HEATING, VENTILATION, AND AIR CONDITIONING FOR A SUPERINSULATED SOLAR HOUSE

BY

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THESIS

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ABSTRACT

Buildings consume about 70% of all electricity in the United States. There are many ways to reduce the energy consumption of a building and specifically a residential building. The Solar Decathlon competition promoted by the Department of Energy and the National Renewable Energy Laboratory force twenty university teams to find as many ways as possible to reduce energy consumption of a small house. For UIUC's 2009 Solar Decathlon entry, the house was designed to be a certified Passive House, a way of designing a building to use 90% less energy for heating and cooling. The major path to certification is through superinsulation and supersealing of the building. This makes the heating and cooling loads small, requiring a novel HVAC system to provide heating, cooling and ventilation. From the simulated week of occupancy during the competition, where UIUC placed second out of twenty teams, the performance of the HVAC system is analyzed. Using a variable capacity compressor, saved UIUC's Solar Decathlon entry 6.75 kWh or 24.32% electrical energy compared to a fixed capacity compressor. The HVAC system, in conjunction with the excellent thermal envelope, helped lead the team to a close second place finish in comfort zone, the measure of the HVAC system performance. In addition, the energy conservation measures of the HVAC system helped secure a second place finish in energy balance. The excellent performance of the HVAC system does not come at a price premium as it is less expensive than a comparable typical central air heat pump. All these benefits helped to define a house made up of many excellent technologies and practices.

ACKNOWLEDGEMENTS

I would like to thank everyone who has helped me along the way on this adventurous project. Primarily, my thanks goes to my advisor, Dr. Xinlei Wang. Without his belief in me after I came to him about being his graduate student and to work on the 2009 UIUC Solar Decathlon House, I would not be where I am today. He has let me take my own path but provided support and guidance when needed and for that I thank him.

Next, I would like to thank all team members of the Illinois' 2009 Solar Decathlon. Without them and their hard work, my HVAC system would not look as good. A few people who helped me out immensely are Alan Mellovitz who proved to be very handy around the house and Joe Simon who took the role of taskmaster and kept everyone, especially me, on task. It was a great accomplishment for our team to build a complete house and win 2nd place.

Mark Johnson and Mike Wilson from Modine Manufacturing and the various people from Illinois' Facilities and Services all helped me immensely, providing technical support, materials, and guidance.

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Of course, I need to say thank you to my family and friends. Without them, I would not be the man I am today and probably would not have finished my degree. To my girlfriend Kara, I thank you for your infinite patience with me while I was busy working on the HVAC system, the Solar Decathlon house, and my thesis. You do not know how much I love you.

Everyone has shaped my graduate experience and I feel this has been an amazing opportunity and achievement. I know I am forever changed for the better because of these two years.

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Chapter 1: INTRODUCTION

The common practice in the United States currently for residential house design and building is that the bigger is better without considering the total energy assumption of the house. There are many problems associated with building such large homes, including scarcity of materials, embodied energy, extra energy usage especially related to heating and cooling the home. This is especially of interest since according to the Energy Information Administration, heating and cooling account for 26.1% of the total energy consumed by a home in a year (EIA, 2010). There are very few super insulated, airtight homes in the United States and many of the concepts for these types of homes are coming from Germany. The initiator of super insulated homes in the United States is located here in Urbana, Illinois. Very few heating, ventilation and air conditioning, or HVAC, systems are designed specifically for small homes and homebuilders of conventional style homes tend to oversize the HVAC unit. Having a wrong sized HVAC unit for a home uses unnecessary amounts of energy to heat and cool at lower efficiency compared to a properly sized unit. In addition, current HVAC units tend to be a collection of different heating, cooling, dehumidifying, and ventilating units if they have all of these. Even with central air system, in the Midwest, the furnace is usually gas for heating and air conditioning is electric. Then if a home needs a dehumidifier, it is separate somewhere else in the house. In addition, most homes do not require constant mechanical ventilation because they are not airtight and have enough air changes per hour to meet minimum ASHRAE standards (ASHRAE, 2009). Because of this scattered and disjointed nature of these technologies currently, there is a definite need to integrate them together into one package. Due to the extreme insulation levels and air tightness of the house, there needs to be a paradigm shift in residential home HVAC design. The need for heating and cooling are dramatically reduced while mechanical ventilation becomes a requirement. Since the United States has a large climatic difference, these types of homes can be viable anywhere and thus, the need for an integrated HVAC unit, which can also adapt to the specific climate, becomes very real and marketable.

1.1 Solar Decathlon

1.1.1 What is Solar Decathlon?

The inspiration for the pursuit of a novel, compact heating, ventilation, and air conditioning (HVAC) unit came from the University of Illinois at Champaign-Urbana(UIUC)'s 2009 Solar Decathlon entry, the Gable Home. "The Solar Decathlon joins 20 college and university teams in a competition to design, build, and operate the most attractive and energy-efficient solar-powered house" (DOE, 2010). This competition is a joint effort between the Department of Energy (DOE) and the National Renewable Energy Laboratory (NREL). The competition was held four times in 2002, 2005, 2007, and 2009. The competition is a great teaching tool for students who usually never have the ability to take classroom theory and apply it to a real life project. Students are exposed to other areas of focus due to the interdisciplinary nature of the project, joining architects, mechanical engineers, electrical engineers and more.

1.1.2 General Rules

Since Solar Decathlon is a competition, there are various rules governing how the building is constructed and how points are awarded for the competition. The house has to be less than 800 ft^2 solar envelope and can only be powered by energy from the sun. The house has to be transportable from the university to Washington D.C. where the competition is held. Overall, there are ten categories in which the houses are judged: Architecture, Engineering, Communications, Lighting, Market Appeal, Comfort Zone, Appliances, Hot Water, Home Entertainment, and Energy Balance. Overall, these categories are broken up in to a set number of points totaling to 1000 for all ten categories.

1.1.3 Comfort Rules

Since the house and HVAC unit were designed for a competition, there were certain rules and limitations imposed. The rules stated during the week of competition the indoor air temperature must be between 72-76 degrees Fahrenheit to get full points and humidity needed to be between 40% and 55% to get full points. These two readings and points made up 100 of the 1000 total points for the competition.

1.1.4 Competition Specific Issues

There were various issues specific to designing the HVAC system for the house during competition. In a typical house, the air inside the house is more or less at steady state. There are heat losses and gains but unless the envelope conditions are drastically changed, the load in the space is known. During the competition, the envelope was drastically changed almost every day. As part of the competition, public tours had to be given for 4-6 hours. During this time, the doors were open and a thousand people move through the home. This changes the indoor conditions of the home dramatically and then after public tours, teams were given one hour to return the homes to set temperature and humidity ranges to get full points. This very quick enthalpy change in the time limit can become difficult in extreme weather and not indicative of normal house operation.

1.2 Passive House

The design for the 2009 Solar Decathlon entry is rooted in Midwestern tradition bringing the old barn look and feel to a modern and technologically advanced building. One of the major design features to save energy, especially on the HVAC side, is the pursuance of Passive House certification. A Passive House is a unique way of thinking of how to build a house. This standard and design comes from Germany, the PassivHaus Institute. The crux of the design is to build a super insulated, airtight home to reduce the heating load to the lowest possible value. This design concept was brought to the US from Germany and the Passive House Institute was born in the US. Katrin Klingenburg heads the US initiative, building the first Passive House certified home in the US in Urbana, IL in 2003, and currently resides there. The reason for building the home in this climate is to test the design concept in a very harsh climate with wide temperature and humidity difference throughout the year. Thus if the design can work in this location, then it can work anywhere in the US. This also provided validation that the design could work in an environment with a large cooling load, which is important because the climate of Germany is very temperate and does not have as high of cooling loads in the summer as most places in the US.

1.2.1 Insulation

There are several key improvements to a home to achieve the levels of performance by a Passive House. One of these improvements is high levels of insulation, which reduces the HVAC load considerably. To make a home super insulated usually means at least R-70 ceilings, R-50 walls and low-e coated, argon filled, triple paned windows with insulated frames reducing thermal bridges for a cold climate. The goal of this is to limit heat transfer to the environment from the house so the temperature inside does not fluctuate wildly during a 24 hour period. A typically residential home will have wall R-values of R-13 and possibly R-19 for a well insulated home, a stark contrast to the Gable Home that had a wall R-value of R-55 with thermal bridging effects and ceiling R-value of R-72. A typical home is constructed with either 2”x4” wood stud walls or possibly 2”x6” wood stud walls, so to achieve the high R-values required for a certified Passive House, the wall construction needs to be different from standard construction methods. In the Gable Home, the walls were constructed with 1”x10” laminated structural bamboo so the wall cavity is 10 inches instead of 4 or 6 inches. Next, a closed-cell polyurethane spray foam was used for insulation in the wall cavity since it provides an R-value of 7/inch. Fiberglass batt has an R-value of 3-3.5/inch so the spray foam allows for much higher R-value walls for the same thickness. Another benefit of the polyurethane spray foam is that it is an air and vapor barrier, meaning the spray foam will not allow moisture exchange through the envelope nor allow infiltration. This is a very important property and one that will be expanded upon later.

1.2.2 Windows

Windows are one of the, arguably, most important parts of a building’s envelope. An ideal thermal envelope has no windows since all they do is let in solar heat and transfer heat to the environment because they have a significantly reduced R-value compared to the adjacent wall. However, no homeowner would want to live in a home with no windows. Thus, a careful balance between daylighting, solar heat gain, and window to wall ratio needs to be attained. Passive Houses require a majority of the windows to be placed on the south facing wall to maximize solar heat gain during the winter while limiting the number of windows on the other three walls. Another key aspect is shading for the summer months because solar heat gain is detrimental during those months. For the Gable Home, there were moveable external shading devices to block direct solar heat gain in the summer while still allowing for exterior views. The Gable Home has triple pane, two low emissivity coatings, argon gas filled, and corked filled frames that are thermally broken.

1.2.3 Mechanical Ventilation

Thus to achieve such a low power consumption per year, Passive Houses rely on conditioning the air of the home using a heat recovery ventilator, HRV, or an energy recovery ventilator, ERV. By using these ventilators, these homes are able to maintain the minimum ventilation requirements set by ASHRAE; however, they are conserving the heat and moisture, for an ERV, of the exhausting air by preconditioning the fresh intake air. By transferring the heat and moisture from the stale exhaust air to the fresh intake air, an ERV is able to reduce the load seen by the cooling unit in the summer and heating unit in the winter drastically. With the combinations of all these extreme building practices, the HVAC loads are reduced, requiring only minimal power input.

1.2.4 Airtightness

Now this mechanical ventilation only gives these savings on conditioning the space when the natural infiltration is as close to zero as possible. To make sure homes have the best possibly air tightening practices, the Passive House standard imposes a maximum of 0.6 air changes per hour, ACH, where the house is depressurized to 50 Pascals and the ACH needs to be less than 0.6. This number is determined in a built Passive House by using a blower door test. If you convert the pressurized ACH to natural ACH then it is ~0.033 ACH from infiltration. This is well below the minimum required by ASHRAE, who requires a minimum of 0.35 ACH. This means the Passive House needs constant mechanical ventilation to maintain the necessary minimum ventilation requirement.

1.2.5 Certification

Passive House differs from other green building certifications because there are only four goals a home has to meet to become Passive House certified. These four goals are maximums of 15 kWh/m^2/year for heating and cooling, 120 kWh/m^2/year for primary energy consumption and less than 0.6 ACH@50Pa depressurization. These four requirements are the same for any Passive House in the world, which has caused some friction for the Passive House Institute US. The heating and cooling requirements in Germany where the standard was developed are vastly different from a majority of the US climate zones. While there are only four requirements, the level of detail to get to those four requirements is very high. A great deal of care and attention needs to be placed on the construction of the house since many contractors are not accustomed to the unique constructions methods required for a Passive House.

1.3 Reasons for HVAC Need

1.3.1 Thermal Loads

Due to the thermal performance of a Passive House certified home, the thermal loads in home are significantly minimized compared to a typical home. PassivHaus Institute claims a savings of 90% in heating and cooling energy consumed compared to a typical house (Passive House, 2008). Since the loads are significantly reduced, the HVAC system needs to be addressed differently. One of the major advantages that a new HVAC system can take advantage of because of the reduced thermal loads is the use of the fresh supply air for heating and cooling. Due to the air tightness level of the building, the house requires a low volume of constant fresh air. This fresh air can carry enough conditioned air to satisfy the load requirements in the building.

1.3.2 Current HVAC Options

The summer design day thermal load in the home, with exact numbers discussed later, is the same as the smallest window air conditioner currently on the market. However, a window air conditioner would be inappropriate and ineffective at conditioning the multi-zone space of UIUC’s 2009 Solar Decathlon house. Thus, there is a need to design a window air conditioner sized heat pump system, which better serves the two different zones of the home. Traditional home furnaces and central air conditioners are vastly oversized for the thermal loads of the Gable Home.

1.3.3 Integrated Approach

Since the fresh supply air can be heated and cooled enough to condition the home, this allows for novel integration of heating, ventilation, and cooling systems. An integrated approach allows for reductions in initial capital expenses compared to separately purchased HVAC systems. Another advantage is the physical space requirement is less than conventional HVAC systems, allowing the unit to fit in tighter locations and free up valuable square footage inside the building. For the Gable Home, this was especially important since the HVAC system was limited to a small space above the bathroom.

1.3.4 Objective

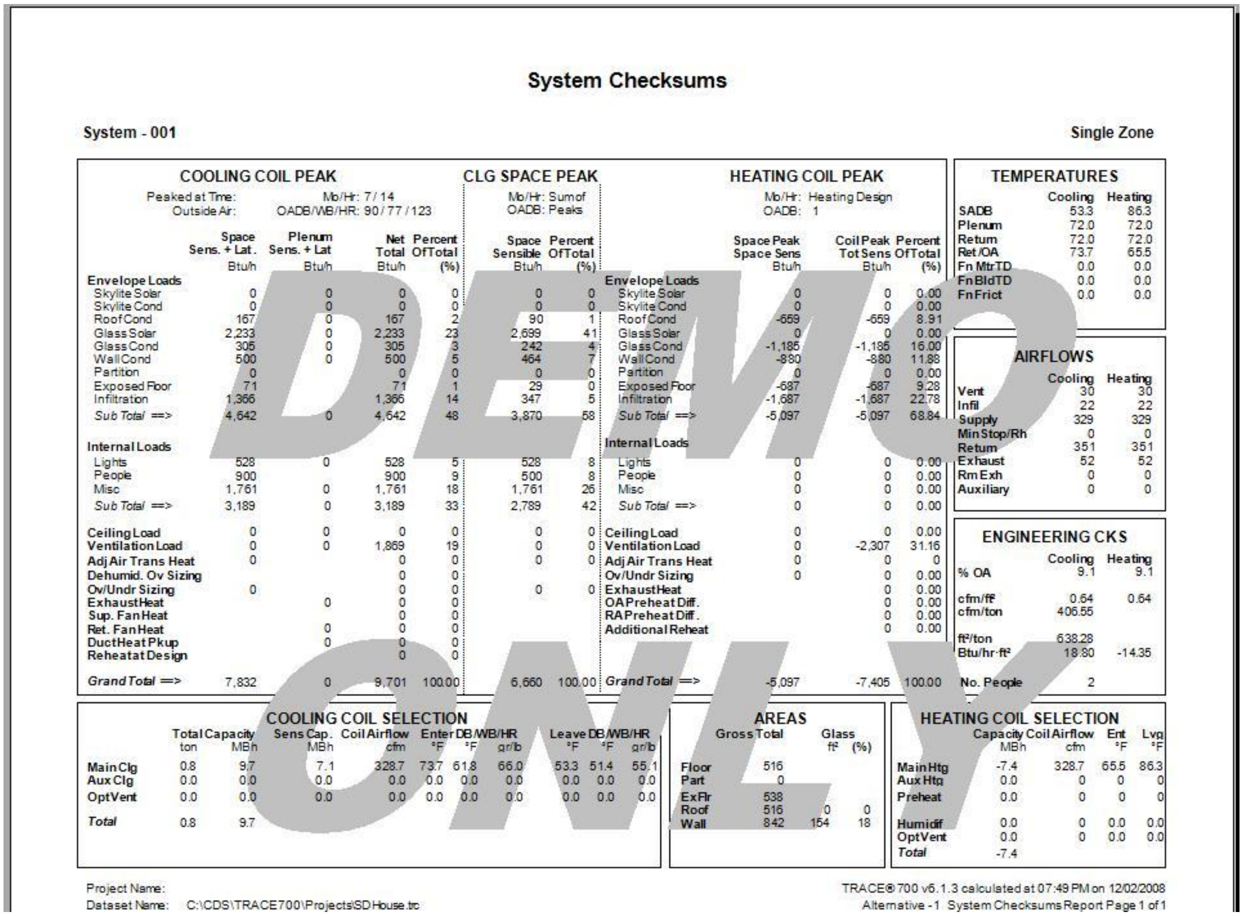
The objective of this project is to design and build a compact, cost-effective, and high efficiency HVAC system for a superinsulated solar house.

Chapter 2: METHODOLOGY

2.1 HVAC Design

2.1.1 Energy Modeling

Trance Trace 700 is a software package developed by Trane to help size HVAC systems in mostly commercial environments but it can be used for residential house modeling as well. This software was used to determine the heating and cooling loads on the house. These loads determine the necessary capacity of the compressor and heat exchangers to maintain the desired set point temperature for all weather conditions. Trane Trace is a straightforward software package to use and only requires general information about the house such as wall, floor, ceiling insulation levels, type and efficiency of HVAC unit, infiltration, etc. After modeling the house in Trace 700 and running a simulation, the heating and cooling loads can be seen in Figure 2-1.



System Checksums

System - 001 | Single Zone

COOLING COIL PEAK — Peaked at Time: Mo/Hr: 7 / 14; Outside Air: OADB/WB/HR: 90 / 77 / 123

CLG SPACE PEAK — Mo/Hr: Sum of; OADB: Peaks

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	167	0	167	2	90	1
Glass Solar	2,233	0	2,233	23	2,699	41
Glass Cond	305	0	305	3	242	4
Wall Cond	500	0	500	5	464	7
Partition	0		0	0	0	0
Exposed Floor	71		71	1	29	0
Infiltration	1,366		1,366	14	347	5
Sub Total ==>	4,642	0	4,642	48	3,870	58
Internal Loads						
Lights	528	0	528	5	528	8
People	900		900	9	500	8
Misc	1,761	0	1,761	18	1,761	26
Sub Total ==>	3,189	0	3,189	33	2,789	42
Ceiling Load	0	0	0	0	0	0
Ventilation Load	0	0	1,869	19	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		0	0	0		
Sup. Fan Heat			0	0		
Ret. Fan Heat		0	0	0		
Duct Heat Pkup		0	0	0		
Reheat at Design			0	0		
Grand Total ==>	7,832	0	9,701	100.00	6,660	100.00

HEATING COIL PEAK — Mo/Hr: Heating Design; OADB: 1

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	-659	-659	8.91
Glass Solar	0	0	0.00
Glass Cond	-1,185	-1,185	16.00
Wall Cond	-880	-880	11.88
Partition	0	0	0.00
Exposed Floor	-687	-687	9.28
Infiltration	-1,687	-1,687	22.78
Sub Total ==>	-5,097	-5,097	68.84
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	0	0	0.00
Ventilation Load	0	-2,307	31.16
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	0	0	0.00
Exhaust Heat		0	0.00
OA Preheat Diff.		0	0.00
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Grand Total ==>	-5,097	-7,405	100.00

TEMPERATURES

	Cooling	Heating
SADB	53.3	86.3
Plenum	72.0	72.0
Return	72.0	72.0
Ret/OA	73.7	65.5
Fn MtrTD	0.0	0.0
Fn BldTD	0.0	0.0
Fn Frict	0.0	0.0

AIRFLOWS

	Cooling	Heating
Vent	30	30
Infil	22	22
Supply	329	329
MinStop/Rh	0	0
Return	351	351
Exhaust	52	52
Rm Exh	0	0
Auxiliary	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	9.1	9.1
cfm/ft²	0.64	0.64
cfm/ton	406.55	
ft²/ton	638.28	
Btu/hr·ft²	18.80	-14.35
No. People	2	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	0.8	9.7	7.1	328.7	73.7	61.8	66.0	53.3	51.4	55.1
Aux Clg	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Total	0.8	9.7								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	516		
Part	0		
ExFlr	538		
Roof	516	0	0
Wall	842	154	18

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-7.4	328.7	65.5	86.3
Aux Htg	0.0	0	0	0
Preheat	0.0	0	0	0
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-7.4			

Project Name:
Dataset Name: C:\CDS\TRACE700\Projects\SDHouse.trc

TRACE® 700 v6.1.3 calculated at 07:49 PM on 12/02/2008
Alternative - 1 System Checksums Report Page 1 of 1

DEMO ONLY

Figure 2-1. Load analysis in Trane Trace.

The second energy modeling package used was DesignBuilder. “DesignBuilder is a state-of-the-art software tool for checking building energy, CO2, lighting and comfort performance. Developed to simplify the process of building simulation, DesignBuilder allows you to rapidly compare the function and performance of building designs and deliver results on time and on budget” (DesignBuilder, 2009). DesignBuilder has much more detailed control of building information than Trane Trace 700. DesignBuilder has a great visual interface for constructing a building from scratch. DesignBuilder uses EnergyPlus for the simulation engine. However, DesignBuilder does not take advantage of every feature available in EnergyPlus. One major short fall is how DesignBuilder computes HVAC energy, which is not as full featured as EnergyPlus. In DesignBuilder, the Gable Home was modeled with the weeklong competition schedule. This way a comparison can be drawn between the simulation of the competition and the actual results of the competition. Figure 2-2 through Figure 2-7 show various information about the DesignBuilder model, winter design day, and summer design day.

With energy modeling, the outputs are only as good as the inputs as Tianzhen et al proved (Tianzhen, 2009). To get good, reliable outputs you need to be very careful that the inputs you enter into the model are correct and are modeled correctly. In addition, there are variances between energy modeling software so even if a building is modeled exactly the same way, the calculation methods are different and result in different outputs. The Trane Trace 700 energy model was created before many details about the Gable Home were finalized which is one reason why the values different from those given by DesignBuilder. EnergyPlus, which is the simulation engine for DesignBuilder, is able to model transient conditions in the building, which is especially important a Passive House. EnergyPlus models the building with sub hourly timesteps , which means there is a more accurate picture of what is happening in the building. Trane Trace 700 on the other hand can only model the building with hourly timesteps. Another important feature of EnergyPlus is the ability to model moisture transfer due to the iterative simulation method. For a house designed to retain conditioning, a byproduct is that it retains moisture. Moisture transfer can be a huge load that can be difficult to model and will cause discrepancy between simulation and experimental data. Overall, DesignBuilder and EnergyPlus were able to simulate the Gable Home well to provide accurate design day loads.

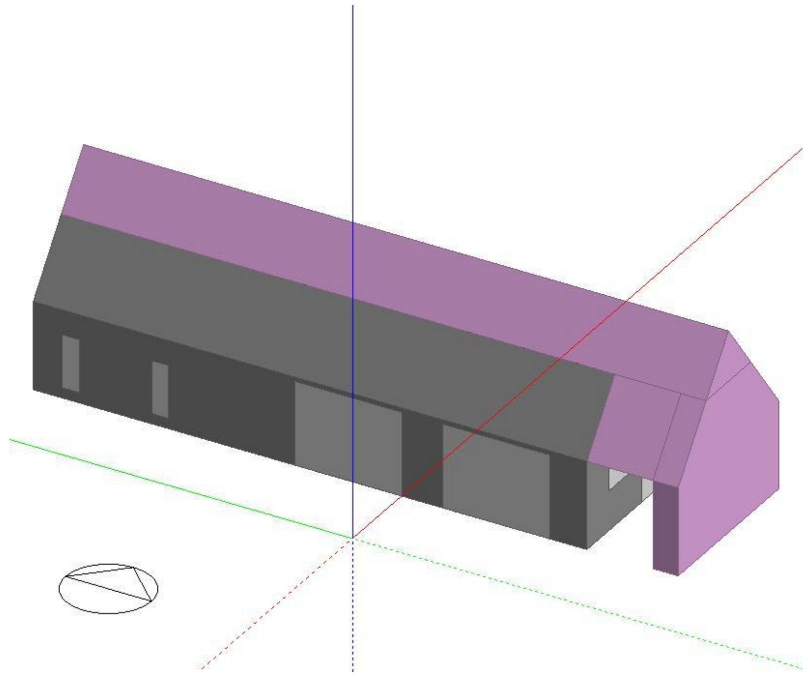


Figure 2-2. Gable Home DesignBuilder model.

