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Stephen L. Caskey
stephen.l.caskey@gmail.com

Derek Kultgen

Eckhard A. Groll

William Hutzel

Tobias Menzi

Field Test Simulation of an Air-Source Heat Pump with Two-Stage Compression and Economizing For Cold Climates

Stephen CASKEY[1]*, Shuai DENG[1], Derek KULTGEN[2], Tobias MENZI[3], Eckhard GROLL[1] PhD, William HUTZEL[2] PE, Stefan BERTSCH[3] PhD

[1]Herrick Laboratories, Mechanical Engineering, Purdue University,
West Lafayette, Indiana, USA
scaskey@purdue.edu, deng23@purdue.edu, groll@purdue.edu

[2]Applied Energy Laboratory, Mechanical Engineering Technology, Purdue University
West Lafayette, Indiana, USA
dkultgen@purdue.edu, hutzelw@purdue.edu

[3]Interstate University of Applied Science of Technology NTB,
Buchs, Switzerland
tobias.menzi@ntb.ch, stefan.bertsch@ntb.ch

*Corresponding author

ABSTRACT

A new air-source heat pump technology optimized for cold climates was designed and fabricated by the authors in close cooperation with three industrial partners. The constructed unit will undergo a field demonstration in a military barrack to identify heat pumps as cost effective systems that have less primary energy consumption when compared to traditional cold climate heating methods. A simulation model developed in EES predicted the designed heat pump performance at different ambient conditions. The EES results were incorporated with a TRNSYS model to couple the military barrack building load with the available heat pump capacity using weather data. The TRNSYS model enables the assessment of the field demonstration performance during the heating season.

The heat pump design is based on two-stage compression with economizing. Commercially available components were selected for all parts of the heat pump. A variable-speed scroll compressor is used as the high-stage compressor matched with a tandem fixed-speed scroll compressor used as the low-stage compressor. The configuration has a predicted capacity of 18.34 kW (62,580 BTU/h) at the design ambient temperature of -20^{o}C (4^{o}F) based on the EES simulation results. The building has a heating load of less than 18 kW for more than 95% of the heating season that lasts 8 months out of the year. The heat pump design therefore is predicted to satisfy the building heating load for the entire heating season. The heating season COP based on TRNSYS hourly simulation results is 3.67 with a yearly heating capacity of 30,970 kWh (105,674 kBTU) and 8,438.37 kWh (28,793 kBTU). The CCHP simulations predict over 30% savings in primary energy and CO_2 emissions with a 25% cost savings for annual heating energy use compared to an 85% AFUE natural gas furnace.

1. INTRODUCTION

1.1 Air-Source Heat Pump Motivation

Buildings consume 40% of the primary energy used in the U.S. (DOE Buildings Energy Data Book) and generate 40% of the U.S. greenhouse gas emissions (Roth, Westphalen, Dieckmann, Hamilton, & Goetzler, 2002). In colder climates with longer heating seasons, heating for buildings is by far the biggest consumer of total energy consumption, accounting for as much as 60% of the energy used. The heating energy requirement can be reduced significantly by improving building efficiency through the use of technologies that can achieve higher heating performances. Ground-source and air-source heat pumps are two types of HVAC technologies available to residential and small commercial buildings capable of offering increased efficiencies. However, a large disadvantage of available air-source heat pumps is how their performances decrease significantly when the ambient temperatures drop below 25 ^{o}F (-4 ^{o}C), whic typically limits their applications to warmer climates. Ground-source heat pumps avoid problematic ambient air temperatures to maintain high efficiencies, but have extensive installations, which amplify the initial costs. Implementing an air-source heat pump optimized for cold climates can abate both energy consumption and produced emissions due to building HVAC systems by replacing existing combustion based heating systems. Moreover, the higher installation costs and related maintenance expenses of ground-source heat pumps can be avoided.

