounts for a negligible portion (6%) of the total annual energy demand for a home in Blacksburg.

6.2. Recommendations for future study

A purely theoretical approach was used for this study as a first step towards designing a new energy efficient heat pump system. While the results of this analysis are still practical, the assumptions associated with the construction of the energy efficient home have a large impact on system sizing for a heat pump. If the proposed design were adopted for mass production, several models would need to be developed for different size homes. An analysis similar to this study would need to be performed for each different case in order to determine optimum system size as each home has different construction characteristics. In addition, the analysis of the effects of building insulation and construction on the GSHP-ERV system would need to be analyzed. For example, a small house with little insulation may require the same size heat pump system as a large home with a large amount of insulation and energy efficient appliances. A separate set of criteria may need to be developed in order to provide quick reference for system size based on home size and calculated total UA-value.

Finally, as this is a study on reducing the energy demand for space conditioning and domestic hot water use, the effects of passive and active solar designs should be considered. Well-designed passive solar homes can significantly reduce the energy demands associated with space heating and cooling. Passive solar hot water systems, such as a thermosiphon, can provide a large amount of hot water to a residence with absolutely no electrical demand. For certain climates, evaporative cooling can also play a huge role in reducing or eliminating the energy demand associated with cooling. The effects of a hybrid system that combines one or several of these intriguing passive solar designs could further reduce annual energy consumption and costs and may even prove to be cheaper to implement.

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Appendix A Proposed System Results for 50°F

The results of an analysis for a geothermal supply temperature of 50°F are shown in this section. The mass flow of water was allowed to change in order to achieve similar performance characteristics to the proposed system described in this research. If the mass flow of water was not allowed to change (i.e., if the mass flow was held constant at 11.5 gal/min), the decrease in geothermal supply temperature from 60°F to 50°F would decrease the COP and EER of the system. More work would be required to compensate for the loss of the difference in temperature; there is more energy in a temperature change from 60°F to 40°F then there is in a temperature change from 50°F to 40°F. Thus in order to provide a system with a COP and EER that is comparable to the analysis of this research, it is important to allow the volumetric flow of water to vary. Table A-1 shows the effects of a geothermal supply temperature of 50°F for the proposed GSHP-ERV system.

It is important to note that the increase in the required geothermal supply source volumetric flow is not practical. The conditions of 60°F were chosen because the required volumetric flow of 11.5 gal/min maximum was well within pressure range specifications for 1” diameter piping commonly found in plumbing applications. A required flow of 23 gal/min, such as the necessary flow for the 50°F geothermal supply temperature case, would require 1-½” diameter piping in order to meet safety regulations [17].

Table A-1: Proposed system performance for a geothermal supply temperature of 50°F.

Scenario	Parameter	WAHP	WWHP
Maximum capacity, combined system heating mode	COP	6.5	6.8
	Compressor power $\dot{W}$ (kW)	1.32	4.46
	Heat output $\dot{Q}$ (Btu/hr)	29,540	105,100
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	5.0	23.0
Maximum capacity, WAHP-only cooling mode	EER (Btu/hr/W)	25	-
	Compressor power $\dot{W}$ (kW)	0.60	-
	Heat removal $\dot{Q}$ (Btu/hr)	15,076	-
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	23	-
Maximum capacity, combined system cooling mode	COP	-	6.8
	EER	25	-
	Compressor power $\dot{W}$ (kW)	0.6	3.83
	Heat output $\dot{Q}$ (Btu/hr)	-	87,900
	Heat removal $\dot{Q}$ (Btu/hr)	15,076	-
	Geothermal heat exchanger outlet temperature (°F)	-	43
Normal capacity, combined system heating mode	COP	6.3	6.9
	Compressor power $\dot{W}$ (kW)	0.9	4.46
	Heat output $\dot{Q}$ (Btu/hr)	18,740	105,000
	Geothermal heat exchanger outlet temperature (°F)	43.7	40
Normal capacity, WAHP-only cooling mode	EER (Btu/hr/W)	25	-
	Compressor power $\dot{W}$ (kW)	0.37	-
	Heat removal $\dot{Q}$ (Btu/hr)	9,400	-
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	23	-
Normal capacity, combined system cooling mode	COP	-	6.7
	EER	25	-
	Compressor power $\dot{W}$ (kW)	0.37	4.0
	Heat output $\dot{Q}$ (Btu/hr)	-	94,400
	Heat removal $\dot{Q}$ (Btu/hr)	9,366	-
	Geothermal heat exchanger outlet temperature (°F)	-	50
Normal capacity, WWHP only heating mode	COP	-	6.9
	Compressor power $\dot{W}$ (kW)	-	4.43
	Heat supplied $\dot{Q}$ (Btu/hr)	-	105,100
	Geothermal Volumetric Flow $\dot{V}$ (gal/min)	-	23
	Geothermal heat exchanger outlet temperature (°F)	-	42.2