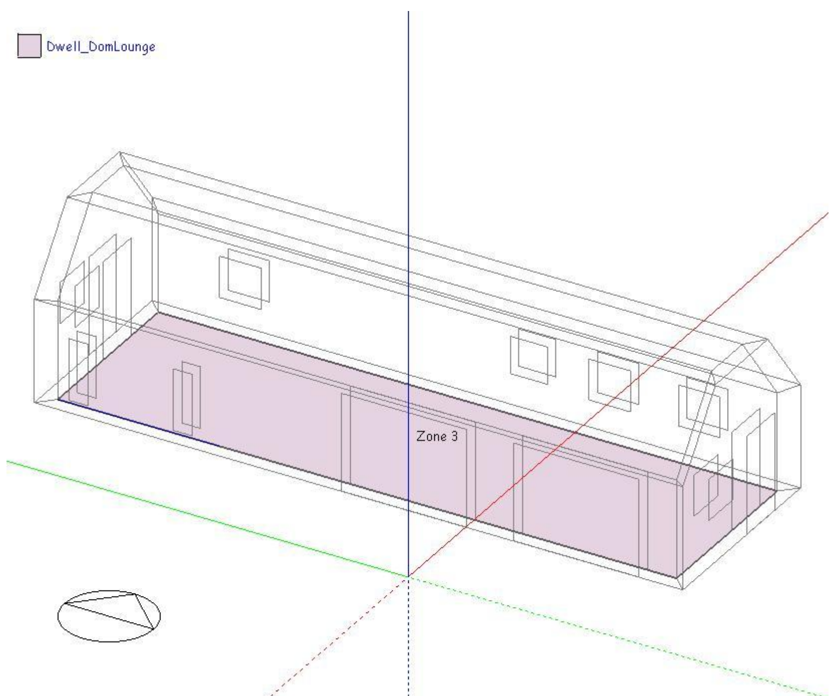
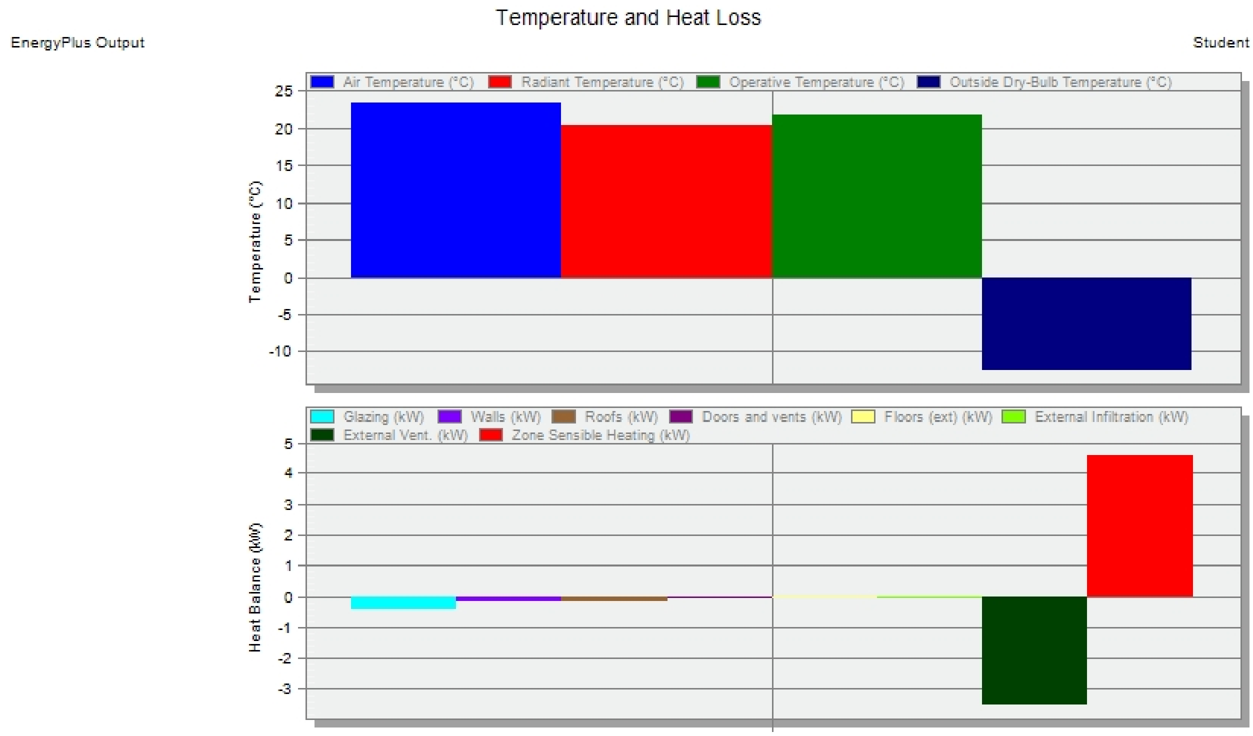
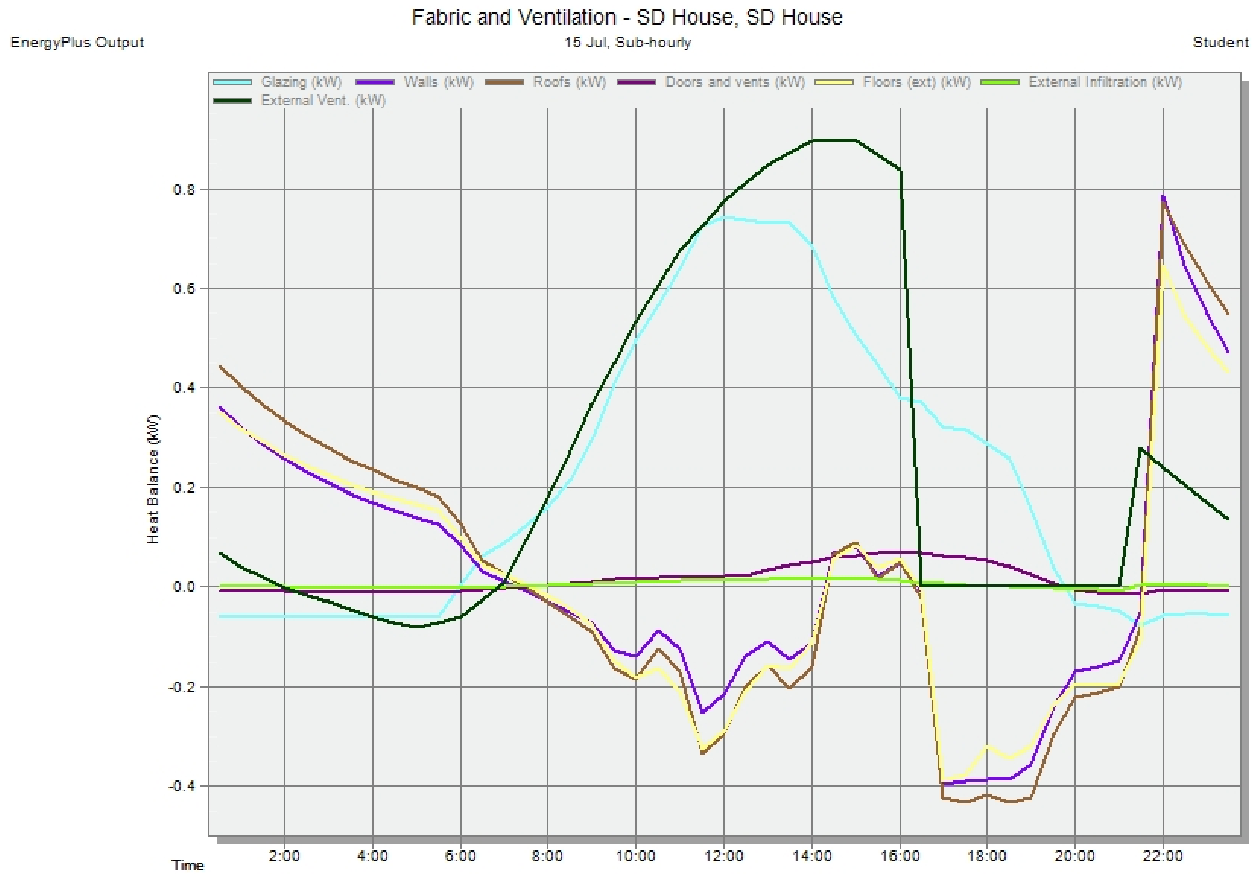


Figure 2-3. Gable Home zone model.



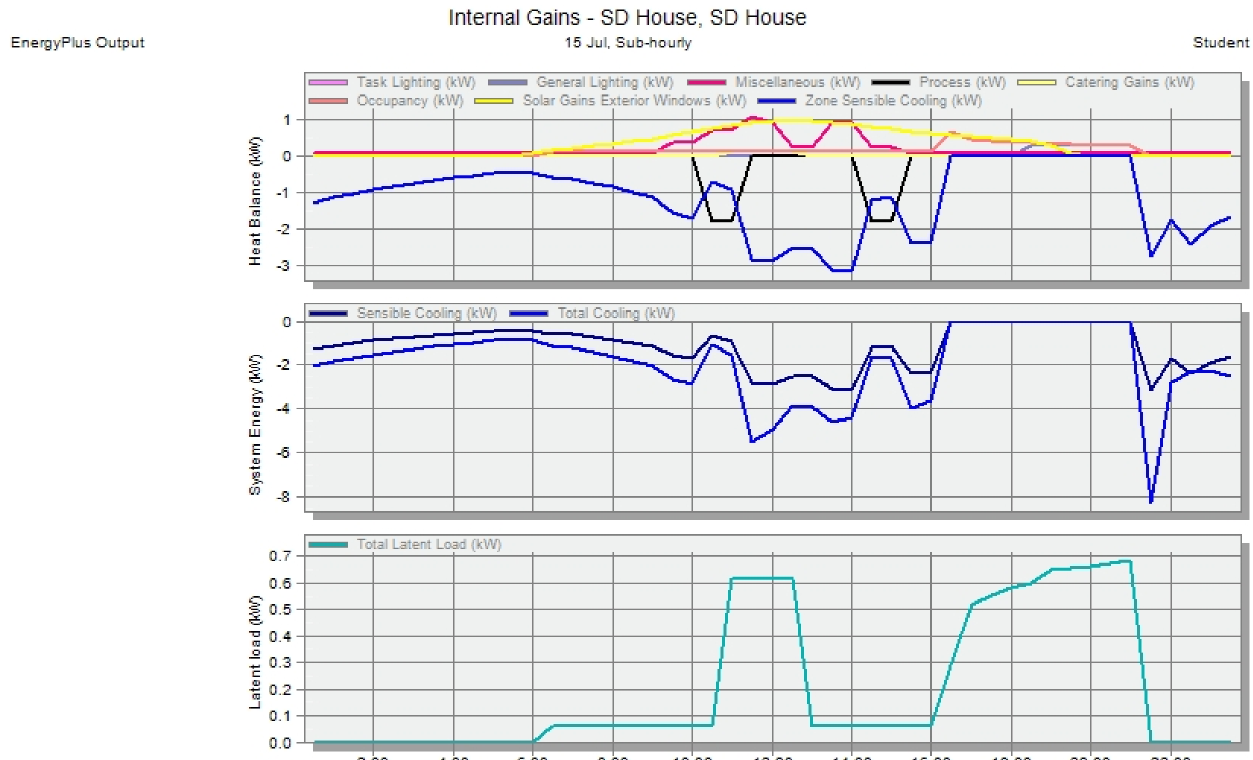
Air Temperature (°C)	23.33
Radiant Temperature (°C)	20.36
Operative Temperature (°C)	21.84
Outside Dry-Bulb Temperature (°C)	-12.50
Glazing (kW)	-0.42
Walls (kW)	-0.19
Roofs (kW)	-0.18
Doors and vents (kW)	-0.09
Floors (ext) (kW)	-0.09
External Infiltration (kW)	-0.06
External Vent. (kW)	-3.53
Zone Sensible Heating (kW)	4.57

Figure 2-4. DesignBuilder heating design day.



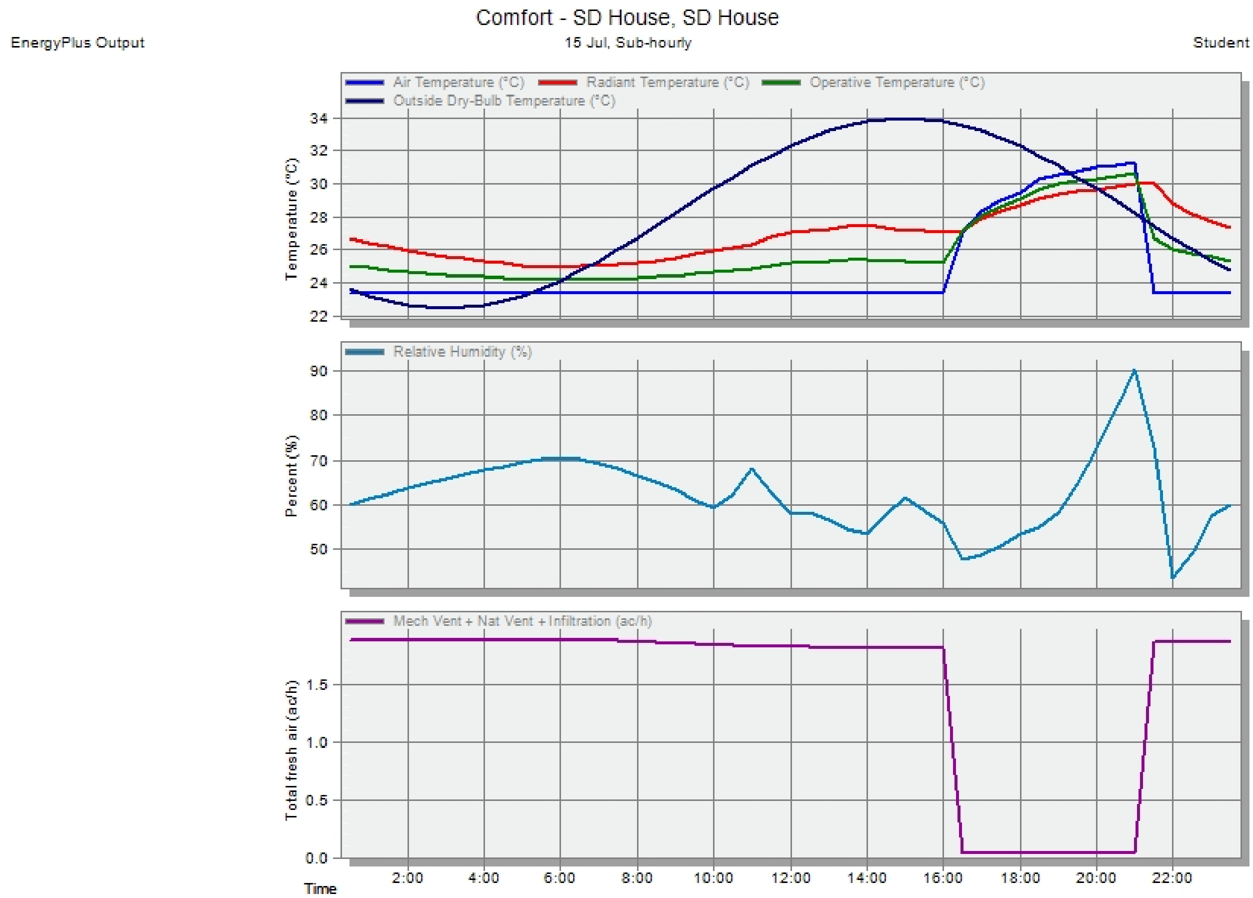
Glazing (kW)	-0.06	-0.06	0.00	0.16	0.50	0.74	0.69	0.38	0.29	-0.03	-0.06
Walls (kW)	0.26	0.17	0.08	-0.03	-0.14	-0.22	-0.11	0.05	-0.39	-0.17	0.79
Roofs (kW)	0.33	0.24	0.13	-0.03	-0.18	-0.29	-0.16	0.05	-0.42	-0.22	0.78
Doors and vents (kW)	-0.01	-0.01	-0.01	0.01	0.02	0.02	0.05	0.07	0.05	-0.01	-0.00
Floors (ext) (kW)	0.27	0.19	0.10	-0.02	-0.18	-0.29	-0.11	0.06	-0.32	-0.19	0.65
External Infiltration (kW)	-0.00	-0.00	-0.00	0.00	0.01	0.01	0.02	0.02	0.00	-0.00	0.00
External Vent. (kW)	-0.00	-0.06	-0.06	0.18	0.53	0.77	0.90	0.84	0.00	0.00	0.24

Figure 2-5. DesignBuilder summer design day envelope loads.



Time	2:00	4:00	6:00	8:00	10:00	12:00	14:00	16:00	18:00	20:00	22:00
Task Lighting (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
General Lighting (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.31	0.00
Miscellaneous (kW)	0.09	0.09	0.09	0.09	0.39	0.95	0.95	0.09	0.09	0.09	0.09
Process (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Catering Gains (kW)	0.00	0.00	0.00	0.00	0.00	0.14	0.00	0.00	0.00	0.00	0.00
Occupancy (kW)	0.00	0.00	0.00	0.14	0.14	0.14	0.14	0.14	0.41	0.33	0.00
Solar Gains Exterior Windows (kW)	0.00	0.00	0.10	0.35	0.68	0.97	0.87	0.64	0.48	0.00	0.00
Zone Sensible Cooling (kW)	-0.89	-0.56	-0.43	-0.83	-1.71	-2.86	-3.16	-2.30	0.00	0.00	-1.71
Sensible Cooling (kW)	-0.89	-0.56	-0.43	-0.83	-1.71	-2.86	-3.16	-2.30	0.00	0.00	-1.71
Total Cooling (kW)	-1.53	-1.06	-0.87	-1.61	-2.83	-4.95	-4.41	-3.59	0.00	0.00	-2.77
Total Latent Load (kW)	0.00	0.00	0.00	0.06	0.06	0.62	0.06	0.06	0.58	0.66	0.00

Figure 2-6. DesignBuilder summer design day internal gains.



	2:00	4:00	6:00	8:00	10:00	12:00	14:00	16:00	18:00	20:00	22:00
Air Temperature (°C)	23.33	23.33	23.33	23.33	23.33	23.33	23.33	23.33	29.45	30.98	23.33
Radiant Temperature (°C)	25.97	25.31	24.94	25.20	25.98	27.02	27.52	27.07	28.68	29.62	28.81
Operative Temperature (°C)	24.65	24.32	24.14	24.26	24.65	25.18	25.43	25.20	29.06	30.30	26.07
Outside Dry-Bulb Temperature (°C)	22.60	22.60	24.10	26.70	29.70	32.30	33.80	33.80	32.30	29.70	26.70
Relative Humidity (%)	63.65	67.68	70.65	66.52	59.12	57.99	53.42	55.94	53.41	72.71	43.72
Mech Vent + Nat Vent + Infiltration (ac/h)	1.88	1.88	1.88	1.86	1.84	1.82	1.81	1.82	0.03	0.03	1.86

Figure 2-7. DesignBuilder summer design day comfort.

2.1.2 Heat Exchanger Data

In Table 2-1 through Table 2-3, Modine Manufacturing Company models the heat exchangers used in the HVAC system. This is important information since it determines if the heat exchangers are the correct size for the application. It also shows whether or not the refrigerant will have the proper physical properties moving through the evaporator and condenser. These models used the correct compressor map and airflows of those in the HVAC system.

Table 2-1. Heat exchanger - system model.

Condenser CFM held constant at 400CFM, Compressor displacement constant, Evap CFM varied					
System Model results - RTPF cond					
Condensation Temp.	°F	99.313	108.78	111.76	113.48
Evaporation Temp.	°F	15.955	35.416	40.003	42.63
Condensation Press.	psia	137.5	158.32	165.27	169.47
Evaporation Press.	psia	30.367	45.51	49.796	52.331
Condenser SubCooling	°F	10	10	10	10
Total SubCooling	°F	10	10	10	10
Evaporator Superheat	°F	10	10	10	10
Total SuperHeat	°F	10	10	10	10
COP		2.7684	3.547	3.6637	3.7306
COP Fan/Pump Included		2.7684	3.547	3.6637	3.7306
EER		9.4546	12.114	12.512	12.741
EER Fan/Pump Included		9.4546	12.114	12.512	12.741
Total Carnot Eff.	%	48.519	52.563	52.615	52.621
Total Carnot Eff. Fan/Pump Included	%	48.519	52.563	52.615	52.621
COP(HPA)		3.7184	4.497	4.6137	4.6806
COP(HPA) Fan/Pump Included		3.7184	4.497	4.6137	4.6806
EER(HPA)		12.699	15.358	15.757	15.985
EER(HPA) Fan/Pump Included		12.699	15.358	15.757	15.985
TCE(HPA)	%	55.451	58.04	57.938	57.86
TCE(HPA) Fan/Pump Included	%	55.451	58.04	57.938	57.86
Cooling Capacity	Btu/h	6379	9545.7	10528	11101
Heating Capacity	Btu/h	8568	12102	13258	13927
Mass Flowrate	lbm/h	96.438	145.33	161.26	170.65
Comp. Disch. Temp.	°F	154.59	149.7	151.13	152.02
Power Input	kW	0.6747	0.788	0.84144	0.87128
Isentropic Eff.	%	61.314	65.529	65.6	65.566
Compressor Eff.	%	58.248	62.253	62.32	62.288
Volumetric Eff.	%	30	30.689	31.22	31.488
Refrigerant		R134a	R134a	R134a	R134a
Evaporator Sec. Fluid Inlet Flow Rate (*)	ft^3/min	70	200	300	400

Table 2-2. Heat exchanger - condenser model.

Condenser Model results					
Name		2 3row cond	2 3row cond	2 3row cond	2 3row cond
Exchanger´s Type		Round tube plate fin	Round tube plate fin	Round tube plate fin	Round tube plate fin
Calculation Model		Detailed	Detailed	Detailed	Detailed
Capacity	Btu/h	8568	12102	13258	13927
Efficiency	%	87.839	87.68	87.939	88.082
UA	Btu/(h °F)	900.85	8.95E+02	904.39	909.48
NTU		2.1069	2.094	2.1152	2.1271
Sensible LMTD	°F	9.511	13.518	14.66	15.314
Refrigerant		R134a	R134a	R134a	R134a
Refrigerant Area	ft²	4.5946	4.5946	4.5946	4.5946
De-superheating area	%	9.5958	9.1758	9.2709	9.3433
Two Phase Area	%	64.486	65.041	65.017	65
Subcooling Area	%	25.919	25.783	25.712	25.657
Mass Flowrate	lbm/h	96.438	145.33	161.26	170.65
Inlet Temperature	°F	154.59	149.7	151.13	152.02
Outlet Temperature	°F	89.313	98.782	101.76	103.48
Inlet Superheat	°F	55.111	40.632	39.022	38.189
Outlet Subcooling	°F	10.001	9.9998	10	9.9999
Pressure Drop	psi	0.34125	0.6754	0.80121	0.87453
Secondary Fluid		Air	Air	Air	Air
Area	ft²	99.354	99.354	99.354	99.354
Flowrate	ft³/min	400	400	400	400
Face Velocity	ft/min	575.99	575.99	575.99	575.99
Inlet Temperature	°F	76.5	76.5	76.5	76.5
Outlet Temperature	°F	96.515	104.77	107.47	109.03
Pressure Drop	in WC	1.1368	1.148	1.1517	1.1539
Evaporator Sec. Fluid Inlet Flow Rate (*)	ft³/min	70	200	300	400

Table 2-3. Heat exchanger - evaporator model.