1.2 Department of Defense Relevance

The development and deployment of a cold-climate heat pump supports a number of mandates to improve the efficiency of federal buildings, including buildings operated by the Department of Defense. One directive is Executive Order 13514 "Federal Leadership in Environmental, Energy, and Economic Performance" that was signed by President Obama in October of 2009. The latest of several Executive Orders, it includes mandatory energy reductions for federal buildings and the overarching goal of achieving net-zero energy buildings by 2030. The Energy Independence and Security Act of 2007, Section 315, specifically discusses "Improved Energy Efficiency for Appliances and Buildings in Cold Climates". Calling for the improved efficiency of mechanical systems as well as an increase use of renewable resources, current heating technologies employed in cold-climates are challenged if to operate from renewable resources. DoD has also developed its' own Energy Security Initiatives with one component of the strategic plan to create more efficient facilities (Energy Security Initiatives, 2010). A mandate for DoD installations requires reductions in energy consumption by 3% per year through 2015. A fully commercialized and widely deployed technology similar to an air-source heat pump optimized for cold-climates would help to achieve these goals.

1.3 Selected Heat Pump Configuration

The heat pump investigated and simulated here is an air-source heat pump using R-410A as the working fluid where two compressors can operate in series resulting in two, separate stages of compression. The discharge port of the low-side (low-pressure) compressor is piped into the suction port of the high-side (high-pressure) compressor with a mixing chamber in between. Attached to the mixing chamber between the compressors is the economizer line

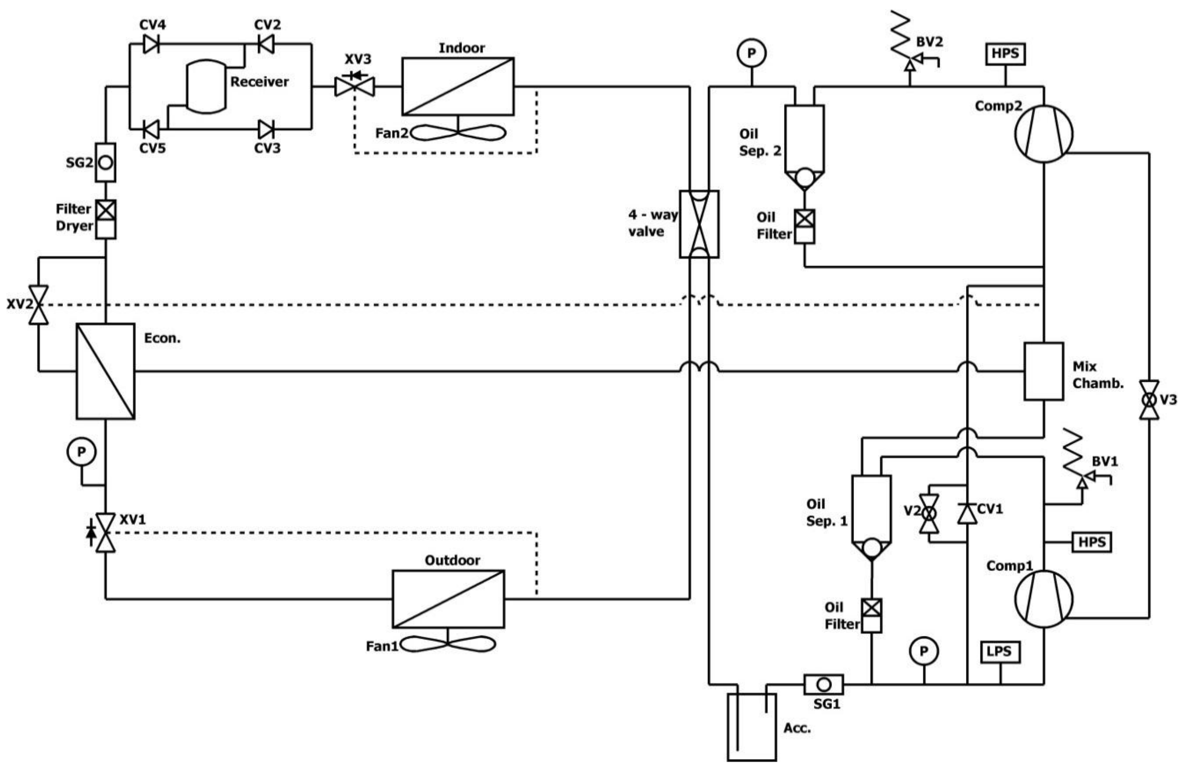

Figure 1: Piping schematic of 2-stage air-source heat pump with economizing and low-side compressor bypass