Appendix B Programs Used for Analysis

The programs outlined in this appendix provide the structure used for estimating the performance of the proposed GSHP-ERV system. Each program has values that can be adjusted based on the system operating mode. For instance, if the WAHP is operating while the WWHP is operating, the outputs from the WAHP cooling calculation can be used as the domestic supply conditions for the WWHP. Each program should be adjusted based on the necessary operating mode.

In addition, several of the values are adjusted via a parametric table in EES®. Parametric tables allow for quick iteration for a single program. For example, the calculations for the combined system in heating mode have parametric table values for the geothermal supply temperature that range from 45-60°F. The outputs for each run of the table can be used to generate graphs such as Figure 5-2.

```
"Calculations to determine needed output for system at maximum capacity"

HDD65=5052                                      "degF*day/yr, from regional data"
UA_efficient=284                                "Btu/hr degF, approximately 141.26 Watts, from
house parameters"
eta_distribution=0.9
q_int=3000                                      "Btu/hr"
T_setpoint=73                                   "degrees F"
PickupFactor=1.2
T_design=-5                                     "degrees F"

T_bal=T_setpoint-q_int/UA_efficient
HDD_T_bal=HDD65-(0.021*HDD65+114)*(65-T_bal)    "degF*day/yr"
Q_delivered=24*UA_efficient*HDD65/(10^6)        "Million Btu/yr"

PumpOutput=UA_efficient*(T_setpoint-T_design)*(PickupFactor/eta_distribution)          "Btu/hr"
PumpOutput_kW=PumpOutput/3414       "kW"
```

```
"Combined System, Heating"



"Water-to-air heat pump"

"Ground source properties"
P_wat=60                                            "psi"

"T_watIN"                                           "Parametric value from 45-60 deg F"
T_watOUT=40                                         "deg F"

h_watIN=ENTHALPY(Water, P=P_wat,T=T_watIN)          "Btu/lb"
h_watOUT=ENTHALPY(Water, P=P_wat,T=T_watOUT)
"Need to find m_dot"

"House Supply Properties"
Q_max=29536                                         "Btu/hr from Calculations to Determine System
Output"

"Superheating and subcooling"
degSH=10                                            "deg F per the ASHRAE manual"
degSC=10                                            "deg F per the ASHRAE manual"

"Compressor efficiency"
eta_c=0.8

"State 1"
T_1=T_watIN+degSH                                   "deg F, allows for superheating of refrigerant.
Note that Tsat is around 5
P_1=70                                              "psi"
h_1=ENTHALPY(R410A,P=P_1, T=T_1)
s_1=ENTROPY(R410A,P=P_1, T=T_1)
"x_1=1"                                             "Two-phase is not practical here! The pressure
would have to be

"State 2"
P_2=P_3
s_2s=s_1
T_2s=TEMPERATURE(R410A,P=P_2,s=s_2s)
h_2s=ENTHALPY(R410A,P=P_2,s=s_2s)

"Accounting for compressor efficiency"
h_2=((h_2s-h_1)/eta_c)+h_1
s_2=ENTROPY(R410A,P=P_2,h=h_2)
T_2=TEMPERATURE(R410A,P=P_2,h=h_2)

"State 3"
T_3=63                                              "accounts for room temperature set as 73 deg F"
x_3=0
h_3=ENTHALPY(R410A,T=T_3,x=x_3)
P_3=PRESSURE(R410A,T=T_3,x=x_3)

"State 3 has subcooling"
T_3p=T_3-degSC
```

```
h_3p=ENTHALPY(R410A,T=T_3p,P=P_3)
s_3=ENTROPY(R410A,T=T_3p,P=P_3)

"State 4"
h_4=h_3p
P_4=P_1
T_4=TEMPERATURE(R410A,P=P_4,h=h_4)
s_4=ENTROPY(R410A,P=P_4,h=h_4)