Evaporator Model results					
Name		RTPF evap 15FPI	RTPF evap 15FPI	RTPF evap 15FPI	RTPF evap 15FPI
Exchanger´s Type		Round tube plate fin	Round tube plate fin	Round tube plate fin	Round tube plate fin
Calculation Model		Detailed	Detailed	Detailed	Detailed
Capacity	Btu/h	6379	9545.7	10528	11101
Sensible Capacity	Btu/h	3941	6292.1	7507.1	8473
SHR	%	61.78	65.916	71.304	76.329
Sensible Efficiency	%	86.993	71.64	64.143	58.508
Sensible UA	Btu/(h °F)	152.62	269.4	328.9	376.12
Sensible NTU		2.0397	1.2602	1.0256	0.87968
Sensible LMTD	°F	25.822	23.356	22.825	22.527
Refrigerant		R134a	R134a	R134a	R134a
Refrigerant Area	ft²	2.2973	2.2973	2.2973	2.2973
Two Phase Area	%	98.015	96.942	96.443	96.181
Superheating Area	%	1.9853	3.0578	3.5572	3.8192
Mass Flowrate	lbm/h	96.438	145.33	161.26	170.65
Inlet Temperature	°F	23.441	44.472	49.456	52.258
Outlet Temperature	°F	25.955	45.415	50.005	52.63
Inlet Quality		0.25105	0.22109	0.21665	0.21432
Outlet Superheat	°F	10	9.999	10.002	10
Pressure Drop	psi	5.2976	8.7422	9.7789	10.399
Secondary Fluid		Air	Air	Air	Air
Area	ft²	49.677	49.677	49.677	49.677
Flowrate	ft³/min	70	200	300	400
Face Velocity	ft/min	100.8	288	431.99	575.99
Inlet Temperature	°F	76.5	76.5	76.5	76.5
Outlet Temperature	°F	23.217	46.927	53.031	56.659
Pressure Drop	in WC	0.04877	0.22834	0.41164	0.62436
Evaporator Sec. Fluid Inlet Flow Rate (*)	ft³/min	70	200	300	400

2.1.3 HVAC Design

The design for the HVAC system went through various iterations over the course of the design phase. This is typical for any design project where each iteration has new problems and they are fixed. The end goal of the design is to make a compact unit that heats, cools, dehumidifies, and ventilates while heating and cooling through ventilation air only.

The first design involved a single flow heat pump that diverted airflow across the two heat exchangers depending if heating or cooling was needed in the conditioned space. The major downside to the design was the large physical dimensions of the design as well as the cost and complexity of the number of ducts and electronic dampers. The next design used a reversible flow heat pump to reduce the complexity of the ducting. Now the heat exchangers change function while staying in the same airflow, meaning either supply or exhaust air streams. This design made the system significantly more compact; however, the layout was not ideal for the location in the Gable Home. The location of the ERV filters, compressor, and heat exchangers would have made servicing them very difficult in the tight space allotted.

The final design uses a reversible refrigerant flow heat pump while allowing access to the unit for easier servicing, an important feature for the prototype during competition. In Figure 2-8, the overall design of the HVAC system can be seen. The ERV is located in the center with the two heat exchangers on either side. There are two return ducts, one in the hallway and one in the bathroom. This allows moisture from showers as well as general bathroom air exchange to pass through the ERV, preserving interior conditioning.

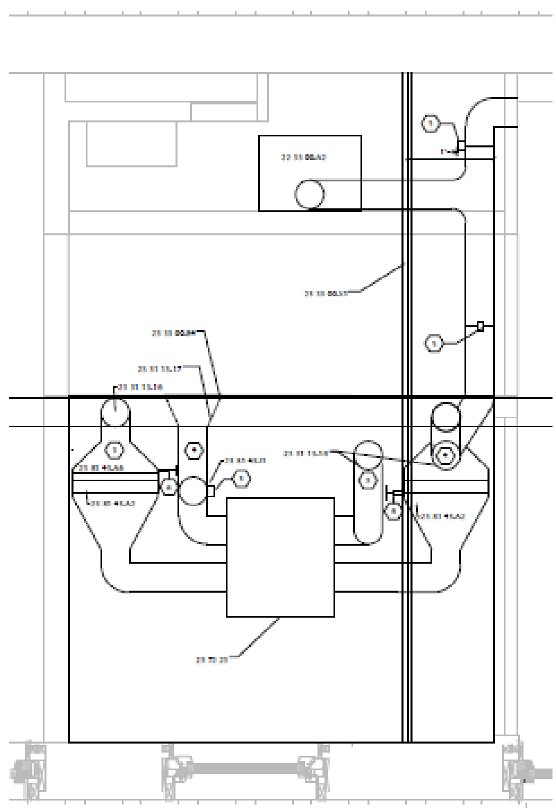


Figure 2-8. Plan view of HVAC system.

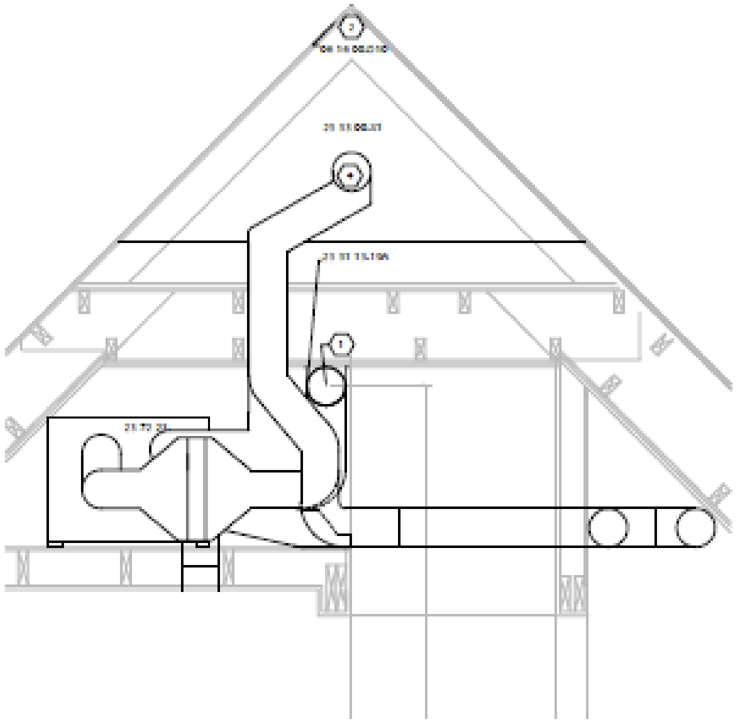


Figure 2-9. Section view of HVAC system.

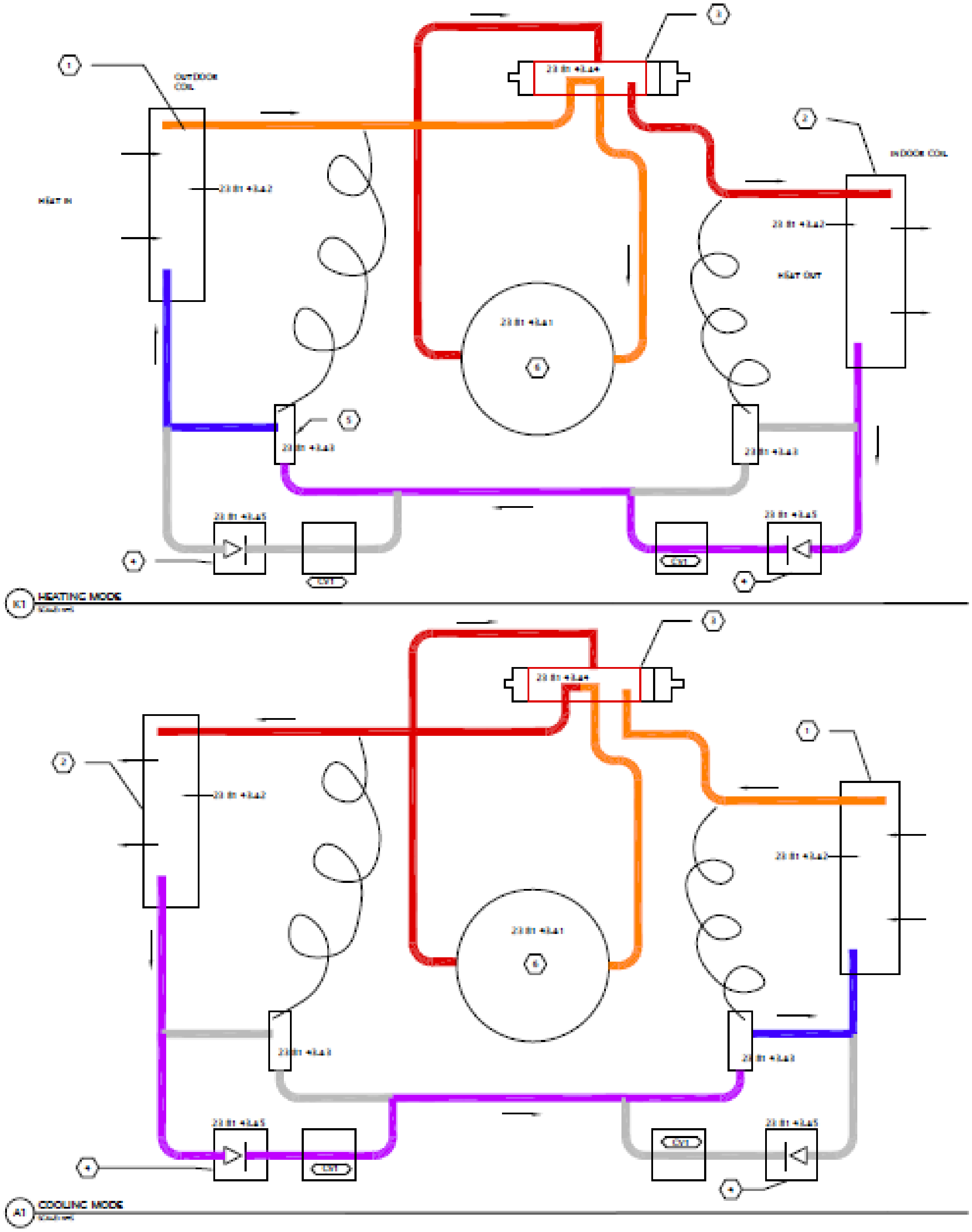


Figure 2-10. Schematic diagram of refrigerant flow for heating and cooling.

2.2 HVAC Construction

2.2.1 Materials and Equipment

The main equipment for the HVAC system are displayed in the following figures with a table outlining the full equipment list with costs.

Figure 2-11. DC variable capacity compressor with attached reversing valve.

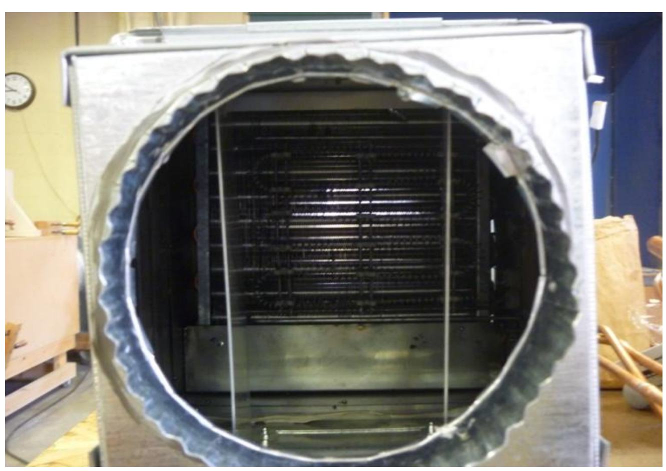


Figure 2-12. View of heat exchanger in ductwork.

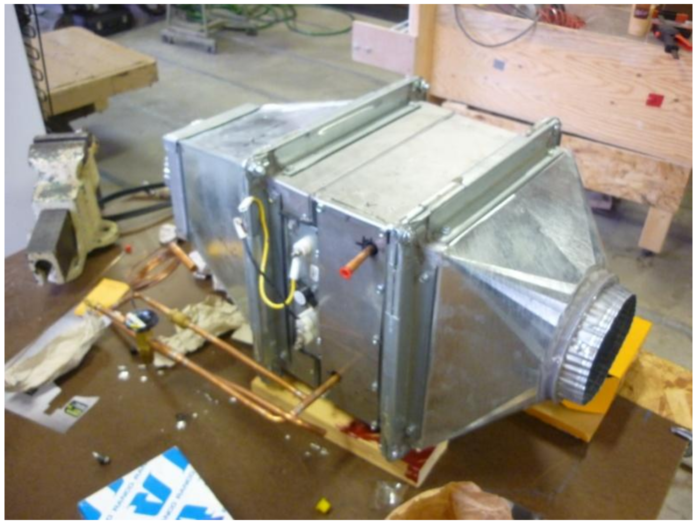


Figure 2-13. Ductwork surrounding supply heat exchanger.

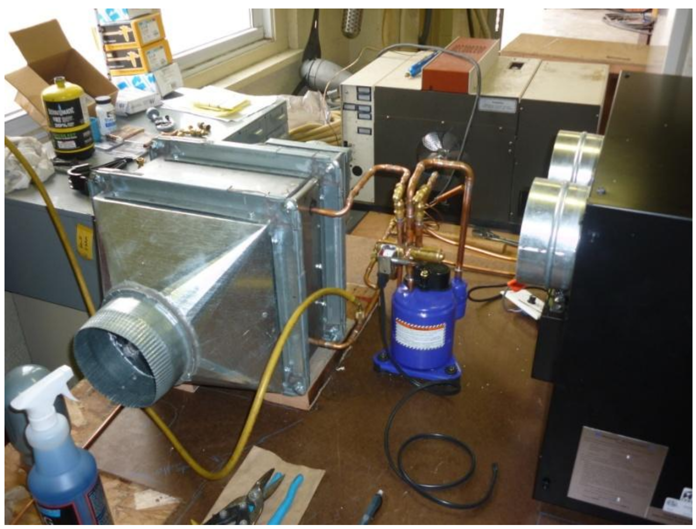


Figure 2-14. View of exhaust ductwork and heat exchanger, compressor, reversing valve, and ERV.

Figure 2-15. View of ERV, supply ductwork, and filter dryer.

Figure 2-16. View of installed HVAC system with duct insulation and control boxes.

Figure 2-17. View of installed HVAC system towards return and supply ducts.

Table 2-4. Bill of materials for HVAC system.

Name	Quantity	Cost	Total	Price Quote
Compressor	1	$383.00	$383.00	Masterflux
Compressor controller	1	$313.00	$313.00	Masterflux
Compressor controller wiring harness	1	$31.00	$31.00	Masterflux
Human interface device	1	$37.00	$37.00	Masterflux
Rectifier	1	$100.00	$100.00	Masterflux
Heat exchanger	3	$130.00	$390.00	Modine Manufacturing Company
Ducting around heat exchangers	2	$550.00	$1,100.00	University of Illinois Facilities and Services Sheet Metal Shop
Energy Recovery Ventilator	1	$1,899.00	$1,899.00	UltimateAir
Reversing valve	1	$92.50	$92.50	Duncan Supply, Inc.
Reversing valve coil	1	$26.42	$26.42	Duncan Supply, Inc.
90 deg duct elbow	10	$2.38	$23.80	Duncan Supply, Inc.
Tee duct	3	$12.19	$36.57	Duncan Supply, Inc.
Y duct	1	$20.37	$20.37	Duncan Supply, Inc.
6" electronic damper	3	$97.24	$291.72	Duncan Supply, Inc.
Bathroom diffuser	1	$12.16	$12.16	Duncan Supply, Inc.
Wall diffuser	5	$12.16	$60.80	Duncan Supply, Inc.
6" duct	45	$1.20	$54.00	Duncan Supply, Inc.
Foil tape	1	$17.36	$17.36	Duncan Supply, Inc.
Thermostatic expansion valve	2	$71.78	$143.56	Grainger Supply
Check valve	2	$14.17	$28.34	Duncan Supply, Inc.
Copper tubing, 50ft	1	$35.99	$35.99	Duncan Supply, Inc.
Refrigerant piping 90 deg elbow	14	$1.39	$19.46	Duncan Supply, Inc.
Refrigerant piping tee	4	$2.59	$10.36	Duncan Supply, Inc.
Filter drier	1	$21.96	$21.96	Duncan Supply, Inc.
Schrader valve	6	$15.99	$95.94	Duncan Supply, Inc.
Pressure transducer	2	$225.00	$450.00	Omega Engineering
Duct insulation	6	$13.64	$81.84	Duncan Supply, Inc.
Inline Duct Fan	2	$130.95	$261.90	Fantech, Inc.
Labor	25	$70.00	$1,750.00	
		Total	$7,788.05	

2.2.2 Construction Methods

First, the HVAC system was laid out in the lab on a template of the exact space available in the loft space in the Gable Home. This was done since the space in the Gable Home was very confining so piecing together parts would prove difficult. Another reason for constructing the refrigerant loops in the lab is due to the use of an acetylene torch for soldering the copper connections. In an enclosed space, this could prove to be hazardous. High silver content solder was used to join the copper refrigerant tubing. Once the refrigerant lines were soldered in the correct locations on the template, the HVAC system was disassembly and re-assembled in the house. Once in the house, the ductwork was assembled according to design specifications. The ducts were taped with UL listed foil tape. This ensures a permanent, good seal for all the duct connections. After the ducts were sealed, they were insulated to reduce heat loss or gain into the ductwork. The refrigerant lines were insulated as well to reduce heat loss on the cold lines. This helps to raise the overall efficiency of the HVAC unit.

2.2.3 Refrigerant

The refrigerant used in the HVAC system is R-134a. This refrigerant has zero ozone depletion potential but a global warming potential of 1300. The choice of the refrigerant was driven by the compressor choice since the compressor required R-134a. Another advantage of using this refrigerant is that it is widely used in automobiles, thus when a new refrigerant with a lower GWP is developed as a drop in replacement, it can be used in the HVAC system.

2.2.4 Charging

The HVAC system was charged by checking the head pressure, the pressure after the condenser, then converting that to its corresponding temperature and then measuring the actual temperature of the refrigerant line using a pocket temperature probe. When you take the difference of these two values this is called subcooling. The subcooling should be between 5 to 10 °F, which indicates a properly charged system. Charging by subcooling is required because the HVAC system uses a thermostatic expansion valve. Once the subcooling is at the proper temperature difference, the superheat needs to be checked.

Superheat is determined by measuring the suction pressure, the pressure after the evaporator, and converting it to its corresponding temperature then measure the actual temperature of the refrigerant line, the difference is the superheat. Superheat is critical to the reliability of the HVAC system. If superheat is near zero then there is the potential to ‘slug’ the compressor with liquid refrigerant and destroy it. Thus it is important to have at least 5 °F of superheat. However, too high of superheat causes the compressor to work harder, decreasing the coefficient of performance.

2.2.5 Operating Conditions

During normal operating conditions, the suction pressure should be around 40 psig and the head pressure should not rise above 200 psig. If the head pressure is above this then the compressor will begin to overheat and thermally shut off. A suction pressure of 40 psig corresponds to a temperature of 45 °F. This is a good evaporator coil temperature because it will allow for proper dehumidification by condensing water vapor out of the air across the coil. At a higher coil temperature, the capacity of the system increases as well the coefficient of performance however dehumidification suffers. The subcooling should be around 5 °F and the superheat should be 5 to 10 °F.

2.3 HVAC Testing

2.3.1 Data Acquisition Equipment

The HVAC system uses the same data acquisition equipment as the power monitor system. There are two pressure transducers (Figure 2-18) on the refrigerant lines to monitor suction line pressure depending whether the HVAC system is in heating or cooling mode. There are two thermocouples to measure the temperature difference across the indoor coil’s air stream. This information is used to determine the capacity of the HVAC system. Then the power usage of the HVAC system is monitored as a part of the whole home power monitoring system. A hot wire anemometer (Figure 2-19) was used to determine air velocities of the airflow out of the diffusers into the space.

Figure 2-18. Omega pressure transducer (Omega, 2010).

Figure 2-19. Hot wire anemometer (Extech, 2010).

2.3.2 Controls

The HVAC system is controlled through a program on a touch screen monitor in the hallway of the Gable Home. An electrical engineering student on the Solar Decathlon team coded the program in LabView. The suction line pressure transducer controls the speed of the compressor. The program tries to maintain a 40 °F evaporator temperature so the program will speed up or slow down the compressor to maintain that temperature. This happens because the speed of the refrigerant flowing through the evaporator will change the coil temperature and thus the pressure of the evaporator. This proved to be a very simple way to control the speed of the compressor without setting up a complex set of equations or a large database to define the operation of the compressor.

2.3.3 Testing Method

During the summer prior to the competition, the HVAC system was tested in the house to determine ideal operation and reliability. Much of the summer was spent redesigning the HVAC system to provide acceptable operating conditions; however, reliability was determined during this time. The competition provided a good test bed to determine the performance characteristics of the HVAC system. The simulated living schedule allowed for more realistic loading conditions compared to the summer testing. Data was collected using the data acquisition unit. The volumetric airflow was taken at various points at the diffusers then averaged to account for locations of higher and lower velocities.

2.4 Indoor Air Quality

2.4.1 Equipment and Method

To help determine the differences in air quality, the concentrations of carbon dioxide and radon were measured in five residences, two airtight houses and three typical houses. While the airtight houses were not the Gable Home, they are both Passive House certified homes. This means they had to meet the same certification guidelines as the Gable Home, therefore, a parallel can be drawn between the houses with regard to the indoor air quality. All five of the residences are located in Champaign and Urbana, Illinois. The three typical houses are representative of typical residential homes in the Champaign/Urbana area.

At each location, the concentration of carbon dioxide was measured, using the Telaire 7001. Measurements were taken in the living room and outside to find the atmospheric conditions at the time of the test. The concentrations were recorded at intervals to determine when the machine was nearing a steady state. To measure the radon concentration, the SafetySiren pro series 3 radon gas detector was used. To get an accurate reading, the radon gas detector had to be placed in the house for at least two days.

Figure 2-20. Telaire 7001.

Figure 2-21. SafetySiren pro series 3 radon gas detector.

For each house, a meeting was setup so the carbon dioxide could be measured and the radon detector could be left with the homeowner. Then a second meeting with the homeowner was required so the detector could be retrieved and the measurements recorded.

Chapter 3: RESULTS AND DISCUSSION

3.1 HVAC Results

3.1.1 Case Study – Gable Home

For the Gable Home, the HVAC system was installed as designed using 100% fresh air for conditioning and both heat exchangers in the interior conditioned space. Unfortunately, when testing the system, the head pressure was too high as well as the superheat. This was due to inadequate airflow across the condenser, which was not able to reject enough heat. The inadequate airflow stemmed from having balanced airflow on the supply and exhaust sides of the ERV, which was used as the only fans in the system. In addition, the HVAC system could not cool the space and in fact was heating up the space while running during the summer. This problem is due to the airflow issue again since the capacity is reduced significantly because the condenser cannot reject the required amount of heat and is due to the ductwork around the condenser not being insulated, causing all the heat removed by the evaporator to be radiated into the space again.

To resolve these issues, the ductwork around the condenser was placed in the unconditioned roof cap, an inline duct fan, blowing outdoor air across the condenser, was added, and a second heat exchanger was added in parallel to the first heat exchanger. Moving the condenser to the roof cap removed the heat load in the house and allowed the inline duct fan to be added to increase the overall airflow across the heat exchanger. The volumetric airflow increased from ~200 CFM to ~500 CFM. Secondly, by adding another heat exchanger in parallel, the total surface area of heat exchange with the airflow is doubled, increasing the amount of heat that can be rejected to the air stream. These changes solved the major issues related to improper superheat and head pressure.

The next major issue related to the Gable Home and specifically the competition itself was the needed to change the temperature from ambient temperature to the competition range in one hour. Under normal operation, a residential HVAC system would not need to change the indoor conditions in one hour and should be at steady state. Effectively the system needs to be oversized for the competition only. Since the compressor is a variable capacity compressor, it is able to speed up or slow down, and thus change capacity, depending on the load across the indoor coil. Thus to increase the capacity of the system, the airflow needed to be increased. This is because if the compressor can speed up or down to match the load then the delta T across the indoor coil stays constant. Thus, from Equation 1, for capacity, Q, to increase, the volumetric flow, m, has to increase.

$$Q = m * c_p * \Delta T \tag{1}$$

Where: Q – Capacity, watts

m – volumetric flow rate, m^3/s

c_p – specific heat, J/Kg-K

ΔT – Temperature, K

The airflow increase was accomplished through an inline duct fan adding recirculation indoor air across the indoor coil in conjunction with the outdoor fresh air. This took the volumetric flow rate from ~200 CFM to ~550 CFM when the ERV was at 100% speed. Another benefit to the added recirculation loop is the ability to use no fresh air but still provide conditioning to the interior space. Without the ERV running, the HVAC system is still able to provide ~350 CFM of airflow. This is advantageous because the house only requires 21 CFM of mechanical ventilation if continuously ventilating, however, the lowest setting for the ERV is 70 CFM. Thus, the ERV needs to cycle approximately 20 minutes every hour at the lowest setting for proper indoor air quality. Before at 200 CFM (highest speed) and 100% fresh air, the system could only condition for approximately six minutes before the space would be over ventilated, causing excess energy consumption. Adding the extra inline duct fans did increase the energy use of the HVAC system however. The added energy consumption of these two fans is 153 W for the inline duct fan in the roof cap and 72 W for the fan added for the indoor recirculation loop with being energy star rated.