bringing a two-phase, high-quality refrigerant mixture from the liquid side of the cycle to mix with the hot discharge gases of the low-side compressor. Figure 1 provides a more detailed depiction of the economizer location within the cycle. The purpose of the two-phase refrigerant mixing with the hot discharge gas of the first-stage compressor is the resulting cooling effect of the low-side discharge gases, which in turn allows for a cooler high-side compressor discharge gas. The lower suction temperature of the high-side compressor increases the isentropic efficiency and allows for higher pressures ratios to be obtainable. In addition, a second benefit occurs on the liquid side of the economizer. To evaporate the economized refrigerant to a two-phase mixture of sufficient quality, a temperature difference needs to exist between the separated refrigerant streams. By introducing an expansion valve, the economized refrigerant stream is expanded from the condenser outlet to an intermediate pressure. Based on the expansion, the economized refrigerant becomes a cooler low quality two-phase mixture. The remaining refrigerant stream at the condenser outlet is maintained at the higher pressure, and thus higher temperature. As these two separate refrigerant streams pass through the economizer, additional subcooling occurs to the high pressure refrigerant stream while the economized refrigerant becomes a high quality two-phase stream that mixes with the discharge gases of the low-stage compressor. The additional subcooling of the high-pressure refrigerant stream allows for a higher system capacity through an increase in evaporation enthalpy difference, which has a larger impact on the system capacity than the reduction in refrigerant mass flow rate through the evaporator. With the added benefits of increased system capacity and a higher system coefficient of performance, COP, two-stage compression results in some difficulties that need to be addressed. In particular, an oil management system that allows single-stage and two-stage compressor operation needs to be carefully evaluated. In the current system, two oil separators are located in the discharge lines for each compressor, as shown in Figure 1. Additionally, a dedicated line for oil equalization between the compressors, including a shut-off valve 3, V3, which closes during two-stage operation, was introduced. The importance of the oil management system is to maintain the longevity of each compressor.

1.4 Compressor Selected

A design heating load of 19 kW (64.8 kBTU/hr) at an ambient temperatures of -20^{o}C (-4^{o}F) was used to select the compressors. Engineering Equation Solver, EES, (Kline) was used to simulate different compressor configurations with either digital or variable speed scroll compressors as the high-side compressor, and different sized fixed-speed scrolls as the low side compressor. During the analysis, it was discovered that the digital high-side scroll was not compatible with an expansion valve that was selected for the system. While high-side digital scroll compressors were found in different configurations to achieve the desired heating output, in the end, only a variable speed scroll compressor could be used due to the expansion valve requirement. The possible speed ranges are from 1800 RPM to 7000 RPM with compressor maps available at speeds of 1800, 2700, 3535, 4500, and 7000 RPM. The compressor maps were obtained from the manufacturer and are based on the standard 3rd order polynomial approach with cross-terms according to ANSI/ARI Standard 540-1999 (Positive displacement refrigerant compressors and compressor units, 1999). Two different sized variable speed compressors were simulated with different fixed-speed compressors. The heating output was best matched with the large displacement variable speed compressor and a tandem, fixed-speed compressor with a total displacement of 102.3 cm^3/rev (6.24 in^3/rev). No single, fixed-speed, compressor using the available single-phase power could produce the necessary mass flow rate as the low-stage compressor. Thus, after consulting with the manufacturer, tandem compressors were selected.

2. SYSTEM DESIGN PROCEDURE

2.1 EES Results Integrated into TRNSYS

The complete heat pump was simulated with EES to determine the capacity available at a range of ambient conditions and compressor speeds. Five speeds were simulated when the variable speed compressor operates alone, and three speeds were simulated when both compressors operate in series. When compressing in series, the simulation can be reduced to using only two speeds due to the small variance on capacity between the compressor speeds. From the two possible compressor configurations and different speeds, seven conditions give seven capacity curves that are a function of ambient temperature. Also, the COP was obtained following the same principle and thus, is a function of the speed and ambient temperature. The capacity and COP are needed as a function of ambient temperature to integrate the EES simulation results into the TRNSYS model. Figure 2 presents a flow chart identifying how the EES and TRNSYS simulations are utilized to predict the performance of the field demonstration.