"Energy balances"
mdot_ra=(Q_max/3600)/(h_2-h_3p)                      "lb/s"
mdot_geoa=mdot_ra*(h_1-h_4)/(T_watIN-T_watOUT)       "lb/s"
VolFlow_geoa=mdot_geoa*0.01602/0.13368*60            "gal/min"
CompWorkA=mdot_ra*(h_2-h_1)*3600                     "Btu/hr"
COP=(Q_max)/(CompWorkA)
CompWorkA_kW=CompWorkA/3414                          "kW"



"Water-to-water heat pump"

"House Supply Properties"
T_sIN=50                                             "deg F"
T_sOUT=120                                           "deg F"
VolFlow=0.006684                                     "ft^3/s, corresponds to 3 gal/min"
mdot_sup=1/0.01602*VolFlow                           "lb/s, make sure to check if valid!"

h_sIN=ENTHALPY(Water, P=P_wat,T=T_sIN)
h_sOUT=ENTHALPY(Water, P=P_wat,T=T_sOUT)

"State 1"
T_5=T_1
P_5=P_1
h_5=h_1
s_5=s_1
"x_5=1"                                              "Two-phase is not practical here! The pressure
would have to be

"State 2"
P_6=P_7
s_6s=s_5
T_6s=TEMPERATURE(R410A,P=P_6,s=s_6s)
h_6s=ENTHALPY(R410A,P=P_6,s=s_6s)

"Accounting for compressor efficiency"
h_6=((h_6s-h_5)/eta_c)+h_5
s_6=ENTROPY(R410A,P=P_6,h=h_6)
T_6=TEMPERATURE(R410A,P=P_6,h=h_6)

"State 3"
T_7=60                                               "accounts for entering water temperature at 50
deg F"
x_7=0
h_7=ENTHALPY(R410A,T=T_7,x=x_7)
P_7=PRESSURE(R410A,T=T_7,x=x_7)
```

```
"State 3 has subcooling"
T_7p=T_7-degSC
h_7p=ENTHALPY(R410A,T=T_7p,P=P_7)
s_7=ENTROPY(R410A,T=T_7p,P=P_7)

"State 4"
h_8=h_7p
P_8=P_5
T_8=TEMPERATURE(R410A,P=P_8,h=h_8)
s_8=ENTROPY(R410A,P=P_8,h=h_8)

"Energy balances"
mdot_r=mdot_sup*(T_sOUT-T_sIN)/(h_6-h_7p)          "lb/s, assuming Cp of water is 1"
mdot_geo=mdot_r*(h_5-h_8)/(T_watIN-T_watOUT)       "lb/s"
VolFlow_geo=mdot_geo*0.01602/0.13368*60            "gal/min"
CompWork=mdot_r*(h_6-h_5)*3600                     "Btu/hr"
COP_WW=mdot_sup*(h_sOUT-h_sIN)*3600/(CompWork)
CompWork_kW=CompWork/3414                          "kW"
Q_out=mdot_r*(h_6-h_7p)*3600                       "Btu/hr"


"Overall System Performance"
COP_tot=(Q_out+Q_max)/(CompWork+CompWorkA)
W_tot=CompWork_kW+CompWorkA_kW
VolFlow_tot=VolFlow_geo+VolFlow_geoa
```

```
"Water-to-air heat pump COOLING MODE (Desuperheater)"

"Ground source properties"
P_wat=60                                          "psi"

T_watIN=50                                        "deg F determined from maximum load sizing"
h_watIN=ENTHALPY(Water, P=P_wat,T=T_watIN)        "Btu/lb"

VolFlow_geo=3                                     "gal/min. Reflects the flow of water through the
desuperheater"
mdot_geo=VolFlow_geo/60*0.13368/0.01602           "lb/s"

"House Properties"
T_room=73                                         "deg F, return air temperature"
Q_remove=15076                                    "Btu/hr from Cooling Load Calculations"

"Superheating and subcooling"
degSH=10                                          "deg F per the ASHRAE manual"
degSC=10                                          "deg F per the ASHRAE manual"

"Compressor efficiency"
eta_c=0.8

"State 1"
T_1=T_room+degSH                                  "deg F, allows for superheating of refrigerant.
Note that Tsat is around 5
P_1=70                                            "psi"
h_1=ENTHALPY(R410A,P=P_1, T=T_1)
s_1=ENTROPY(R410A,P=P_1, T=T_1)
"x_1=1"                                           "Two-phase is not practical here! The pressure
would have to be

"State 2"
P_2=P_3
s_2s=s_1