3.1.2 Competition Results

For the weeklong competition, both the Department of Energy and Team Illinois collected data of Gable Home's performance. First, in Figure 3-1 and Figure 3-2, the indoor and outdoor temperatures for the week can be seen respectively. Many aspects related to the performance of the home can be taken from these two figures. First attesting to the well performing nature of the thermal envelope, at night even when the outdoor temperatures were in the high 40s to low 50s without the HVAC system running, the home only lost 1-2°F during the night. This truly shows how super insulating and super sealing a home can help to offset the need for a large HVAC system. The gaps in Figure 3-1 and Figure 3-3 are due to the public tours required as a part of the competition so the doors were fully open for those hours.

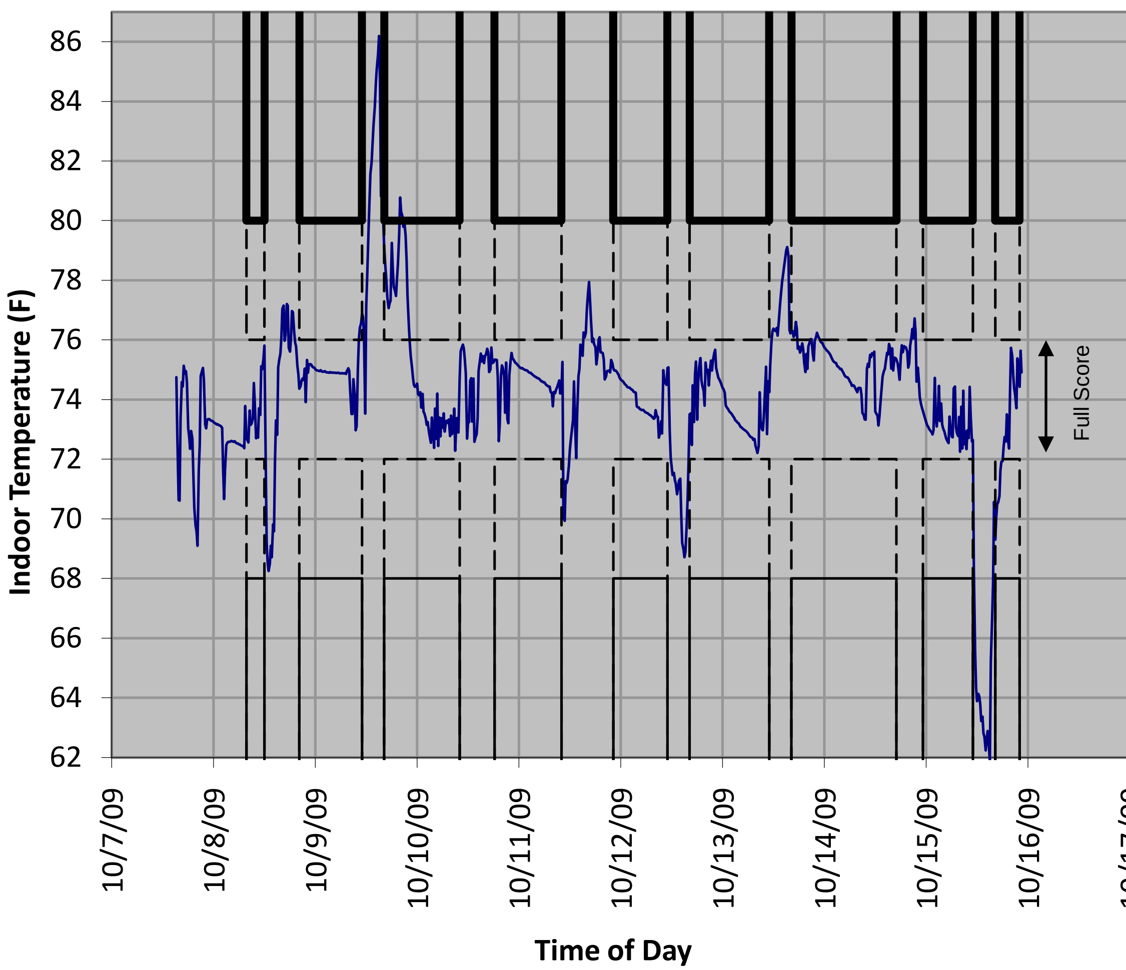


Figure 3-1. Competition week indoor temperature.

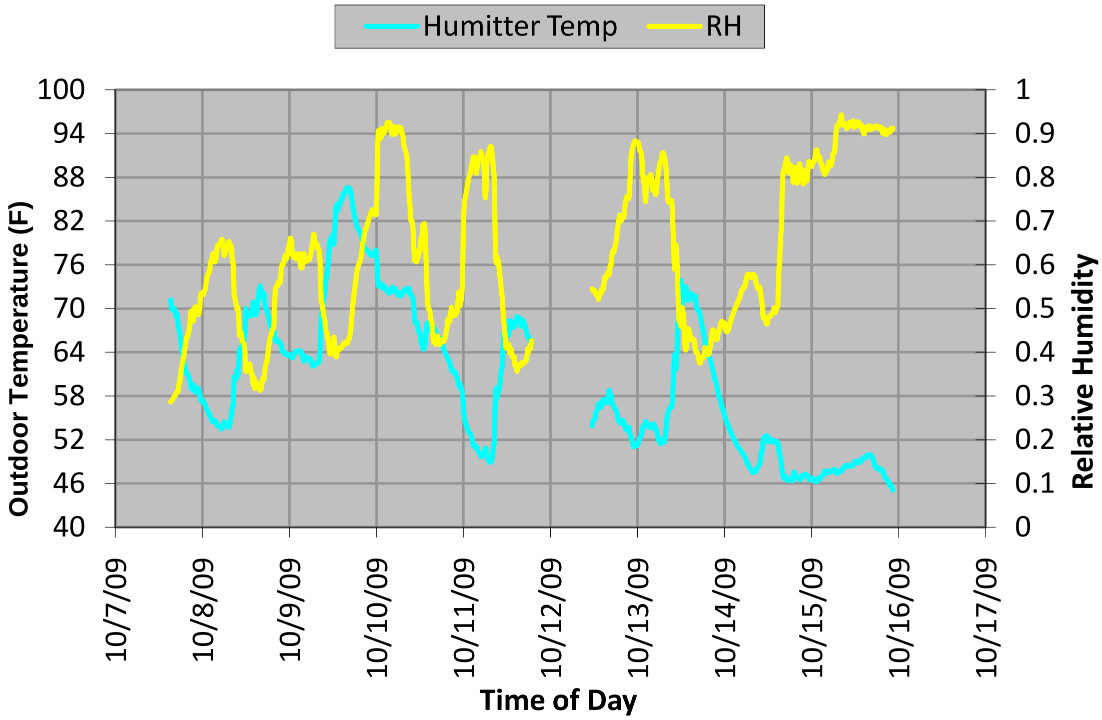


Figure 3-2. Competition week outdoor temperature and humidity.

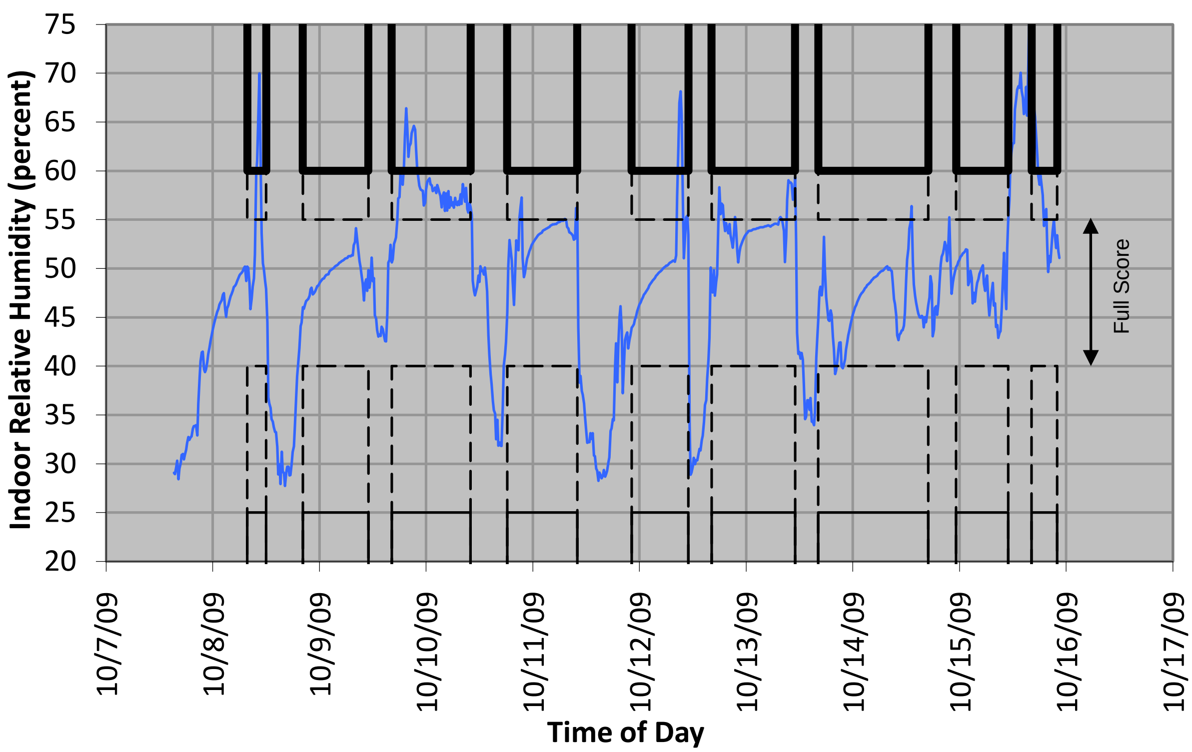


Figure 3-3. Competition week indoor humidity.

The HVAC system performed well during the competition for the limited time it was required. It was used enough to get valuable data however. There was only one day during the competition where the weather was hot and humid, reaching a high of 86°F and 42% relative humidity. On this day, cooling was required after the day's public tours. The indoor temperature and humidity matched the outdoor temperature and humidity at the conclusion of the public tours. The HVAC system and heat pump hot water heater both ran at maximum capacity to overcome the load in the interior space. In Figure 3-4, the cooling capacity of the HVAC system and the power usage of the HVAC system can be seen. From the design day energy modeling, the HVAC system should only need to output ~1800 watts of cooling capacity to maintain temperature, however this is only to cancel the effects of solar heat gain, heat gain through the envelope and internal heat gains. The model assumes the temperature began the day at the desired setpoint and internal mass is at the same setpoint temperature and humidity level. This is not the case during the competition. The HVAC system needs to overcome all the same loads as the design day in addition to increased sensible and latent loads in the indoor air and internal mass. The HVAC system put out 4,000 to 5,000 watts of cooling capacity and the heat pump hot water heater put out about 1800 watts of cooling capacity. Thus, the house had over three times the cooling capacity than required for design day at steady state. In two hours, the indoor temperature dropped by 9 °F, which is a huge pull down for a home, however, for the competition that was still not quick enough unfortunately. In the end, it took nine hours to go from an indoor temperature of 86 °F to 74 °F with two hours where the HVAC system was not run. This is a very competition specific issue since in a normal home the house would be at steady state and would not require as much cooling capacity. This also shows the amount of internal mass a Passive House has since even triple the cooling capacity could not bring down the temperature quick enough.

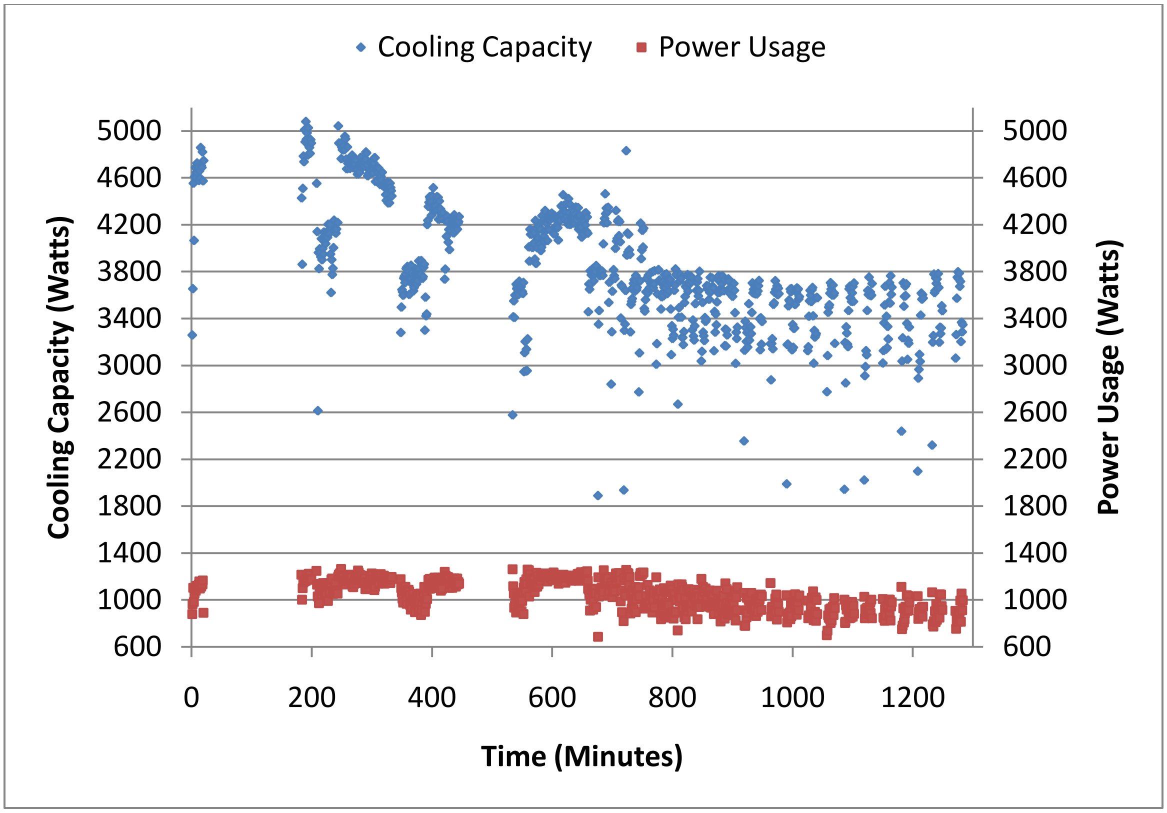


Figure 3-4. Competition week cooling capacity.

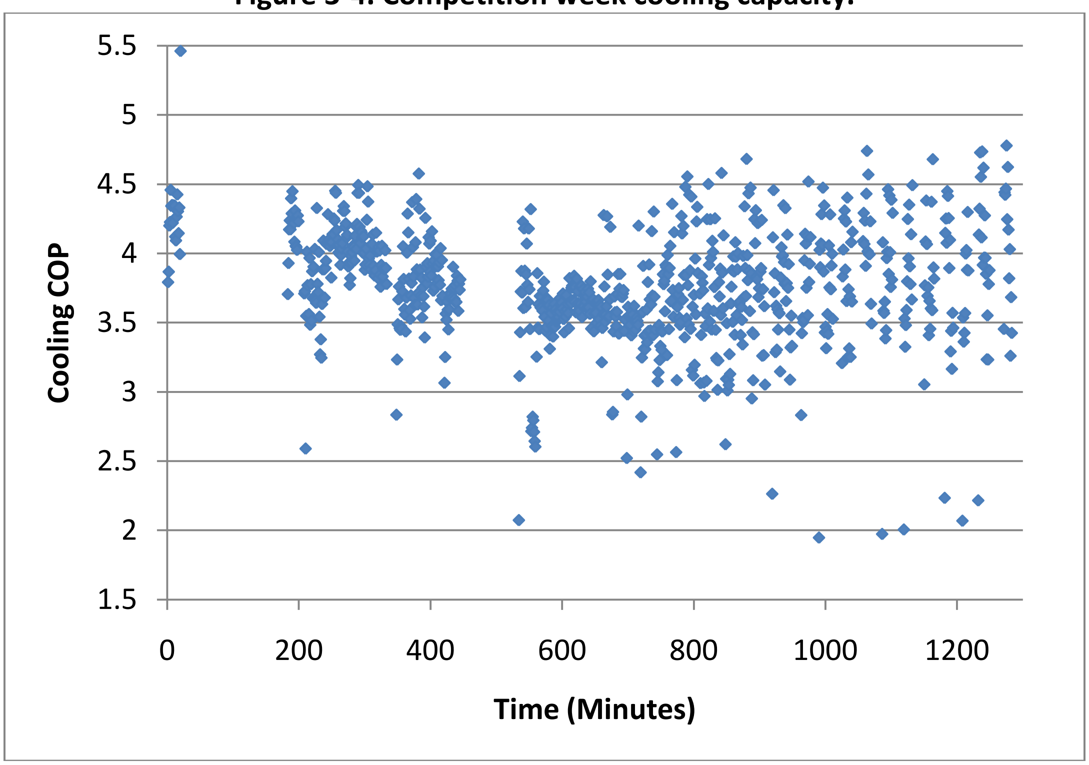


Figure 3-5. Competition week cooling COP.

For the rest of the competition the outdoor temperatures were very cold. Now the very things that hurt the home during cooling actually help during heating. The high internal mass helped to maintain a constant indoor temperature with minimal heating input. In Figure 3-6, the heating capacity and power usage can be seen. The heating capacity and power usage is lower than that for cooling. The HVAC system was able to heat the house much quicker than it was able to cool the house. This is partially because heating is only a sensible heat gain while cooling removes both sensible and latent heat.

The cooling and heating coefficients of performance, COP, can be seen in Figure 3-5 and Figure 3-7 respectively. These COPs are calculated from the cooling and heating capacities and the total system power used in Equation 2 and Equation 3 respectively. In the figures, the COPs follow the usage of the HVAC system in the competition.

$$COP_{cooling} = \frac{Q_{Cooling}}{W} \tag{2}$$

$$COP_{heating} = \frac{Q_{Cooling}}{W} + 1 \quad \text{or} \quad COP_{heating} = \frac{Q_{heating}}{W} \tag{3}$$

Where: Q – Capacity, watts

W – Work, watts

COP – Coefficient of Performance, W/W

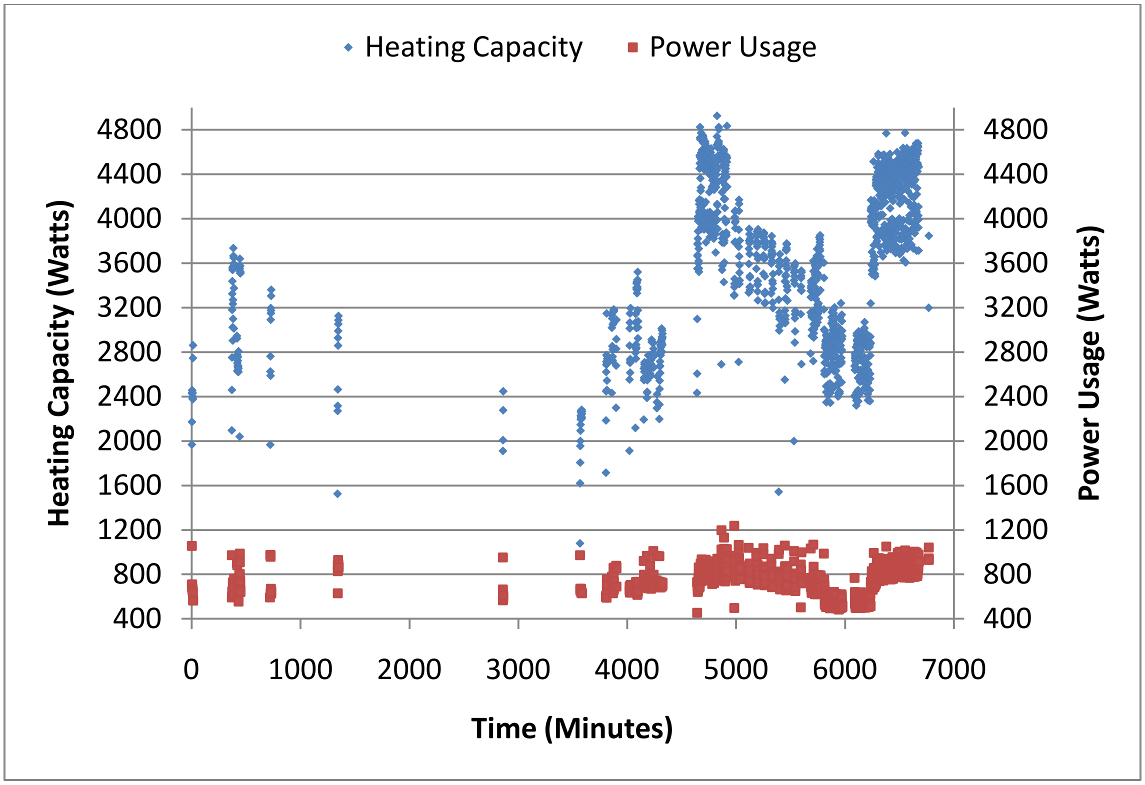


Figure 3-6. Competition week heating capacity.

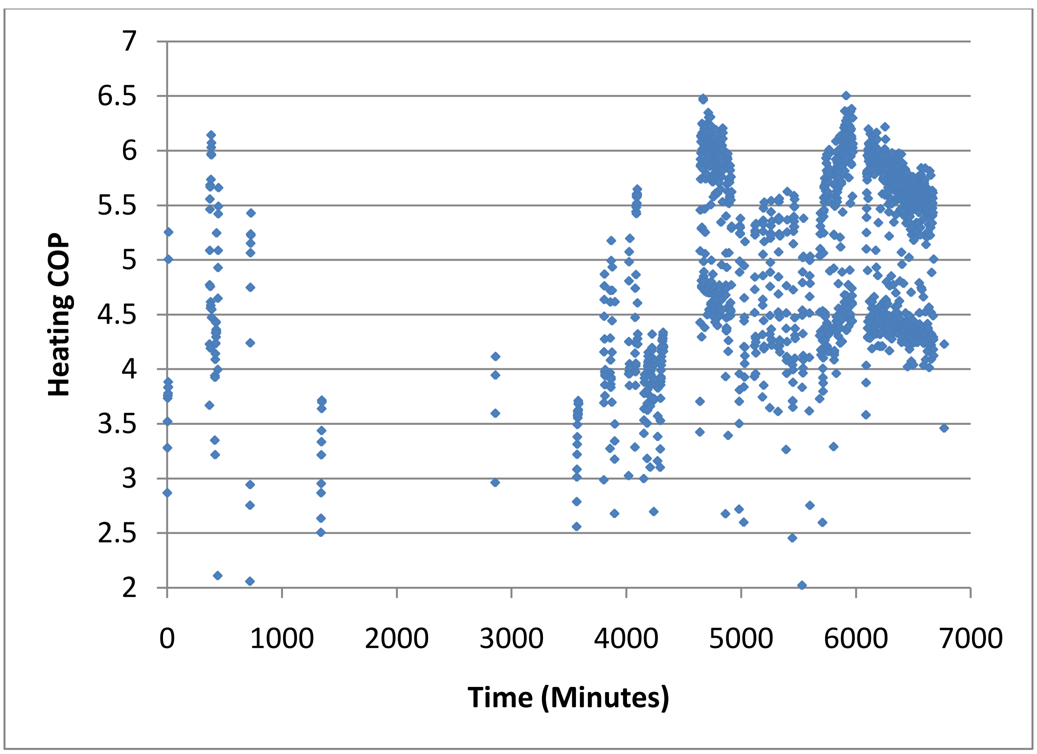


Figure 3-7. Competition week heating COP.