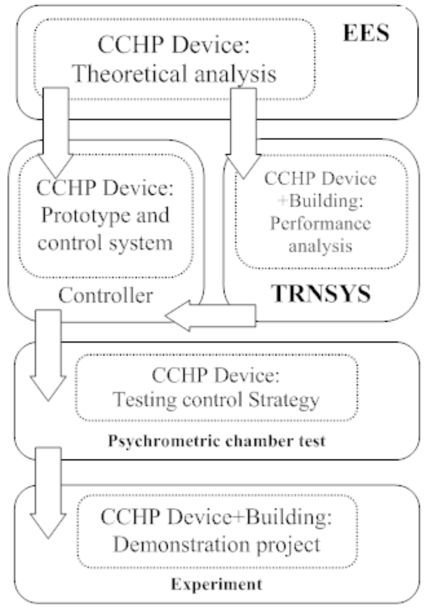

Figure 2: How simulation results are utilized for the field demonstration

The capacity as a function of ambient temperature and compressor speed allows TRNSYS to obtain the amount of heating delivered to the conditioned space. With the various available speeds of the high-side compressor, two capacities are possible at one ambient condition. To solve this conflict, ranges of ambient temperatures are defined for different compressor speeds. This allows only one possible heating capacity for the heat pump from the ambient temperature. The ranges are established by first assuming a linear heating load for the building between the design heating load and no heating load at 20 °C (68 °F). The available heat pump capacity is compared to the linear load at the different compressor speeds. The locations where the heat pump capacity was above or within roughly 1 kW (3.4 kBTU/hr) were selected as the ambient temperatures for a compressor speed. The temperature ranges for the different compressor speeds can be seen in Table 1, where the open and closed brackets note which compressor speed is used at the boundaries of the temperature ranges. The same temperature ranges are used to select the compressor speed when referencing the heat pump COP from the EES results.

Table 1: Ambient temperature ranges used to determine heat pump capacity from EES

Compressor Configuration		**Single Stage**					**Two-Stage**
High-Side Comp. Speed	RPM	*1800*	*2700*	*3535*	*4500*	*7000*	*3535-4500*
Temperature Range	°C	[20,3]	(3,0]	(0,-4]	(-4,-11]	(-11,-16]	(-16,-20]
	°F	[68,37]	(37,32]	(32,25]	(25,12]	(12,3]	(3,-4]

2.2 Military Barracks

The military barrack for the field demonstration was selected from available buildings located at Camp Atterbury, outside Edinburgh, Indiana. The building selected has two identical sleeping areas that are connected by a shared lavatory and shower room. One important aspect of this configuration is the barrack has two separate HVAC systems that supply each sleeping area independently. When the heat pump is installed, one current system will be bypassed and the heat pump will provide all heating and cooling to one half of the building. A layout of the building with a picture of the exterior is shown in Figure 3. The area for one half of the building is 244m^2 (2,626 ft^2). The walls of the barracks are made of cinder blocks. All windows are single pane glass. The supply and return ducts are insulated within the unconditioned attic space. The foundation is a concrete slab, which is covered with stone tile. The currently installed HVAC system is a natural gas furnace with a split system air conditioner that is assumed to be a couple years old. All HVAC equipment is housed within a separate mechanical room that is unconditioned. The heating set point for the barracks is 20 °C (68 °F) and the cooling set point is 23.3 °C (74 °F).

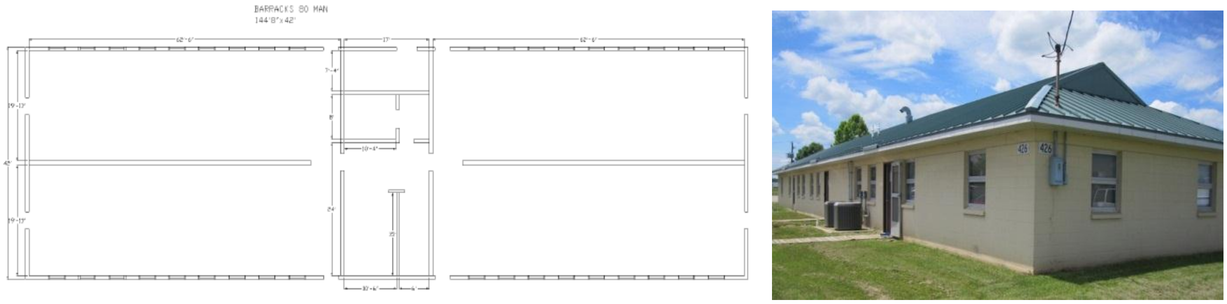

Figure 3: Schematic and a photograph of the selected barrack at Camp Atterbury, Indiana