T_2s=TEMPERATURE(R410A,P=P_2,s=s_2s)
h_2s=ENTHALPY(R410A,P=P_2,s=s_2s)

"Accounting for compressor efficiency"
h_2=((h_2s-h_1)/eta_c)+h_1
s_2=ENTROPY(R410A,P=P_2,h=h_2)
T_2=TEMPERATURE(R410A,P=P_2,h=h_2)

"State 3"
T_3=T_watIN                                       "accounts for room temperature set as 73 deg F"
x_3=0
h_3=ENTHALPY(R410A,T=T_3,x=x_3)
P_3=PRESSURE(R410A,T=T_3,x=x_3)

"State 3 has subcooling"
T_3p=T_3-degSC
h_3p=ENTHALPY(R410A,T=T_3p,P=P_3)
s_3=ENTROPY(R410A,T=T_3p,P=P_3)

"State 4"
h_4=h_3p
```

```
P_4=P_1
T_4=TEMPERATURE(R410A,P=P_4,h=h_4)
s_4=ENTROPY(R410A,P=P_4,h=h_4)

"Energy balances"
mdot_r=Q_remove/3600/(h_1-h_4)                          "lb/s"
h_watOUT=mdot_r*(h_2-h_3p)/mdot_geo+h_watIN        "lb/s"
T_watOUT=TEMPERATURE(Water,h=h_watOUT,P=P_wat) "deg F"

CompWork=mdot_r*(h_2-h_1)*3600                          "Btu/hr"
COP=(mdot_r*(h_2-h_3p)*3600)/(CompWork)
CompWork_kW=CompWork/3414                               "kW"
EER=Q_remove/(CompWork_kW*1000) "Btu/hr/W or Btu/Whr"
```

```
"WWHP Operation”

“Ground source properties"
P_wat=60                                            "psi"

"T_watIN"                                           "Parametric value from 45-60 deg F"
T_watOUT=40                                         "deg F"

h_watIN=ENTHALPY(Water, P=P_wat,T=T_watIN)          "Btu/lb"
h_watOUT=ENTHALPY(Water, P=P_wat,T=T_watOUT)
"Need to find m_dot"

"House Supply Properties"
T_sIN=50                                            "deg F"
T_sOUT=120                                          "deg F"
VolFlow=0.006684                                    "ft^3/s, corresponds to 3 gal/min"
mdot_sup=1/0.01602*VolFlow                          "lb/s, make sure to check if valid!"

h_sIN=ENTHALPY(Water, P=P_wat,T=T_sIN)
h_sOUT=ENTHALPY(Water, P=P_wat,T=T_sOUT)

"Superheating and subcooling"
degSH=10                                            "deg F per the ASHRAE manual"
degSC=10                                            "deg F per the ASHRAE manual"

"Compressor efficiency"
eta_c=0.8

"State 1"
T_1=T_watIN+degSH                                   "deg F, allows for superheating of refrigerant.
Note that Tsat is around 5
P_1=70                                              "psi"
h_1=ENTHALPY(R410A,P=P_1, T=T_1)
s_1=ENTROPY(R410A,P=P_1, T=T_1)
"x_1=1"                                             "Two-phase is not practical here! The pressure
would have to be

"State 2"
P_2=P_3
s_2s=s_1
T_2s=TEMPERATURE(R410A,P=P_2,s=s_2s)
h_2s=ENTHALPY(R410A,P=P_2,s=s_2s)

"Accounting for compressor efficiency"
h_2=((h_2s-h_1)/eta_c)+h_1
s_2=ENTROPY(R410A,P=P_2,h=h_2)
T_2=TEMPERATURE(R410A,P=P_2,h=h_2)

"State 3"
T_3=60                                              "accounts for entering water temperature at 50
deg F"
x_3=0
h_3=ENTHALPY(R410A,T=T_3,x=x_3)
P_3=PRESSURE(R410A,T=T_3,x=x_3)

"State 3 has subcooling"
```

```
T_3p=T_3-degSC
h_3p=ENTHALPY(R410A,T=T_3p,P=P_3)
s_3=ENTROPY(R410A,T=T_3p,P=P_3)

"State 4"
h_4=h_3p
P_4=P_1
T_4=TEMPERATURE(R410A,P=P_4,h=h_4)
s_4=ENTROPY(R410A,P=P_4,h=h_4)

"Energy balances"
mdot_r=mdot_sup*(T_sOUT-T_sIN)/(h_2-h_3p)          "lb/s"
mdot_geo=mdot_r*(h_1-h_4)/(T_watIN-T_watOUT)       "lb/s"
VolFlow_geo=mdot_geo*0.01602/0.13368*60            "gal/min"
CompWork=mdot_r*(h_2-h_1)*3600                     "Btu/hr"
COP=mdot_sup*(h_sOUT-h_sIN)*3600/(CompWork)
CompWork_kW=CompWork/3414                          "kW"
Q_out=mdot_r*(h_2-h_3p)*3600                       "Btu/hr"
```