Table 3-1. Competition week cooling.

	Max Cooling Capacity (W)	Max Cooling COP	Max Cooling Power (W)	EER	SEER	Max Cooling Capacity (Tons)	Max Cooling Capacity (BTU/hr)
Average	4398.47	3.83	1169.47	13.1	15.0	1.25	15020
Max	5079.65	5.46	1265.53	18.6	21.3	1.45	17347
Min	3258.17	3.06	879.34	10.5	12.0	0.93	11126
Median	4320.29	3.79	1180.10	12.9	14.8	1.23	14753
Stdev	292.93	0.30	51.66	1.0	1.2	0.08	1000

	Cooling Capacity (W)	Cooling COP	Cooling Power (W)	EER	SEER	Cooling Capacity (Tons)	Cooling Capacity (BTU/hr)
Average	3529.74	3.72	976.38	12.7	14.5	1.00	12054
Max	4237.51	4.78	1262.25	16.3	18.6	1.21	14471
Min	1888.29	1.95	685.80	6.6	7.6	0.54	6448
Median	3626.60	3.73	973.51	12.7	14.6	1.03	12385
Stdev	331.20	0.47	97.24	1.6	1.8	0.09	1131

	Total Cooling Capacity (W)	Total Cooling COP	Total Cooling Power (W)	EER	SEER	Total Cooling Capacity (Tons)	Total Cooling Capacity (BTU/hr)
Average	3861.34	3.77	1050.08	12.9	14.7	1.10	13186
Max	5079.65	5.46	1265.53	18.6	21.3	1.45	17347
Min	1888.29	1.95	685.80	6.6	7.6	0.54	6448
Median	3742.77	3.76	1065.35	12.8	14.7	1.07	12781
Stdev	528.01	0.41	125.17	1.4	1.6	0.15	1803

Table 3-2. Competition week heating information.

Total	Heating Capacity (W)	Heating COP	Heating Power (W)	EER	Heating Capacity (BTU/hr)
Average	3576.10	5.07	733.33	16.65	12212
Max	4925.87	6.50	1197.04		16821
Min	1525.28	2.02	481.74		5209
Median	3638.80	5.35	759.61		12426
SD	738.08	0.84	131.05		2520

Competition Week only	Heating Capacity (W)	Heating COP	Heating Power (W)	EER	Heating Capacity (BTU/hr)
Average	3532.77	4.80	756.13	15.96	12064
Max	4925.87	6.48	1197.04		16821
Min	1525.28	2.06	555.11		5209
Median	3569.95	4.71	753.54		12191
SD	847.41	0.99	94.10		2894

Steady State	Heating Capacity (W)	Heating COP	Heating Power (W)	EER	Heating Capacity (BTU/hr)
Average	3636.04	5.34	709.40	17.50	12417
Max	4772.28	6.50	1051.39		16297
Min	2319.95	3.46	481.74		7922
Median	3859.73	5.60	775.35		13181
SD	743.68	0.66	156.17		2540

Table 3-3. HVAC system volumetric airflows.

Living Room	Max Cooling	Cooling	Heating
CFM	303	195	195
m^3/s	0.14	0.09	0.09
Bedroom			
CFM	233	163	163
m^3/s	0.11	0.08	0.08
Total HVAC			
CFM	535	358	358
m^3/s	0.25	0.17	0.17

In Table 3-1, the important information regarding the performance of the HVAC system during cooling can be seen. In the table, max cooling is the name for when the ERV ran at full capacity and the recirculation inline duct fan ran at full capacity as well. This provides the greatest airflow across the indoor coil to cool the interior space the quickest. In the table, cooling means the ERV does not run at all and only the recirculation inline duct fan runs and total cooling is the combination of both max cooling and cooling data. In Table 3-1 and Table 3-2, all of the values in each column are independent of each other. The given COP is not derived from the cooling capacity and power consumption given for the same row. The values come from a larger dataset. For example, the maximum COP could occur at a different time than the maximum cooling capacity or power consumption.

The same information can be seen in Table 3-2 but for heating in this case. In Table 3-2, the first group includes all heating data during the competition week plus two days after the competition week. The second group is the heating data for just the competition week. Finally, the third group is the heating data for the day after the competition week where the HVAC system was run for 12.5 hours continuously with no cycling due to the door being open for public tours. The last group provides very important insight into the advantages of the variable capacity compressor, which will be discussed later.

Overall, the HVAC system performed very well for a student built system. Looking at the total cooling data in Table 3-1, the average COP was 3.77, which corresponds to a SEER rating of

~15. The SEER rating federal minimum for commercially sold air conditioning units is SEER 13. To qualify for an Energy Star rated air conditioner, the unit has to have a SEER rating of 14 or greater. Thus, HVAC system is better than the federal minimum as well as qualifies for an Energy Star rating.

From Table 3-2, the average COP for the competition week was 4.8, corresponding to an EER of 15.96. While the steady state values are more indicative of the performance potential of the HVAC system in heating, the competition week data is more realistic to actual usage since the steady state data does not include cycling, which consumes more power. To qualify for Energy Star rating for air source heat pumps in heating, the EER must be 12 or greater. The testing requirements require the HVAC system to go through one complete defrost cycle when measuring for EER. While the HVAC system did not go through a defrost cycle during the competition, the system should still qualify for an Energy Star rating because it is so much greater than the minimum.

From Table 3-1, the extra airflow of the ERV in max cooling mode provides an extra 900 watts of cooling capacity compared to cooling mode. This is an extra 19.8% of cooling capacity; however, this comes with a 16.5% increase in power usage.

The cooling capacity ranges from a maximum of 5079 watts to a minimum of 1888 watts for a 62.8% reduction in cooling capacity when the load is lower. The same goes for the power usage, which goes from 1265 watts to 685 watts for a reduction of 45.8%. For heating, the heating capacity ranges from 4925 watts to 1525 watts during the competition week equating to a 69% reduction. The power usage during heating ranges from 1196 watts to 555 watts, or a reduction of 54%. For both cooling and heating, the capacities and power usages more than halve when the load is less.

The true percentage of energy saved though can be seen in Table 3-4 where for the competition week if a fixed capacity compressor were used instead of the variable capacity compressor, the energy use would have been 17.35 kWh. This value was found by taking the maximum power consumption of the compressor over the competition and using that number for every minute the compressor was operating. This is not a true representation of a fixed capacity compressor however because the compressor will be on less often compared to a variable capacity compressor. This is due to the fact that the fixed capacity compressor is always functioning at peak capacity. Since a fixed capacity compressor was not tested in the same house this is the best estimation of the energy usage. By using the variable capacity compressor though, 2.92 kWh was saved or 16.81%. The savings in heating energy was even greater than for cooling with 36.83% energy saved. That brings the total HVAC savings to 24.32% compared to a fixed capacity compressor. In the scheme of the competition, the 6.75 kWh of saved energy by using the variable capacity compressor gained the team an extra three points out of 1000. That may not seem like many points, but Team Illinois lost to Team Germany by nine points.

Table 3-4. Competition week HVAC energy saved.

	Max Cooling	Cooling	Total Cooling	Heating	Total HVAC
Max Energy Use (kWh)	6.69	10.67	17.35	30.03	47.38
Energy Saved (kWh)	0.50	2.42	2.92	11.63	14.55
Percentage Saved (%)	7.49%	22.65%	16.81%	38.74%	30.71%

Competition only

	Max Cooling	Cooling	Total Cooling	Heating	Total HVAC
Max Energy Use (kWh)	6.69	10.67	17.35	10.41	27.77
Energy Saved (kWh)	0.50	2.42	2.92	3.84	6.75
Percentage Saved (%)	7.49%	22.65%	16.81%	36.83%	24.32%

Steady State

	Heating
Max Energy Use (kWh)	13.32
Energy Saved (kWh)	4.34
Percentage Saved (%)	32.57%

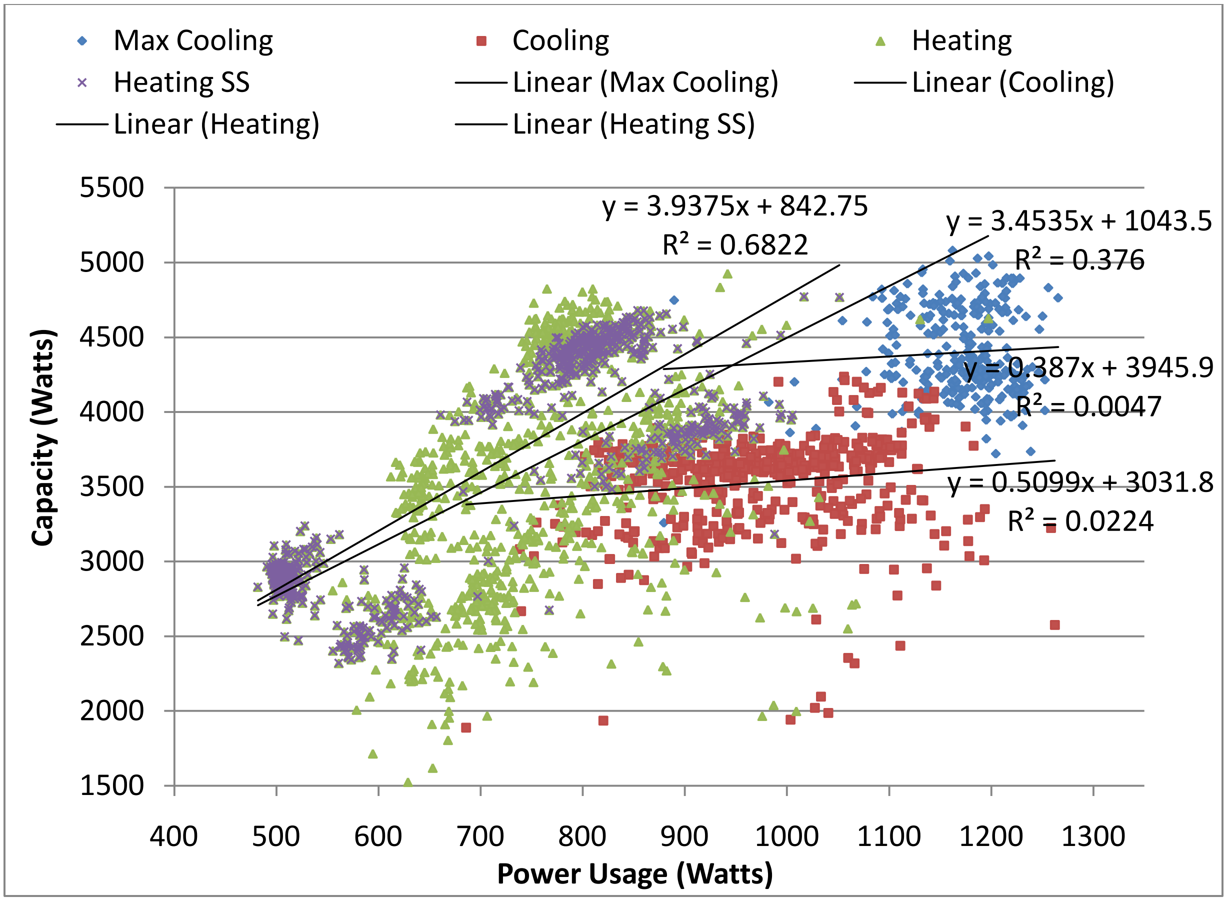


Figure 3-8. Competition week power usage versus capacity.

Figure 3-8 shows graphically how heating scales better than cooling for the variable capacity compressor. The trend line for heating has a larger sloping, meaning the HVAC system has a higher capacity increase than the corresponding increase in power usage. For heating, the capacity increases 3.5 watts for every watt of power increase. However, for cooling, the capacity only increases 0.5 watts for every watt of power increase. A majority of the difference between the two values can be attributed to the added capacity due to the compressor during heating. This means the HVAC system requires less heat from the evaporator if ~20% of the heating capacity is due to the heat added by the compressor.

These results are favorable and show that even in a small HVAC unit where the absolute savings might be smaller, the percentage savings scales down from larger residential and commercial HVAC systems. If residential air conditioners all used variable capacity compressors then the savings would be comparable to going to SEER 16 air conditioner from a SEER 13.

3.1.3 Winter Results

From December 14th 2009 until January 24th 2010 the Gable Home was monitored while on the University of Illinois campus. Figure 3-9 shows the indoor temperature and humidity for the bedroom and living room. During the monitoring period, the HVAC system was never turned on, there was not occupancy, and there were no internal loads except for the inverters for the solar panels. This means the indoor temperature will eventually settle on a temperature range dictated by the outdoor temperature and the solar insolation. This range can be seen in Figure 3-9 during the month of January where the temperature stays roughly between 40 to 55 °F. This is despite the fact that the outdoor temperature during this same period is -10 to 20 °F as seen in Figure 3-10. This is the benefit of a properly designed passive house. In a typical home if the HVAC system did not run during the winter, the indoor temperature would more closely follow the outdoor temperature. The Gable Home is acting like a giant thermos keeping in all the solar heat gained during the day and slowly releasing it during the night. The solar heat gain can really be seen in Figure 3-9 where the living room temperature is always two to three degrees higher than the bedroom. This is due to the two large windows in the living for the most solar heat gain possible.

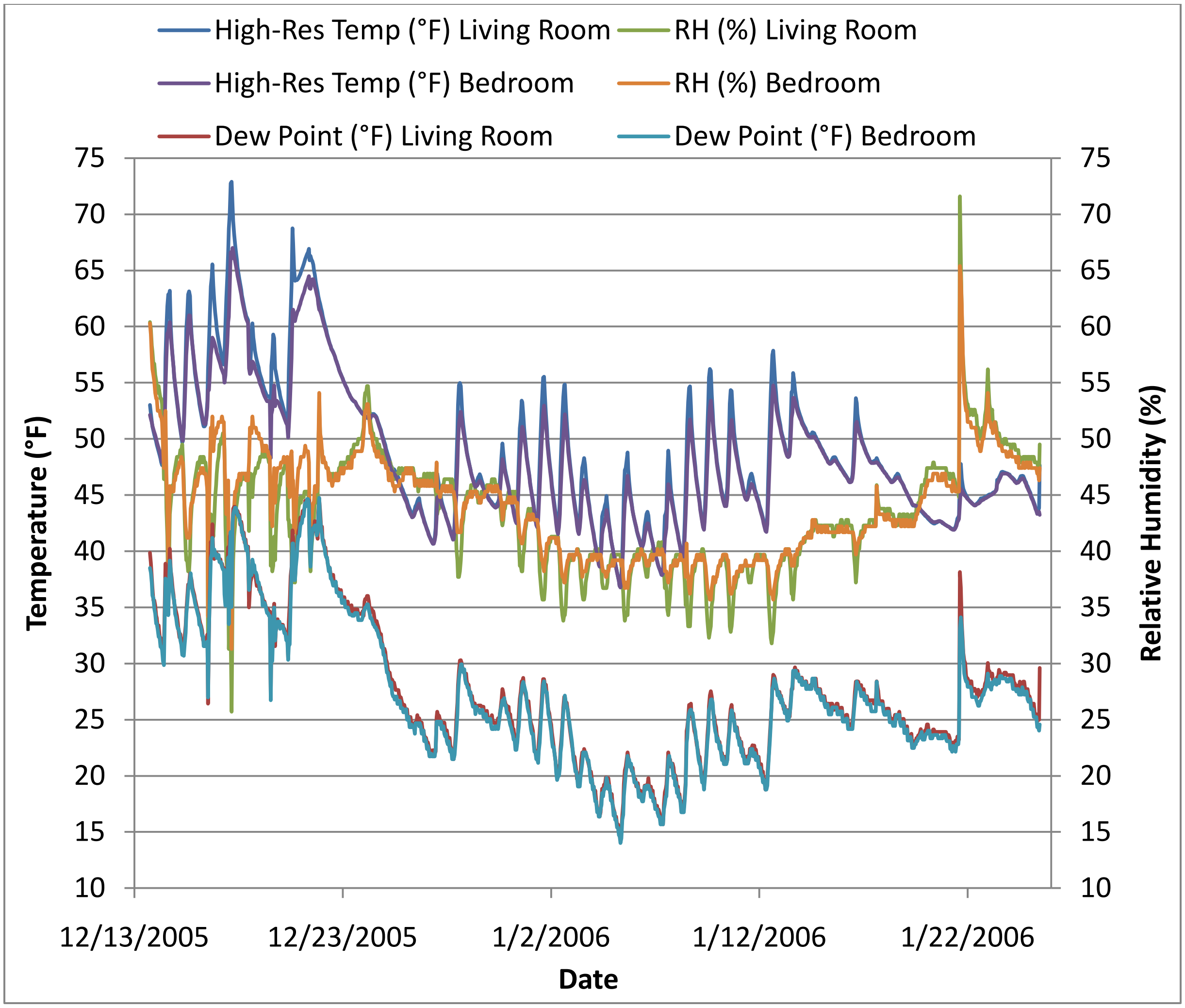


Figure 3-9. Winter indoor temperature and humidity.

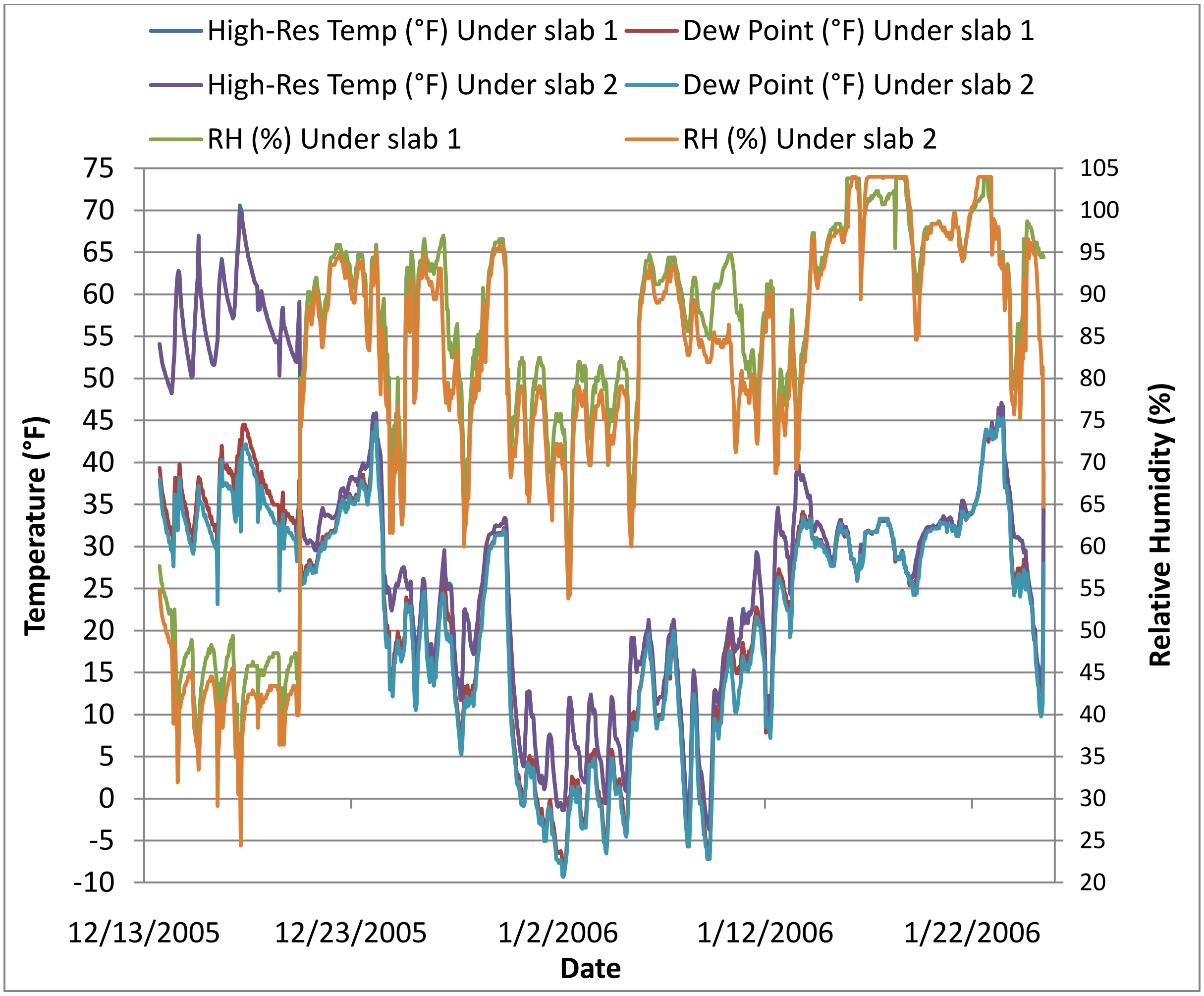


Figure 3-10. Winter outdoor temperature and humidity.

3.2 IAQ Results

3.2.1 Passive Houses

The comparison between the two airtight homes is quite interesting because they use the same construction methods, have no basement, and are very similar homes in size and design. The major difference between the two houses is airtight house one did not have a functioning ventilation unit at the time of testing while airtight house two did have a function unit. The ventilation unit is required for these kinds of homes because they are airtight compared to conventional construction homes. Without a ventilation unit, the occupants would become seriously ill from accumulated pollutants, especially carbon dioxide and radon.

First, the radon levels between the two airtight homes are completely different. Airtight home one had the highest radon concentration of all the homes tested, while airtight home two had the lowest radon concentration, as seen on Figure 3-13. Without the ventilation unit operating in airtight home one, the radon levels are above the EPA recommended maximum level. This adversely affects the health of the occupants and is a major contributor to the poor indoor air quality of the home. In airtight house two, however the radon level is well below the maximum level. The dramatic difference between the two houses shows how the ventilation unit is required to maintain acceptable indoor air quality. In addition, compared to the conventional homes, the mechanical ventilation is desirable compared to natural infiltration because you have continuous and controlled ventilation of the home. There is also an energy savings with having the house ventilation unit indirectly control radon concentration because you do not need to have an auxiliary fan for only controlling radon, which are typically installed in conventional homes. In addition, the exhausting air, which contains the radon, exchanges heat and moisture with the fresh intake air instead of just venting to the atmosphere. These are energy advantages the airtight homes employ to lower energy costs but also maintain a higher indoor air quality.

Secondly, the CO2 levels follow similar trends as the radon levels in the two airtight homes. Again because airtight home one did not have a functioning ventilation unit, the CO2 levels were elevated, the second highest of the tested homes, as seen on Figure 3-12. The CO2 levels would have kept increasing as well since there were four people present during testing and there was very little fresh air exchange. It would take about 25.5 hours for one complete air exchange of the home from natural infiltration alone. This infiltration level is too slow compared to the generation levels of CO2 from four people. Using Equation 4 from *Indoor Air Quality and HVAC Systems* with a generation of 0.011 cfm/person from *ASHRAE Fundamentals*, after one hour in the airtight home the CO2 level inside should be 687 ppm (ASHRAE, 2009 & Bearg, 1993). Now for this infiltration, the CO2 will reach steady state in 151.5 hours, or 6.3 days, at a level of 8077 ppm as seen in Table 3-5. This is well above the acceptable range for CO2, which has a threshold limit value for a time-weighted average of 5,000 ppm. This level of CO2 begins to have serious adverse effects on the human body. Now, if the air exchange per hour is 0.35 instead the 0.033 through mechanical ventilation then the steady state level of CO2 will be 1162 ppm. This value is the limit for what is acceptable for CO2 concentrations in a home as ASHRAE recommends levels below 1000 ppm, which is why 0.35 ACH is the minimum level of acceptable ACH. This comparison shows why mechanical ventilation is required for proper indoor air quality concerning CO2. Without the mechanical ventilation, the inhabitants will begin to feel the adverse effects of high CO2 concentration in as few as 10 hours. The best way to reduce CO2 in the house is the same as radon, more ventilation. Since you are continuously mechanically ventilating the house, an airtight home will have more uniform air change as opposed to the infiltration of fresh air in a conventional home. This means the airtight home will exchange more fresh air for stale air, lowering the CO2 concentration level in the home.

Table 3-5. Carbon dioxide generation CO2.