2.3 Heat Pump & Furnace Primary Energy, Cost and Emissions

The primary energy consumed for the production of electricity used by the heat pump for the entire heating system will be compared to the primary energy consumption of a natural gas furnace. The gas furnace has an AFUE rating of 85% and is assumed to have a blower electricity consumption of 200 kWh per year (Lutz, Franco, Lekov, & Wong-Parodi, 2006). The heating load of the building for one year is used to obtain the amount of primary energy used by the furnace in the form of natural gas. The heat pump is assumed to have all electricity consumed to be generated by natural gas at the rate of 2.627 kWh per kWh of generated electricity (Deru & Torcellini, 2007). Transmissions losses are modeled with an efficiency of 90%. The associated CO_2 emissions from the burning of natural gas from the primary energy consumption are compared. The EPA rates the amount of CO_2 emitted from natural gas as 1,920,000 kg/10^6m^3 (120,000 lb/10^6 ft^3) where the volume is of natural gas fired (AP 42, Fifth Edition, Compliation of Air Pollutant Emission Factors, Vol. 1, Ch 1.4, 1998). A cost analysis of both systems is performed with an electricity rate of \$0.1151/kWh and a natural gas cost of \$10.42/1000 ft^3 (Average Annual Price, 2011). The energy costs are assumed to be constant throughout the year.

3. MODELING

3.1 Overall TRNSYS Model

The two main components within the TRNSYS model are the cold climate heat pump and the military barrack building. Additional components, also known as types within TRNSYS, are used for the supply and return ducts, indoor blower, and HVAC controller. The different connection made between each type is shown in Figure 4. The weather data TRNSYS uses is TMY2 referencing the location Monroe County, Indiana. The entire calendar year is used when running the simulation, but only the results for the heating season are investigated. The cooling season is not considered within the analysis of this paper since the heat pump is assumed to achieve a similar performance

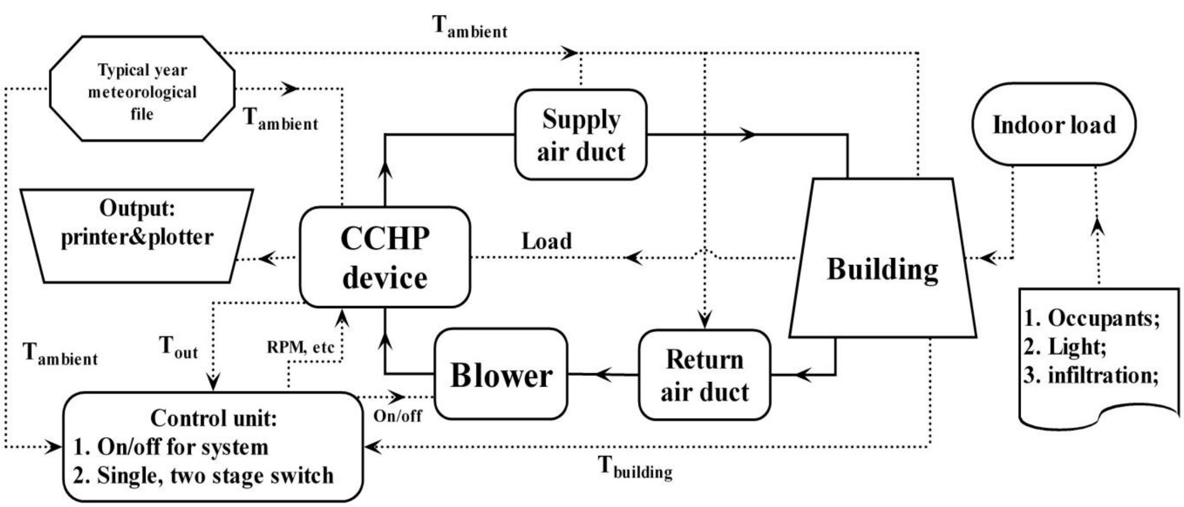

Figure 4: TRNSYS model layout

when compared to the currently installed split system air conditioner. The TRNSYS simulation is run on an hourly basis and all calculated values at the hour are held constant until the next hour is calculated. The monthly electricity consumption and COP, coefficient of performance, are calculated using an hourly bin method where all hourly values are integrated over each month. Equation 1 shows the integration where the known hourly heat pump capacity, $\dot{Q}_{out}$, and COP_{hour} are used.

$$COP_{month} = \frac{\int_0^{h_f} \dot{Q}_{out}\, dh}{\int_0^{h_f} \dot{W}_{elec}\, dh} = \frac{\int_0^{h_f} \dot{Q}_{out}\, dh}{\int_0^{h_f} \frac{\dot{Q}_{out}}{COP_{hour}}\, dh} \tag{1}$$

The integration bounds are between zero for the first hour of the each month and h_f for the hour corresponding to the last hour for the month. The hourly electric consumption, $\dot{W}_{elec}$, of the heat pump is substituted using the heating COP definition.

3.2 Input Parameters for TRNSYS Types

The occupants' load, lighting energy and the infiltration rate are modeled at hourly values that either vary over a 24 hour period or are constant for the 24 hours. Four types of days are considered giving four separate hourly variations in a 24 hour period; typical workday, a Saturday, a Sunday, and a holiday. The number of occupants for the barracks is 40. All occupants are assumed to be seated at rest resulting in a heat gain per person of 60 W (205 BTU/hr). The hourly percentage of the number of occupants for the four different 24 hour days is plotted versus time in Figure 5. The occupant percentage is drastically reduced after 6:00 am for most of the days due to the soldiers leaving the barracks in the morning. The percentage increases back to a base level by 8:00 to 9:00 pm as the soldiers return for the night. 16 fluorescents lights result in 2.15 W/m^2 (0.68243 BTU/hr-ft^2) with a constant lighting fraction of 0.002

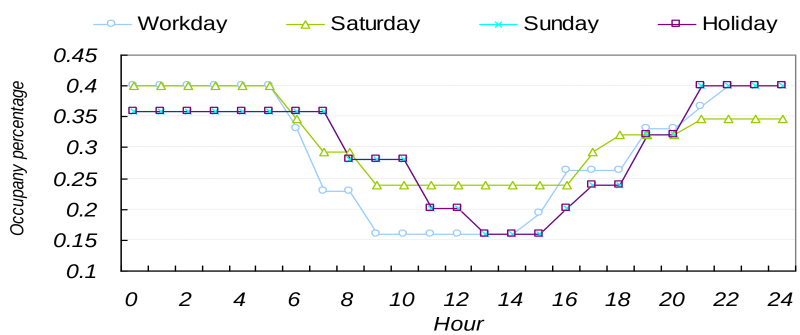

Figure 5: Percent of the total number of occupants for 4 different types of days

over a 24 hour period. The total amount of infiltration is 0.001229 m^3/s-m^2 (0.0242 ft^3/min-ft^2). Similar to the occupant percentage, the amount of infiltration will vary by a percentage of the total over a 24 hour period and is dependent on the type of day. The increased amount of activity in the barracks results in higher infiltration percentages when soldiers are opening the exterior doors to leave in the morning and return in the evening. The trends are shown with the plot in Figure 6.

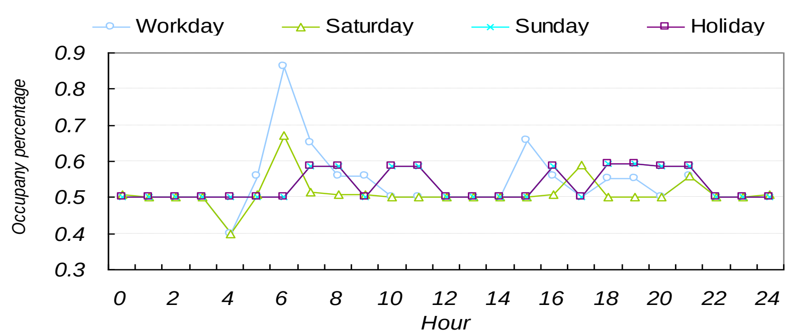

Figure 6: Percent of total infiltration for 4 different types of days

The blower type is a fixed speed blower with a fixed maximum flow capacity of 3500 m^3/hr (2060 ft^3/min). The power consumption is calculated as a linear function of mass flow rate. Due to the heat addition from the blower to the incoming air stream, the air outlet temperature is slightly higher than the inlet air temperature. The type allowed for an input parameter to be defined, a conversion coefficient, as the ratio of released thermal energy to the consumed blower power. The default value of 0.10 was kept.