	Airtight	**Minimum**
	0.033 ACH	**0.35 ACH**
Time (hr)	**C_{inside} (ppm)**	**C_{inside} (ppm)**
0.5	563	553
1	687	651
5	1607	1036
10	2599	1140
50	6652	1162
100	7846	1162
150	8075	1162
151.5	8077	1162
200	8119	1162
720	8129	1162

$$C_{indoor} = \frac{F \times 10^6}{V_{eff} I}(1 - e^{-It}) + C_{outdoor} \tag{4}$$

Where: C_{indoor} – CO2 concentration indoor, ppm

F – Generation rate of CO2, 0.011 cfm/person

V_{eff} – Effective volume, ft^3

I – Ventilation rate, air changes per hour

t – Time, hours

$C_{outdoor}$ – CO2 concentration outdoor, ppm

3.2.2 Normal Home

The typical residential houses had noticeable differences in air-quality even compared to each other. There was no measurement for the radon in typical house one; however, there were results for the other two typical houses refer to Figure 3-13. The radon for typical house three had a much higher radon concentration than typical house two. Both houses had their basement doors open so that would partially explain why both levels were near the recommended maximum levels of radon since radon tends to accumulate through the basement. The residents in typical house three mentioned that their basement door was always left open for the cat to go between the basement and the main two floors. This possibly explains why the radon concentration was higher in house three than house two. Further testing is recommended because radon can be a serious health concern.

In terms of carbon dioxide, house two was considerable lower than the other typical houses refer to Figure 3-12. This house had fewer in habitants as well as no pets compared to the other two houses. The other two houses had the same number of inhabitants and both had pets. Typical house one had a higher carbon dioxide concentration and its inhabitants were all adults compared to only two of typical house three. Also, typical house one had more pets including a dog, cats, sugar gliders, a bird, and some reptiles while typical house number three only had two cats and a reptile. There are many differences in the between the two houses and could have easily been the cause of the carbon dioxide differences.

3.2.3 Passive House/ Normal Home

Analyzing the CO_2 it's possible to see that the house with the lowest value is the airtight house two, while the largest is the typical house one, with the largest value almost twice the smallest value. The differences between the houses can be explained by the amount of people, the pets present in each home, and the ventilation systems.

For the radon, the values obtained are high and exceed the values permitted by EPA. The airtight house two is the only one that has an acceptable value of 0.9 pCi/l. The highest value was found at airtight house one although it is easily explained by the absence of ventilation. The high concentrations for the typical houses reveal that the air exchange rate is not enough for proper removal of the radon and extra action is needed to solve this issue.

3.2.4 Threshold Limit Value-Time Weighted Averages (TLV-TWA)

The indoor air quality can be calculated to see if the homes have acceptable indoor air quality or not. Using Equation 5, which is Equation 2.10 from *Indoor Air Quality Engineering*, the normalized air contaminated concentration can be determined from the sum of the concentrations of the various pollutants over the TLV-TWA of those pollutants. In this case, there are two pollutants: CO2 and radon. The TLV-TWA for CO2 is 5000 ppm and radon is 4 pCi/L. The results of the normalized air contaminated concentrations can be seen on Figure 3-11. From the results, the house with the worst indoor air quality is airtight house one. This is the result of the large radon concentration in the home. The two typical homes with radon measurements also have poor indoor air quality since the normalized air contaminated concentrations are greater than one. Airtight house two had the best indoor air quality with a C_{nc} = 0.32. With a functioning ventilation unit, airtight house one should be near this value as well, reducing the CO2 and radon in the home. The two typical homes should look into solutions to create an acceptable indoor air quality. The best solution is to increase the ventilation rate in the home to alleviate the high radon and CO2. Both homes could install a radon fan and that alone would more than likely bring them to an acceptable indoor air quality.

$$C_{nc} = \sum \frac{C_i}{TLV_i}$$

$$C_{nc} = \frac{C_{CO_2}}{TLV_{CO_2}} + \frac{C_{Radon}}{TLV_{Radon}} \tag{5}$$

Where: C_{nc} – Normalized air contaminate concentration

TLV_i – Threshold limit value, ppm

C_i – Concentration, ppm

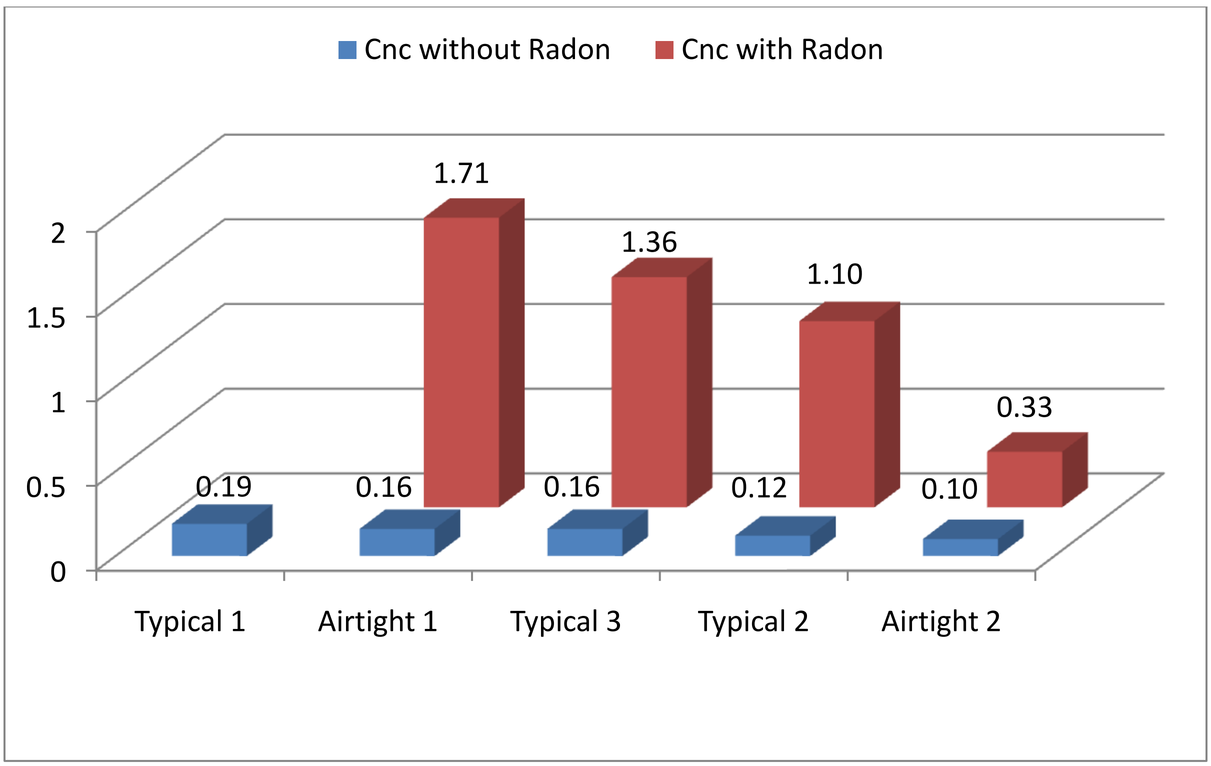


Figure 3-11. Normalized air contaminant concentration (C_{NC}).

3.2.5 CO2 Concentrations

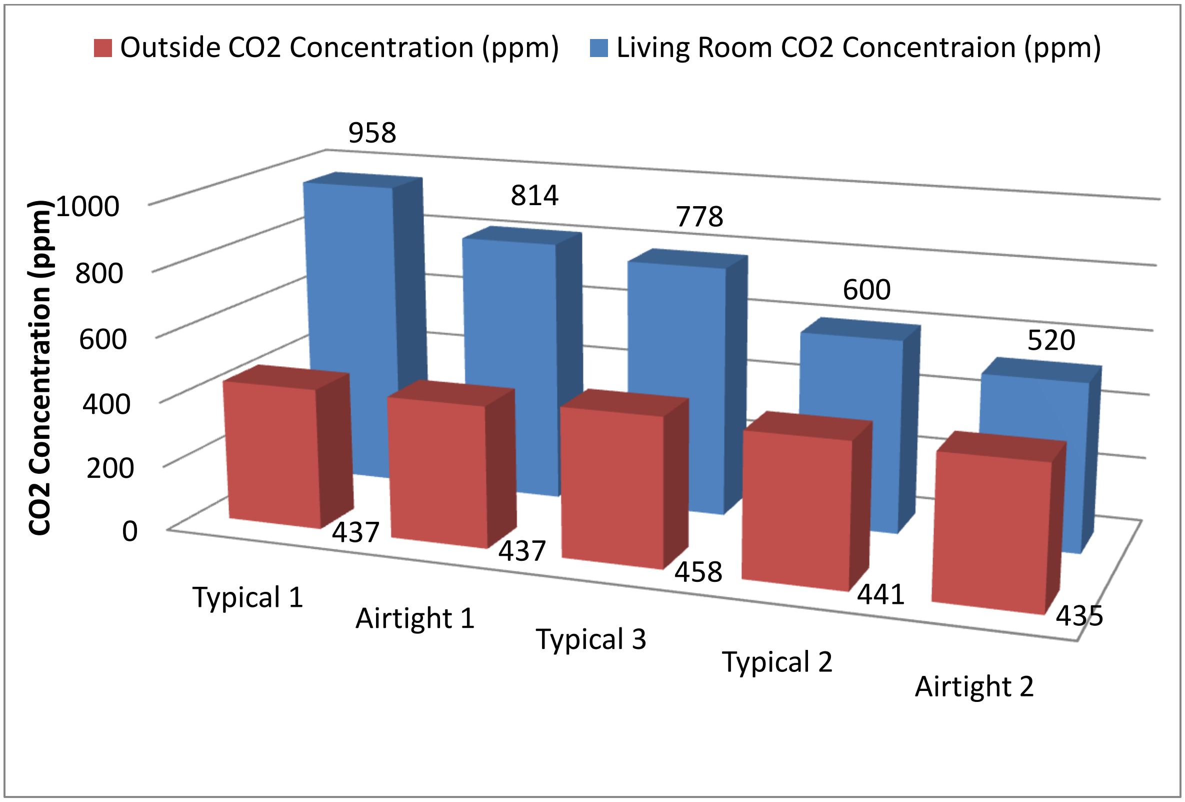


Figure 3-12. Carbon dioxide concentrations inside and outside.

3.2.6 Radon Concentrations

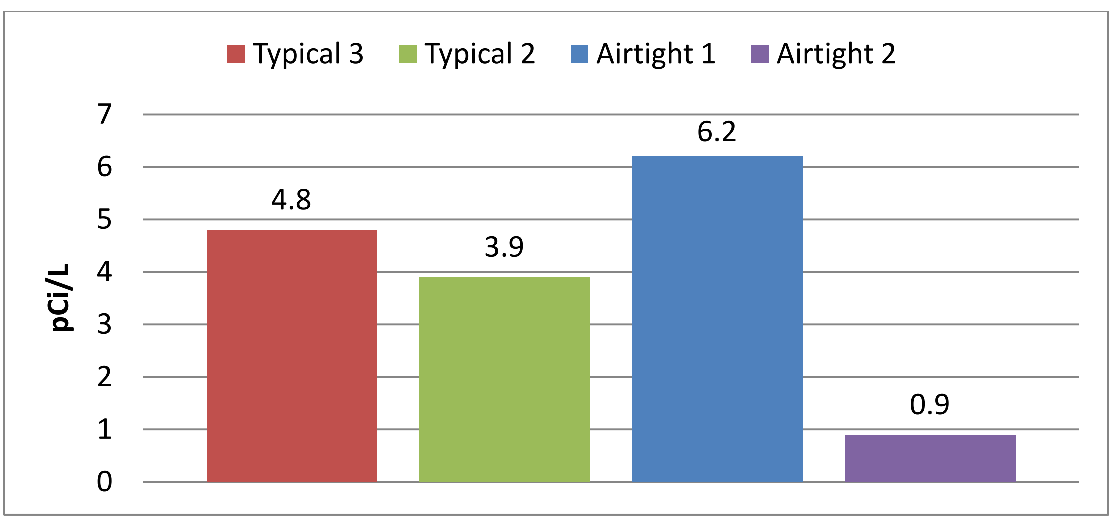


Figure 3-13. Radon concentrations.

3.3 Heat Pump Hot Water Heater

3.3.1 Integration with HVAC System

One advantage of a custom HVAC system is the ability to integrate with other technologies in the Gable Home. The more system that are integrated, the more energy can potentially be saved. This was the case with the heat pump hot water heater. The heat pump water heater purchase came with a duct adapter to take the outlet air and divert it into ductwork then integrated into the exhaust ductwork from the ERV.

3.3.2 Advantages

By integrating the heat pump hot water heater ductwork into the exhaust ductwork of the ERV, the cooling effect of the heat pump can be diverted away from the interior space. This is important in the winter months where the cooling effect of the heat pump can be as large as internal gains, effectively negating them. In the winter, the internal gains are beneficial and desired because then the HVAC system does not need to run as often, saving energy. However, the opposite is true in the summer when no internal gains are wanted. This is when the cool effect of the heat pump hot water heater is desired. Therefore, by integrating into the HVAC system and controlling the flow of the heat pump water heater's airflow through electronic dampers, the cooling effect can be taken advantage of when cooling is required in the space and diverted to the exterior when it is not required.

3.4 Solar Decathlon

3.4.1 Overall Rank

Overall, the Gable Home did very well in the 2009 Solar Decathlon competition, placing 2nd out of 20 teams. The competition was fierce for the weeklong event; however, in the end, Team Germany took first place. The Gable Home took first place in Hot Water, Home Entertainment, and Appliances and took second place in Lighting, Comfort Zone, and Energy Balance. Therefore, the Gable Home placed very well in all the objective categories. Where the team did not do as well was in the subjective categories.

3.4.2 Comfort Rank

Team Illinois placed 2nd behind Team Germany in Comfort Zone, however it was only by fractions of a point. In Figure 3-14, the indoor temperature data for the competition can be seen of Team Germany compared to Team Illinois. Overall, the Gable Home was a much better performing house thermally. The temperatures during the scoring periods were much more consistent compared to Team Germany. This is especially true during the nights where the Gable Home would only lose a few degrees with the HVAC system running while Team Germany either had to run their HVAC system or risk falling out of the scoring zone. Team Germany did do better at humidity control than Team Illinois looking at Figure 3-15.

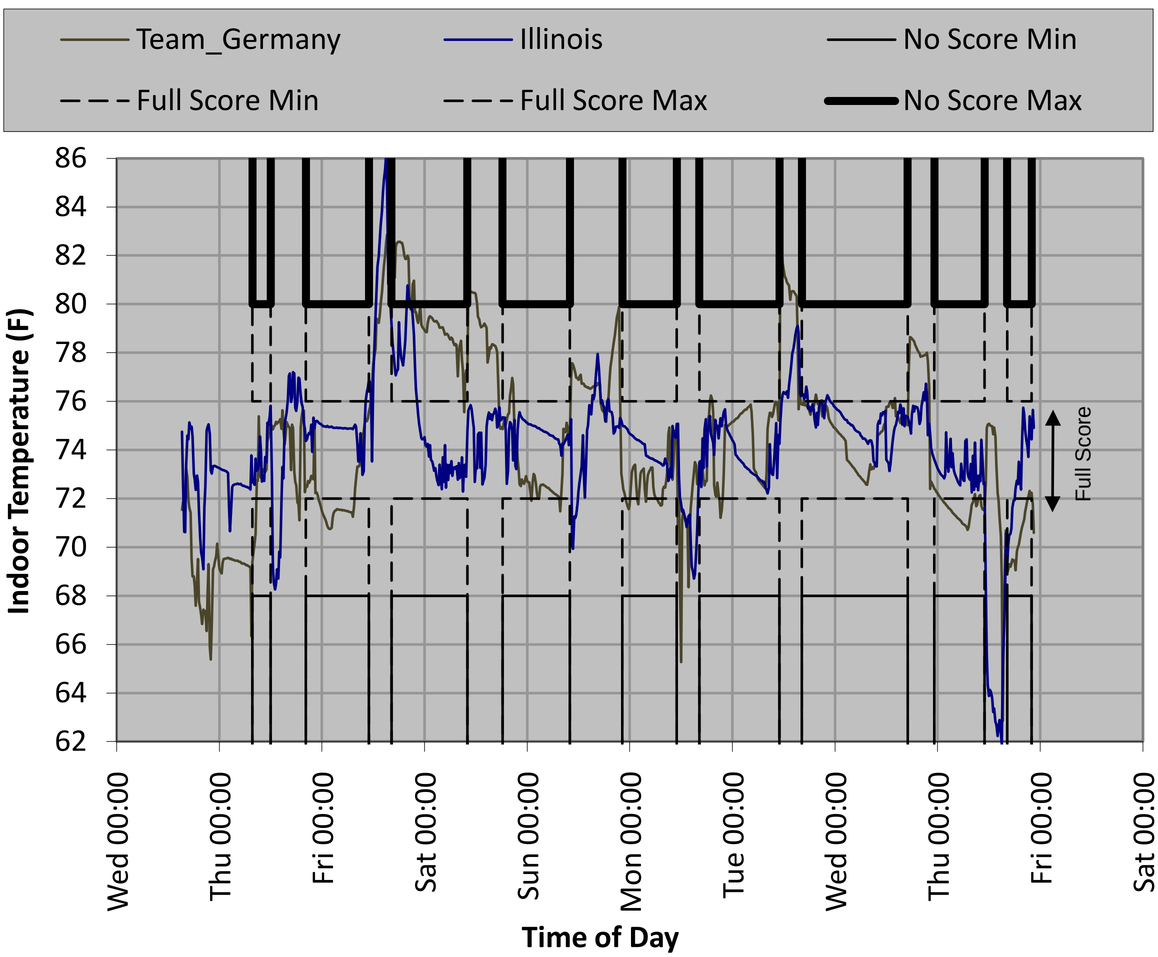


Figure 3-14. DOE indoor temperature.

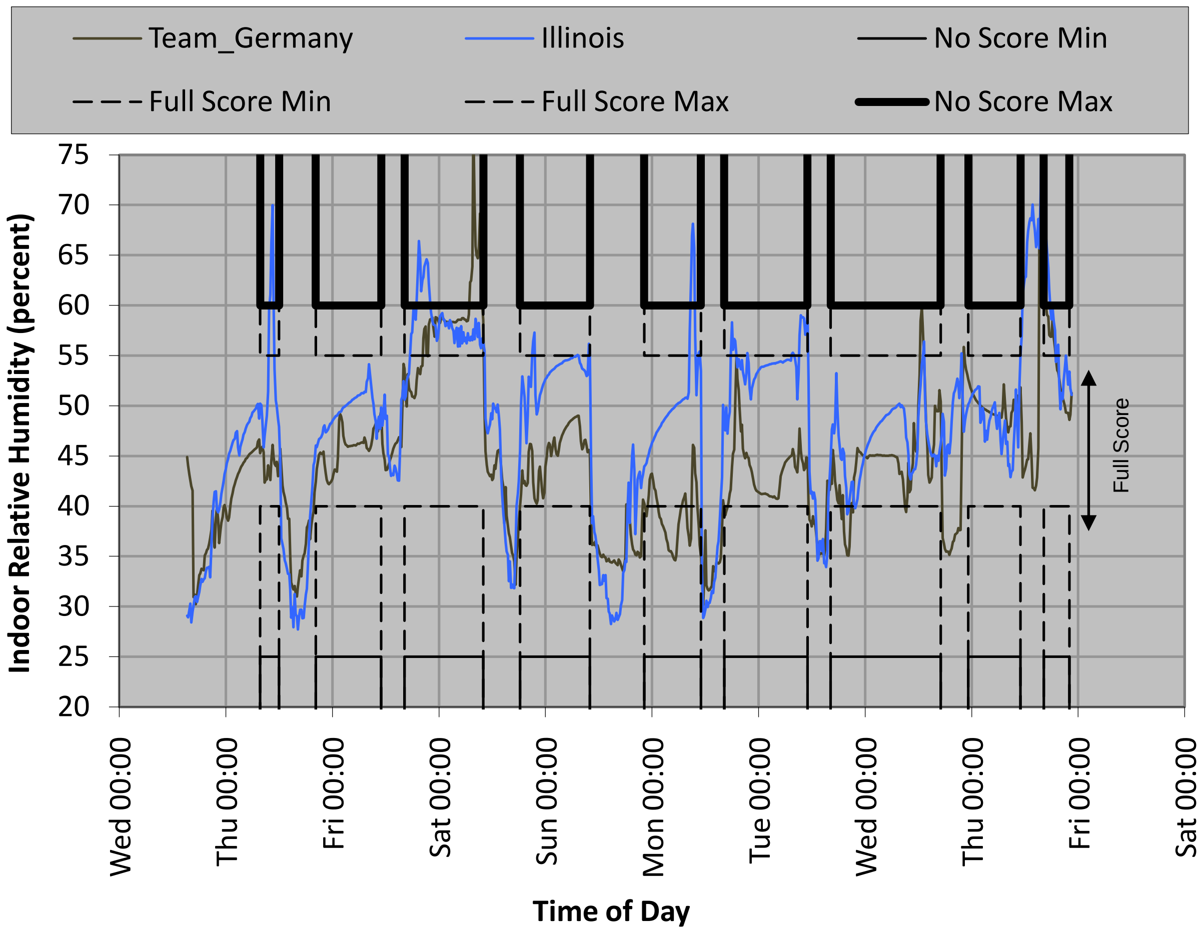


Figure 3-15. DOE indoor relative humidity.

3.4.3 Energy Balance

For the 2009 Solar Decathlon competition, the Department of Energy for the first time decided to make all the houses grid-tied. Before this competition, all the homes were connected to batteries. Grid-tied photovoltaics are much more realistic to how a typical house would be connected. By having the houses grid-tied for the competition brought many advantages and some disadvantages as well. Teams no longer have to buy and maintain expensive battery banks. Also all power produced can be accounted for and none of it is lost, which is what would happen if the battery banks were full. A disadvantage though is that after teams go below -10 kWh on energy balance, they get zero points for the Energy Balance event. This means they have little motivation to conserve energy anymore. This can be seen in Figure 3-16 where some teams used more power from the grid that any of the produced in excess.

From Figure 3-16, Team Germany and Team Illinois by far had the most excess generation of all the teams. What is interesting is the approach both teams took to reach these values. Team Germany had a house that had PV panels on all four sides and the room, doubling the capacity of Team Illinois' array. So while Team Germany focused on excess generation, Team Illinois focused on conservation. When looking at Figure 3-17, the differences between the strategies can be seen. During the day, Team Germany did generate more electricity but not much more than Illinois for the differences in the PV array sizes. Also at night Illinois can be seen using less electricity than Team Germany. These two teams highlight the different approaches towards zero energy buildings; however, conversation usually is a much more cost effective way of achieving that goal.

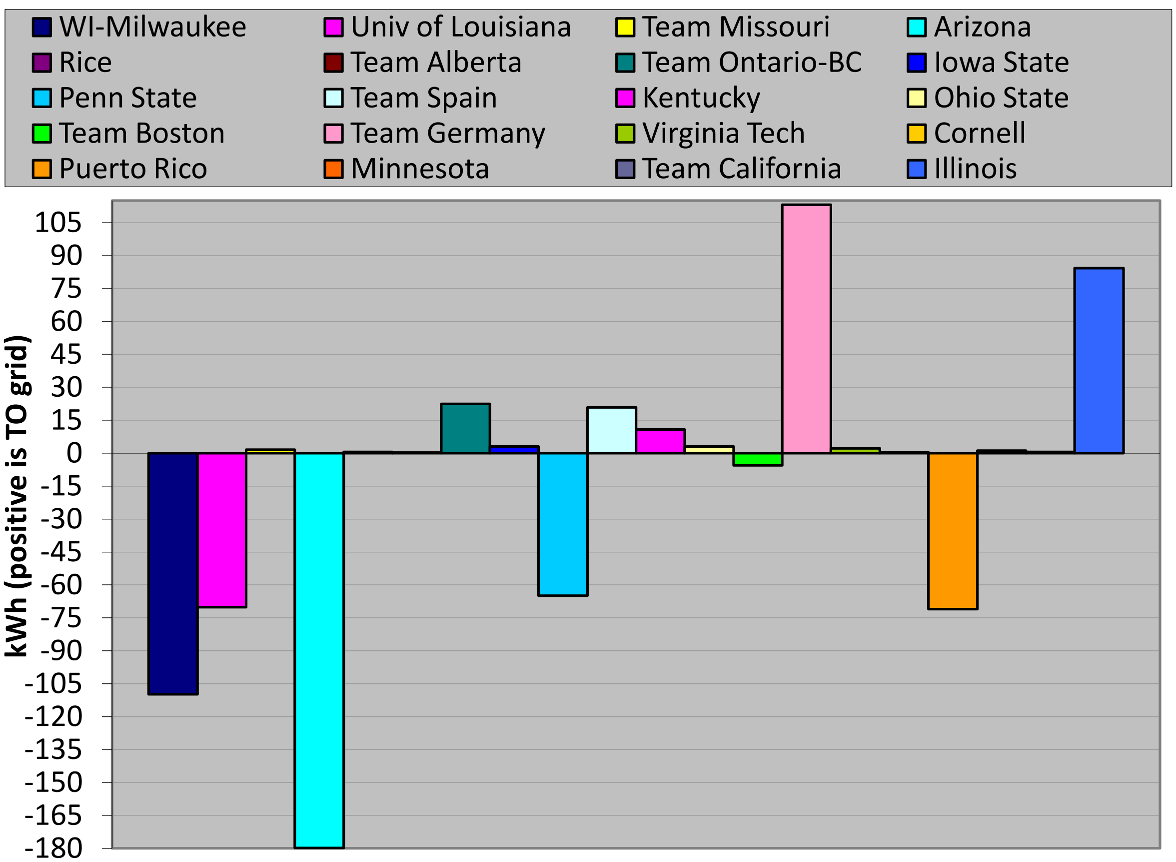


Figure 3-16. Energy balance all teams.

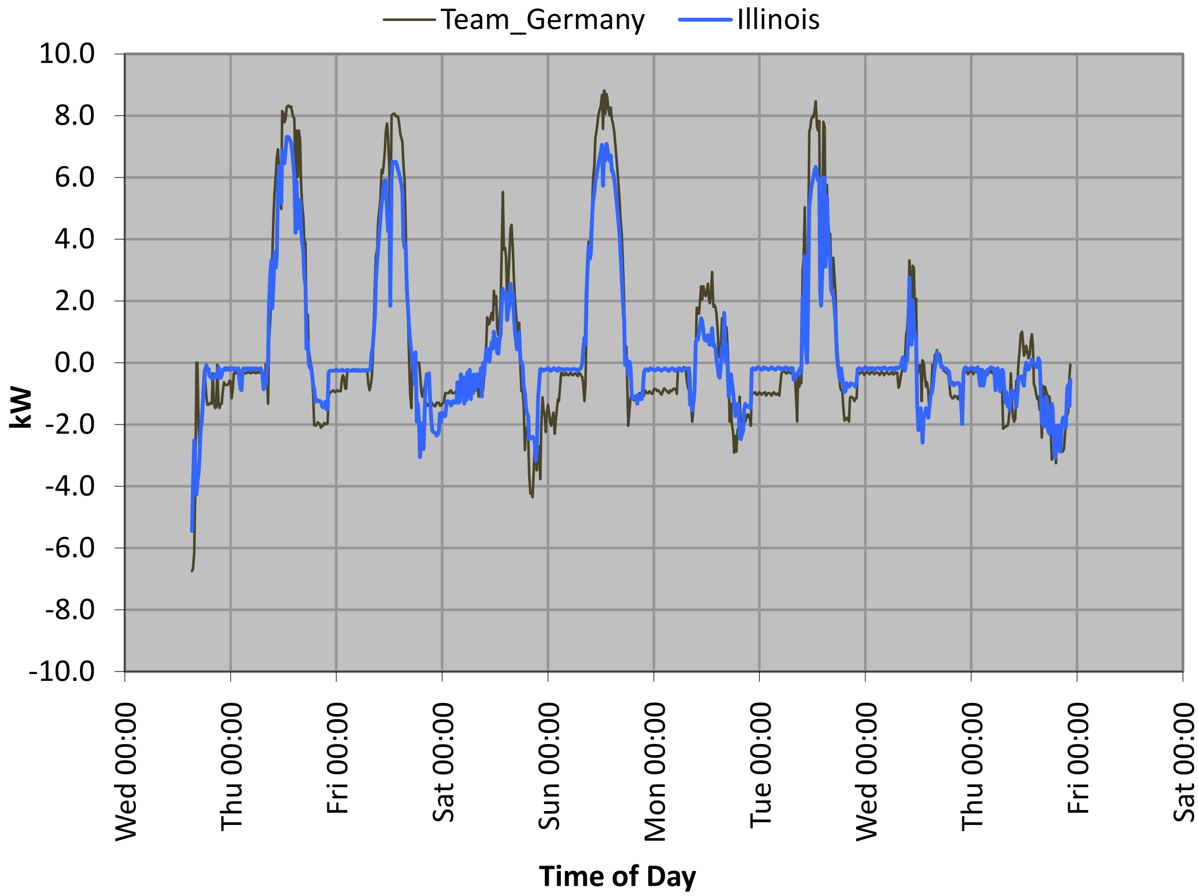


Figure 3-17. Total AC power.

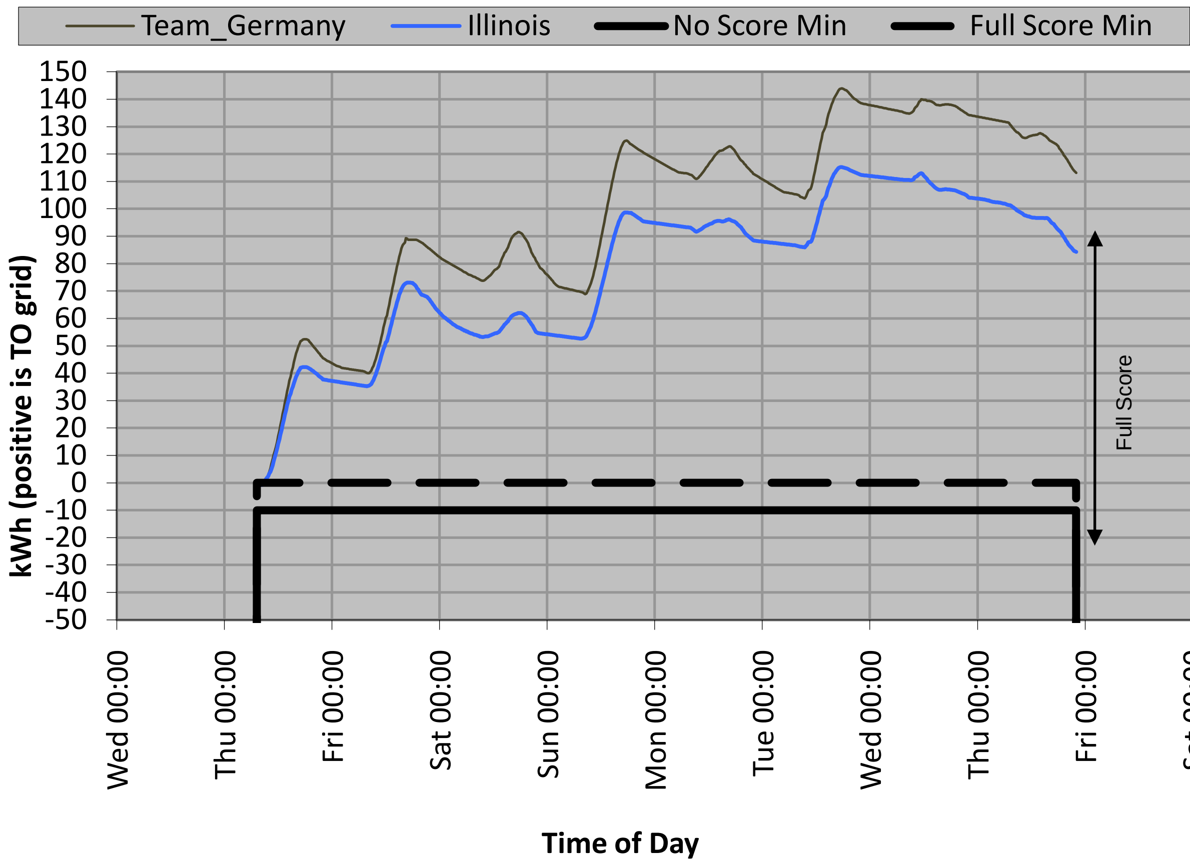


Figure 3-18. Energy balance Illinois/Germany.

3.5 Energy Modeling

3.5.1 Competition Week

As previously mentioned in 2.1.1 Energy Modeling, DesignBuilder was used to model the competition week. All power usages of appliances and internal gains were determined and put into the model. The competition schedule was followed exactly in the energy model so as to closely follow the real competition. In the model, the doors are fully open during the public tour hours and the HVAC system is off. The model differs from the actual results in that in the DesignBuilder model the does not turn off the HVAC at night such as what happened during the competition. In addition, the weather file is not the same as the October 2009 weather in Washington D.C. however; the weather file is close to the actual conditions.

Comparing the total energy consumption simulated in DesignBuilder to the actual energy consumption, DesignBuilder has a weeklong consumption of 112.3 kWh while Team Illinois ended the competition at 84 kWh. However, the value for the actual consumption includes PV generation so the PV generation needs to be added to the DesignBuilder value. Over the course of the week, the Gable Home produced 211.2 kWh bringing the new total to 99 kWh. This value is only 17.4% away from the actual value. This difference can easily be explained through the differences in the weather file and the HVAC system running at night in the DesignBuilder model. There might be other subtle differences but for such a unique operating condition during the competition week, 17.4% is a very good number to reach for simulation.

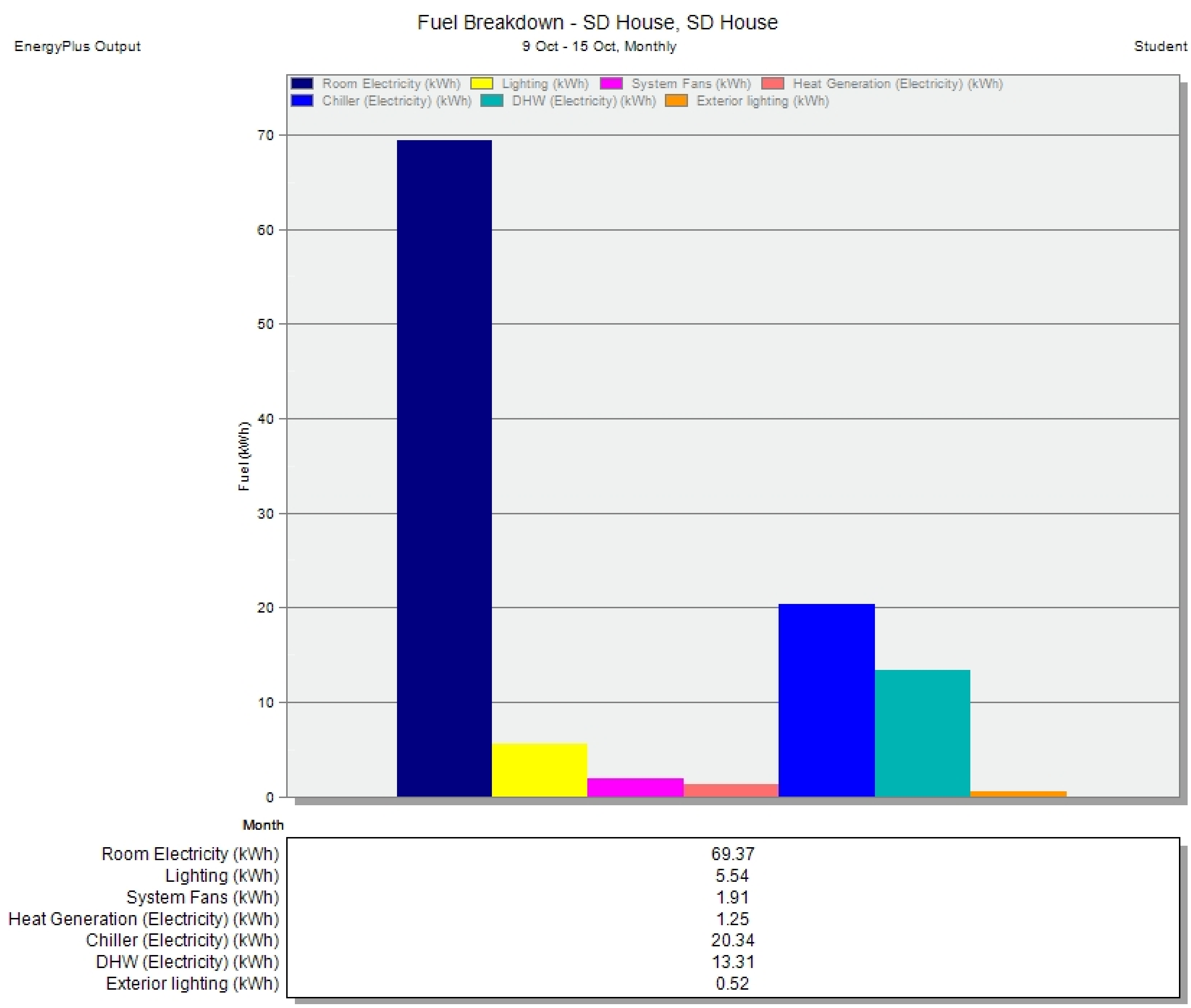
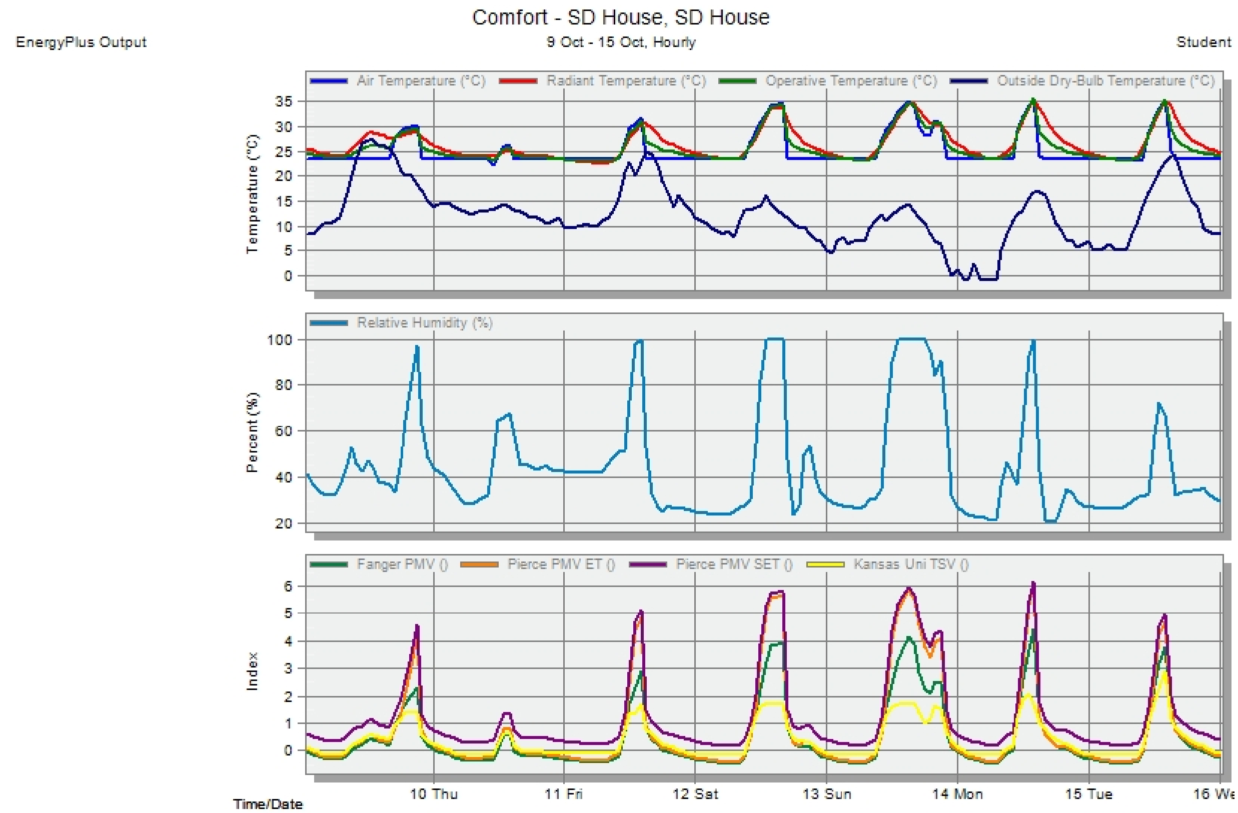
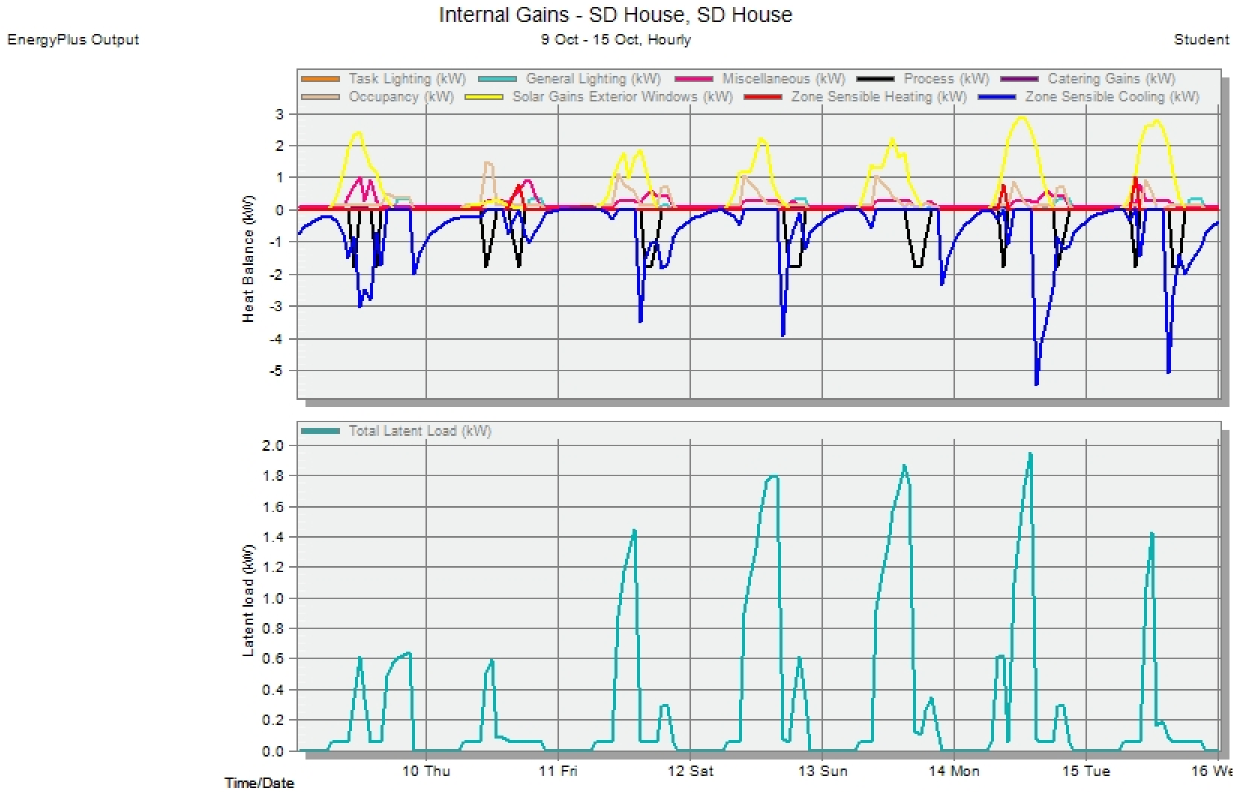


Figure 3-19. Competition week fuel breakdown.



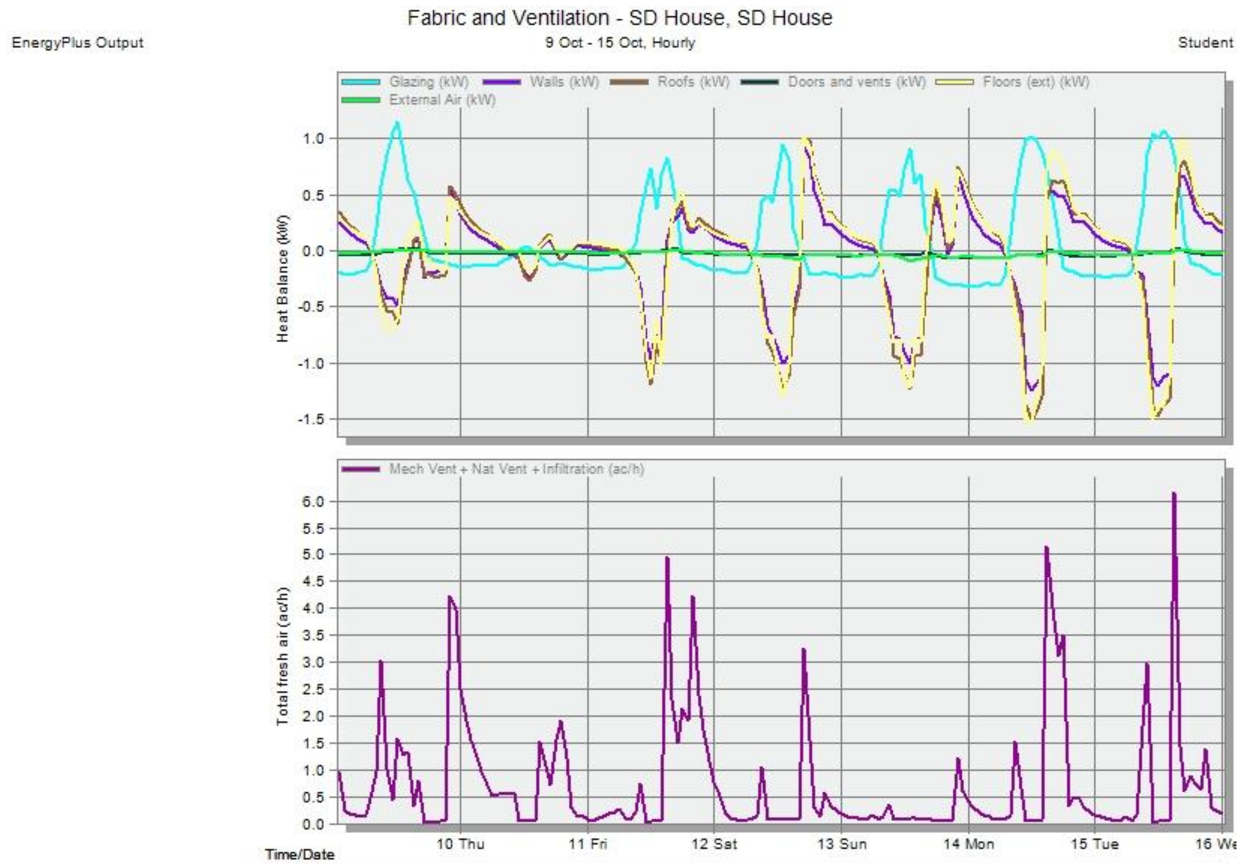
Air Temperature (°C)	23.33	23.33	23.33	29.66	23.33	23.33	23.33	23.33	23.87	23.33
Radiant Temperature (°C)	28.17	24.13	23.96	28.03	23.81	29.13	24.82	26.43	33.77	23.05
Operative Temperature (°C)	25.75	23.73	23.65	28.84	23.57	26.23	24.07	24.88	28.82	23.19
Outside Dry-Bulb Temperature (°C)	26.10	12.80	10.60	22.80	9.40	10.60	11.10	1.10	16.70	5.00
Relative Humidity (%)	37.31	28.38	44.38	73.04	24.14	23.60	30.86	27.20	42.77	26.60
Fanger PMV ()	0.27	-0.34	-0.23	1.72	-0.41	0.29	-0.23	-0.04	1.16	-0.49
Pierce PMV ET ()	0.36	-0.27	-0.14	2.40	-0.35	0.30	-0.15	0.01	1.23	-0.43
Pierce PMV SET ()	0.91	0.31	0.45	2.85	0.23	0.87	0.42	0.59	1.74	0.16
Kansas Uni TSV ()	0.45	-0.13	-0.07	1.35	-0.14	0.48	-0.07	0.14	1.00	-0.13

Figure 3-20. Competition week comfort.



Task Lighting (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
General Lighting (kW)	0.00	0.00	0.31	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Miscellaneous (kW)	0.26	0.09	0.09	0.26	0.09	0.17	0.09	0.09	0.24	0.09
Process (kW)	-1.76	0.00	0.00	0.00	0.00	-1.76	0.00	0.00	0.00	0.00
Catering Gains (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Occupancy (kW)	0.14	0.00	0.14	0.80	0.00	0.14	0.14	0.00	0.13	0.00
Solar Gains Exterior Windows (kW)	1.15	0.00	0.00	1.75	0.00	0.03	1.38	0.00	1.90	0.00
Zone Sensible Heating (kW)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.05
Zone Sensible Cooling (kW)	-0.70	-0.22	-0.36	0.00	-0.12	-1.10	-0.55	-1.06	-5.45	0.00
Total Latent Load (kW)	0.06	0.00	0.06	1.18	0.00	0.06	0.06	0.00	0.07	0.00

Figure 3-21. Competition week internal gains.



Glazing (kW)	0.52	-0.14	-0.13	0.73	-0.19	-0.21	0.53	-0.32	0.72	-0.22
Walls (kW)	0.08	0.07	-0.01	-0.96	0.07	0.83	-0.39	0.39	0.51	0.02
Roofs (kW)	0.09	0.12	0.01	-1.19	0.11	0.99	-0.49	0.52	0.46	0.06
Doors and vents (kW)	0.02	-0.03	-0.03	-0.01	-0.04	-0.04	-0.03	-0.07	-0.00	-0.05
Floors (ext) (kW)	0.22	0.12	0.02	-1.14	0.10	0.94	-0.59	0.49	0.73	0.09
External Air (kW)	0.00	-0.02	-0.02	-0.01	-0.03	-0.04	-0.03	-0.05	-0.01	-0.04
Mech Vent + Nat Vent + Infiltration (ac/h)	0.33	0.52	0.29	0.03	0.09	1.54	0.34	0.41	5.14	0.14

Figure 3-22. Competition week envelope loads.

Chapter 4: ECONOMICS AND COMMERCIALIZATION

4.1 Cost of HVAC System

The cost of the HVAC system totals $7,526.15 from Table 2-4. This is for a fully integrated system that includes heating, ventilation and air conditioning. This also includes labor and ductwork for installation. This cost is also for the system as is in the prototype stage. The costs would come down for the production model and then again as the number of units in production increases.

4.1.1 Cost of Best Practice

Looking at what a comparable standard central air heat pump system would cost, it can be shown how a fully integrated solution is more cost effective. Looking online, a 1.5 Ton SEER 15 central air heat pump costs $2,856 that includes a variable capacity compressor (ACWholesalers, 2010). This does not include labor and other assorted costs such as ductwork. Assuming labor doubles the cost of the HVAC system and $1000 for ductwork, the new total is $6712. While this total is still less than Gable Home's HVAC system, this standard central air system does not include an energy recovery ventilator, which is a large expense. The unit used in the Gable Home cost $1,899, which brings the total to $8,611. Gable Home's HVAC system now costs 12.6% less than the conventional central air system while still in the prototype stage, meaning even greater savings can be realized.

4.2 LEED

4.2.1 Number of Points

Leadership in Energy and Environmental Design, LEED, is a rapidly growing certification for green buildings. Many commercial buildings have become LEED certified and with the new 2009 version of LEED certification, there is a new LEED for Homes. The HVAC system could potential help to earn points in the following areas:

- Energy and Atmosphere
    - Optimize Energy Performance (34 points)
    - Space Heating and Cooling Equipment (4 points)
    - Residential Refrigerant Management (1 point)
- Indoor Environmental Quality
    - Outdoor Air Ventilation (3 points)
    - Local Exhaust (2 points)
    - Distribution of Space Heating and Cooling (3 points)
    - Air Filtering (2 points)

4.2.2 Marketing

Since LEED has a well-known and growing brand name in the green building industry, promoting how the HVAC system can help to earn LEED points is a valuable marketing tool. If a single equipment purchase can affect numerous points and help to get either certification or a higher certification then people would be more willing to purchase the equipment.

4.3 Market Analysis

4.3.1 Passive House Market Penetration

In the United States, Passive Houses do not have much market penetration. Currently there are less than 50 to 100 Passive Houses. However as more people get trained and become Certified Passive House Consultants, more Passive Houses will be built, increasing the demand for a small, efficient HVAC system. Then if you look at how Passive Houses have grown in Germany, the trend looks similar. In the beginning, Passive Houses in Germany were not well accepted and did not have that much market penetration. However, over time people saw the benefit of Passive Houses and began to build more and more of them, driving down the costs to build Passive Houses even more. Finally, certain municipalities and cities have made it code that any new building has to be a certified Passive House. Now there are an estimated 15,000 Passive Houses all over the world according to Rosenthal (New York Times, 2008). This exponential growth can hopefully be the same in the United States and with the exponential growth of certified Passive Houses, there is also an exponential growth in the need of HVAC systems designed for Passive Houses.

4.3.2 Low Energy Home Redesign

The design of the HVAC system using 100% fresh air to condition the home is highly dependent on the home being Passive House certified. If the home is not then the loads will be too large for the supply air to carry the conditioning. However, the design used for the Solar Decathlon competition in the Gable Home could be used for low energy home. These homes have reduced HVAC loads compared to a typical home but are greater than Passive Houses. This is where the extra capacity of the redesigned system can be used. Using the HVAC system in low energy houses greatly expands the market for which it can be marketed towards. Since in the US there are very few Passive Houses, it is a very small market for now.

Chapter 5: CONCLUSIONS

5.1 Indoor Air Quality

There were many differences in data, which had to be accounted for the difference in between typical houses and airtight houses, and the comparison between both types of houses. Some included the different radon concentrations and the lack of ventilation and others included the presence of pets or more residents within the house. The main conclusion the data supported was that the super insulated, airtight houses benefit from better indoor air quality (IAQ). In the airtight house with its ventilator running, it had the lowest carbon dioxide and radon concentrations of all the houses tested. By having continuous ventilation, it was able to have fresh air brought into the house and thus keep the concentrations of the pollutants tested at low concentrations.

5.2 Energy Modeling

Energy modeling is proven to be beneficial to predicting how a technology or building will perform before building it; however, the model is good only if the inputs to the model are good. In the case of the DesignBuilder model, there were good inputs because of the knowledge of the complete week of competition. If this exact schedule and loads in the building were not known then the model would have been off by more than the 17.4% it was off.

5.3 HVAC System

The HVAC system performed very well during the competition. The design decision to use a variable capacity compressor paid off during the competition, saving the team 6.75 kWh. This compressor also proved to scale very well with heating and still scale well for cooling but not as good as heating. The variable capacity compressor also was useful since it could speed up to provide the extra capacity needed to condition the house after public tours during the competition, but then reduce the speed and power consumption after house returned to steady state temperature. The ability of the HVAC system to cool the house by 9 °F in a little over two hours is great for a typical house, but for the competition it was still not good enough.

So for future competitions, an even larger system would be required if a single hour pull down is desired.

5.4 Future Work

Future work that can be done on the HVAC system would be to test and monitor the HVAC system functioning as designed before the change for the competition, using 100% fresh air supply conditioning. If the house operated more like a typical house, the HVAC system should be able to condition the house in all weather but it would have to be tested and monitored to be sure. In addition, continued work towards commercialization could be pursued. This could involve continued redesign to reduce costs or increase performance. Some of these redesigns could be reducing the static pressure of the system, make the condenser larger, and a more tightly integrated package.

The current HVAC system has a very high static pressure due to the arrangement of the heat exchangers and number of duct bends. Since the current heat exchangers are 18 fins per inch and louvered fins, they provide a very large heat transfer surface at the cost of pressure drop. The pressure drop is so large that the inline duct fan blowing across the outdoor coil is actually forcing air back down the exhaust duct and into the ERV. This means all the fans in the system are fighting each other, wasting energy and reducing performance. One way to overcome the large pressure drop is to increase the size of the outdoor coil. By making the coil physically larger, the number of fins per inch can be reduced while maintaining the same heat transfer surface area. To accomplish this goal, the HVAC system would have to be dramatically redesigned since the outdoor coil could no longer be housed in the ductwork. Of course, this would then be a perfect opportunity to make the whole HVAC system much more compact and tightly integrated. A custom box could be used to house the ERV, heat exchangers, compressor and any extra fans needed instead of using separate components connected by ductwork. This would allow for larger heat exchangers with unique shapes as opposed to a small square to fit in a duct. Overall, a complete redesign would have many benefits that would allow for higher performance and lower energy consumption.

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APPENDIX

Appendix A

MASTERFLUX

SIERRA05-0982Y3
Brushless DC Variable Speed Compressor Technical Data Sheet

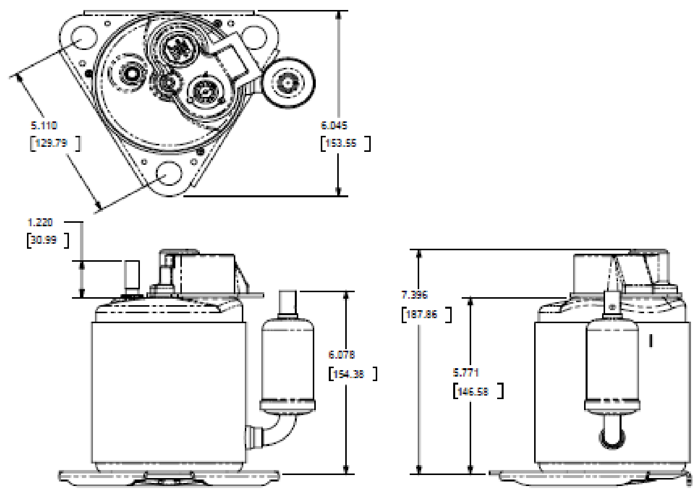
General Information

Compressor Part Number - Accumulator (81.6 cc)	SIERRA00012
Sales Compressor Drawing	DCMX01-004
Sales Controller Drawing	DGMX007
Wiring Diagram	DEMX05

Application Information

Application	HBP, A/C
Refrigerant	R134a
Voltage Range	135-315V DC
Evaporator Temperature Range	-23.3° C to 12.8° C (-10° F to 55° F)
Condenser Temperature Range	26.7° C to 65.6° C (80° F to 150° F)
Maximum Discharge Temperature	130° C (265° F)
Maximum Compression Ratio	8:1

Compressor Dimensions



the Sierra

Issued: 4/29/04, Revision: 1 - Preliminary

www.masterflux.com

Figure A-1. Masterflux variable speed compressor information.

MASTERFLUX

SIERRA05-0982Y3
Brushless DC Variable Speed Compressor Technical Data Sheet

Controller Features

Over/under voltage protection
Compressor thermal monitoring
Controller thermal monitoring
Locked rotor protection

Controller Dimensions

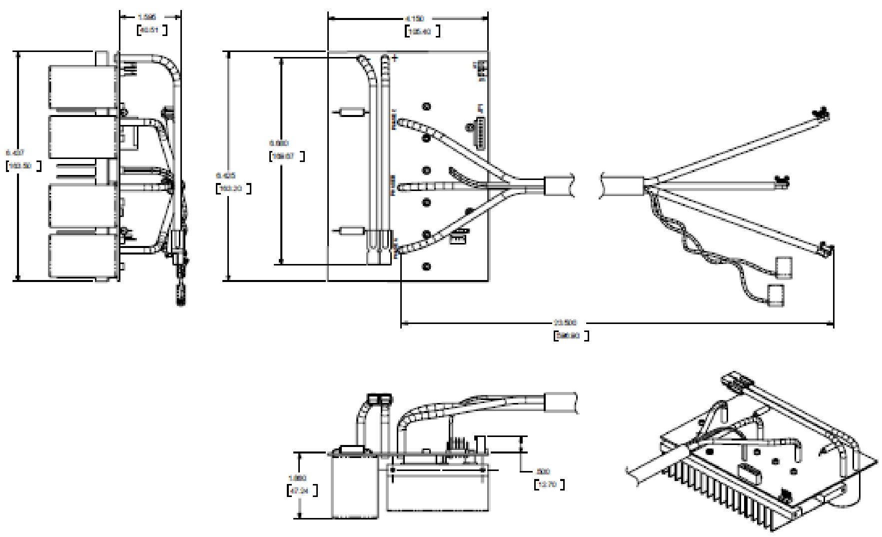


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Figure A-2. Masterflux compressor wiring diagram.

MASTERFLUX COMPRESSOR PERFORMANCE CHART (100V) SIERRA05-0982Y3

Capacity (54.4° C Condenser) watt

	Evaporator Temperature (° C)									
rpm	-12.0	-9.4	-6.7	-3.9	-1.1	1.7	4.4	7.2	10.0	12.8
1800	471	512	567	634	713	805	910	1028	1159	1302
2300	641	699	773	862	967	1088	1224	1376	1544	1727
2800	805	861	940	1041	1164	1309	1477	1668	1881	2116
3500	1047	1139	1257	1401	1571	1767	1990	2238	2513	2813

Power Consumption (54.4° C Condenser) watt

	Evaporator Temperature (° C)									
rpm	-12.0	-9.4	-6.7	-3.9	-1.1	1.7	4.4	7.2	10.0	12.8
1800	360	376	390	402	413	422	430	435	440	442
2300	404	417	430	443	456	469	482	494	507	519
2800	458	475	492	510	529	548	567	587	607	627
3500	599	636	670	702	732	760	786	809	831	850

Current (54.4° C Condenser) amp

	Evaporator Temperature (° C)									
rpm	-12.0	-9.4	-6.7	-3.9	-1.1	1.7	4.4	7.2	10.0	12.8
1800	3.60	3.76	3.90	4.02	4.13	4.22	4.30	4.35	4.40	4.42
2300	4.04	4.17	4.30	4.43	4.56	4.69	4.82	4.94	5.07	5.19
2800	4.58	4.75	4.92	5.10	5.29	5.48	5.67	5.87	6.07	6.27
3500	5.99	6.36	6.70	7.02	7.32	7.60	7.86	8.09	8.31	8.50

COP (54.4° C Condenser) watt/watt

	Evaporator Temperature (° C)									
rpm	-12.0	-9.4	-6.7	-3.9	-1.1	1.7	4.4	7.2	10.0	12.8
1800	1.31	1.36	1.45	1.58	1.73	1.91	2.12	2.36	2.63	2.94
2300	1.59	1.68	1.80	1.95	2.12	2.32	2.54	2.78	3.05	3.33
2800	1.76	1.81	1.91	2.04	2.20	2.39	2.61	2.84	3.10	3.37
3500	1.75	1.79	1.88	1.99	2.14	2.32	2.53	2.76	3.02	3.31

Mass Flow (54.4° C Condenser) kg/hr

	Evaporator Temperature (° C)									
rpm	-12.0	-9.4	-6.7	-3.9	-1.1	1.7	4.4	7.2	10.0	12.8
1800	11.31	12.22	13.38	14.80	16.48	18.41	20.60	23.05	25.76	28.72
2300	15.25	16.58	18.20	20.13	22.36	24.89	27.71	30.84	34.27	38.00
2800	18.60	20.58	22.86	25.44	28.33	31.53	35.02	38.83	42.94	47.35
3500	24.84	27.14	29.91	33.16	36.90	41.12	45.81	50.99	56.65	62.79

* all data points are with 11.1° C superheat and 8.3° C subcooling

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Figure A-3. Masterflux metric low speed compressor data.

MASTERFLUX COMPRESSOR PERFORMANCE CHART (150V) SIERRA05-0982Y3

Capacity (54.4° C Condenser) — watt

	Evaporator Temperature (° C)									
rpm	**-12.0**	**-9.4**	**-6.7**	**-3.9**	**-1.1**	**1.7**	**4.4**	**7.2**	**10.0**	**12.8**
3700	1110	1241	1396	1575	1780	2009	2263	2542	2845	3174
4500	1459	1584	1745	1941	2174	2443	2747	3088	3464	3877
5300	1693	1856	2061	2306	2592	2919	3287	3696	4145	4636
6500	2448	2593	2789	3035	3332	3680	4078	4527	5027	5578

Power Consumption (54.4° C Condenser) — watt

	Evaporator Temperature (° C)									
rpm	**-12.0**	**-9.4**	**-6.7**	**-3.9**	**-1.1**	**1.7**	**4.4**	**7.2**	**10.0**	**12.8**
3700	735	821	891	946	984	1007	1015	1006	982	942
4500	981	1040	1091	1134	1168	1194	1211	1220	1220	1212
5300	1148	1210	1269	1323	1373	1419	1461	1498	1531	1561
6500	1666	1756	1832	1895	1945	1981	2005	2016	2014	1999

Current (54.4° C Condenser) — amp

	Evaporator Temperature (° C)									
rpm	**-12.0**	**-9.4**	**-6.7**	**-3.9**	**-1.1**	**1.7**	**4.4**	**7.2**	**10.0**	**12.8**
3700	4.90	5.47	5.94	6.30	6.56	6.72	6.76	6.71	6.55	6.28
4500	6.54	6.94	7.28	7.56	7.79	7.96	8.07	8.13	8.13	8.08
5300	7.65	8.07	8.46	8.82	9.15	9.46	9.74	9.99	10.21	10.40
6500	11.11	11.70	12.21	12.63	12.96	13.21	13.37	13.44	13.43	13.32

COP (54.4° C Condenser) — watt/watt

	Evaporator Temperature (° C)									
rpm	**-12.0**	**-9.4**	**-6.7**	**-3.9**	**-1.1**	**1.7**	**4.4**	**7.2**	**10.0**	**12.8**
3700	1.51	1.51	1.57	1.67	1.81	1.99	2.23	2.53	2.90	3.37
4500	1.49	1.52	1.60	1.71	1.86	2.05	2.27	2.53	2.84	3.20
5300	1.48	1.53	1.62	1.74	1.89	2.06	2.25	2.47	2.71	2.97
6500	1.47	1.48	1.52	1.60	1.71	1.86	2.03	2.25	2.50	2.79

Mass Flow (54.4° C Condenser) — kg/hr

	Evaporator Temperature (° C)									
rpm	**-12.0**	**-9.4**	**-6.7**	**-3.9**	**-1.1**	**1.7**	**4.4**	**7.2**	**10.0**	**12.8**
3700	26.04	29.04	32.46	36.31	40.58	45.28	50.40	55.95	61.92	68.31
4500	35.53	37.88	40.97	44.81	49.41	54.75	60.85	67.70	75.29	83.64
5300	42.52	45.02	48.47	52.86	58.21	64.50	71.74	79.93	89.07	99.16
6500	56.14	59.25	63.43	68.66	74.96	82.31	90.72	100.20	110.73	122.32

* all data points are with 11.1° C superheat and 8.3° C subcooling

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Figure A-4. Masterflux metric high speed compressor data.

MASTERFLUX COMPRESSOR PERFORMANCE CHART (100V) SIERRA05-0982Y3

Capacity (130° F Condenser) BTU/hr

	Evaporator Temperature (° F)									
rpm	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
1800	1608	1750	1935	2164	2436	2751	3109	3511	3957	4445
2300	2190	2388	2639	2944	3303	3715	4180	4699	5272	5898
2800	2749	2941	3209	3554	3975	4472	5046	5696	6422	7225
3500	3576	3889	4292	4784	5365	6035	6795	7643	8581	9608

Power Consumption (130° F Condenser) watt

	Evaporator Temperature (° F)									
rpm	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
1800	360	376	390	402	413	422	430	435	440	442
2300	404	417	430	443	456	469	482	494	507	519
2800	458	475	492	510	529	548	567	587	607	627
3500	599	636	670	702	732	760	786	809	831	850

Current (130° F Condenser) amp

	Evaporator Temperature (° F)									
rpm	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
1800	3.60	3.76	3.90	4.02	4.13	4.22	4.30	4.35	4.40	4.42
2300	4.04	4.17	4.30	4.43	4.56	4.69	4.82	4.94	5.07	5.19
2800	4.58	4.75	4.92	5.10	5.29	5.48	5.67	5.87	6.07	6.27
3500	5.99	6.36	6.70	7.02	7.32	7.60	7.86	8.09	8.31	8.50

EER (130° F Condenser) BTU/h/watt

	Evaporator Temperature (° F)									
rpm	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
1800	4.46	4.66	4.96	5.38	5.90	6.52	7.24	8.06	9.00	10.05
2300	5.42	5.72	6.13	6.64	7.24	7.92	8.68	9.51	10.41	11.37
2800	6.00	6.19	6.52	6.96	7.52	8.17	8.90	9.71	10.58	11.52
3500	5.97	6.12	6.40	6.81	7.32	7.94	8.64	9.44	10.33	11.31

Mass Flow (130° F Condenser) lb/hr

	Evaporator Temperature (° F)									
rpm	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
1800	24.94	26.93	29.50	32.63	36.32	40.59	45.42	50.82	56.78	63.31
2300	33.62	36.54	40.13	44.38	49.29	54.86	61.10	67.99	75.55	83.77
2800	41.02	45.37	50.39	56.09	62.46	69.50	77.21	85.60	94.66	104.39
3500	54.77	59.82	65.94	73.11	81.35	90.65	101.00	112.41	124.89	138.42

* all data points are with 20° F superheat and 15° F subcooling

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Figure A-5. Masterflux english low speed compressor data.

MASTERFLUX COMPRESSOR PERFORMANCE CHART (150V) SIERRA05-0982Y3

Capacity (130° F Condenser) BTU/hr

	Evaporator Temperature (° F)									
rpm	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**	**35.0**	**40.0**	**45.0**	**50.0**	**55.0**
3700	3792	4237	4766	5380	6078	6861	7729	8681	9717	10839
4500	4983	5409	5958	6630	7424	8342	9382	10545	11831	13239
5300	5781	6339	7037	7875	8852	9969	11225	12621	14157	15832
6500	8362	8856	9524	10365	11379	12566	13927	15461	17168	19049

Power Consumption (130° F Condenser) watt

	Evaporator Temperature (° F)									
rpm	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**	**35.0**	**40.0**	**45.0**	**50.0**	**55.0**
3700	735	821	891	946	984	1007	1015	1006	982	942
4500	981	1040	1091	1134	1168	1194	1211	1220	1220	1212
5300	1148	1210	1269	1323	1373	1419	1461	1498	1531	1561
6500	1666	1756	1832	1895	1945	1981	2005	2016	2014	1999

Current (130° F Condenser) amp

	Evaporator Temperature (° F)									
rpm	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**	**35.0**	**40.0**	**45.0**	**50.0**	**55.0**
3700	4.90	5.47	5.94	6.30	6.56	6.72	6.76	6.71	6.55	6.28
4500	6.54	6.94	7.28	7.56	7.79	7.96	8.07	8.13	8.13	8.08
5300	7.65	8.07	8.46	8.82	9.15	9.46	9.74	9.99	10.21	10.40
6500	11.11	11.70	12.21	12.63	12.96	13.21	13.37	13.44	13.43	13.32

EER (130° F Condenser) BTU/h/watt

	Evaporator Temperature (° F)									
rpm	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**	**35.0**	**40.0**	**45.0**	**50.0**	**55.0**
3700	5.16	5.16	5.35	5.69	6.18	6.81	7.62	8.63	9.89	11.50
4500	5.08	5.20	5.46	5.85	6.36	6.99	7.75	8.64	9.70	10.92
5300	5.04	5.24	5.55	5.95	6.45	7.02	7.68	8.42	9.24	10.15
6500	5.02	5.04	5.20	5.47	5.85	6.34	6.94	7.67	8.52	9.53

Mass Flow (130° F Condenser) lb/hr

	Evaporator Temperature (° F)									
rpm	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**	**35.0**	**40.0**	**45.0**	**50.0**	**55.0**
3700	57.41	64.02	71.57	80.05	89.47	99.82	111.11	123.34	136.51	150.60
4500	78.34	83.50	90.32	98.80	108.93	120.71	134.15	149.24	165.99	184.40
5300	93.73	99.25	106.85	116.54	128.33	142.20	158.17	176.22	196.37	218.60
6500	123.77	130.63	139.84	151.37	165.25	181.46	200.01	220.89	244.11	269.67

* all data points are with 20° F superheat and 15° F subcooling

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Figure A-6. Masterflux english high speed compressor data.