3.3 Cold Climate Heat Pump Device

With separate data obtained from the EES simulation results for the heat pump, a custom module was created within the TRNSYS model to reference the available heating capacities and COPs at the different ambient temperatures and compressor speeds. The CCHP device capacity output and performance is decided by the ambient temperature and the respective temperature range it falls into to select a speed and compression mode. The implementation of the CCHP device within overall TRNSYS model is shown in Figure 4. The solid lines identify the air flow path and the dotted lines are for passing variables between the different types. The control unit is responsible for monitoring the building temperature, $T_{building}$, and the ambient temperature, $T_{ambient}$. If the building temperature drops below the heating set point or the ambient temperature is below 20 °C (68 °F) the CCHP system is turned on by the controller.

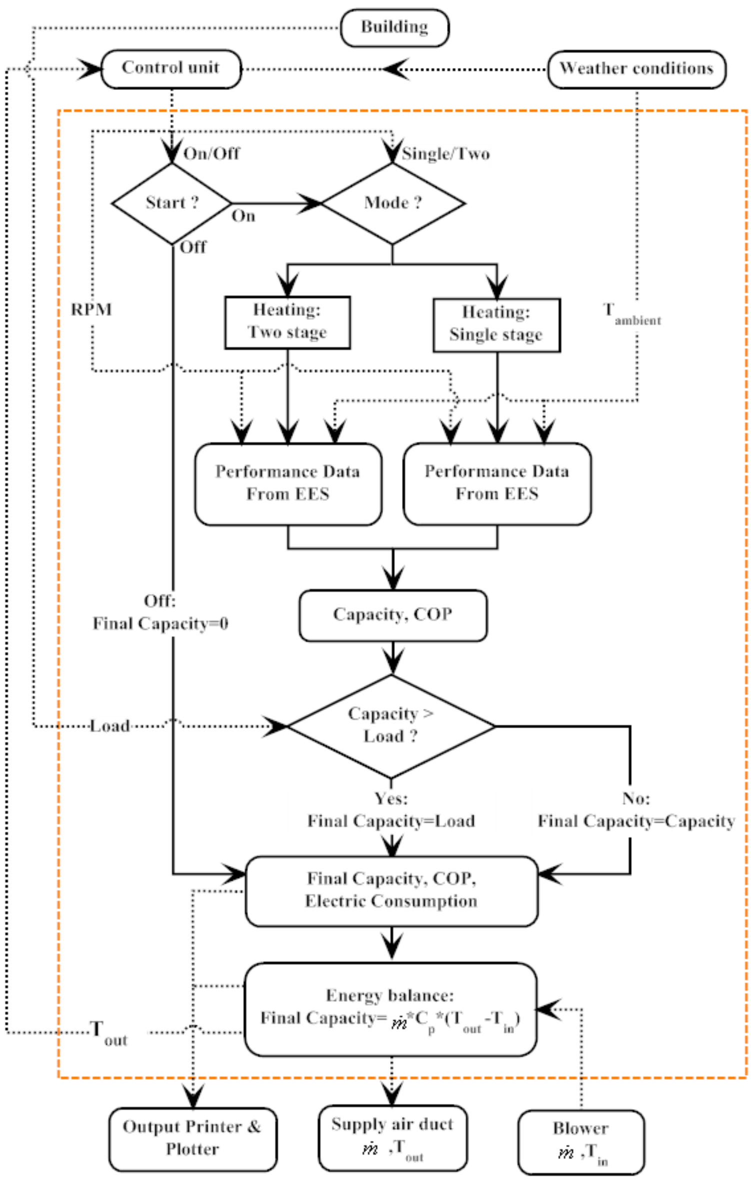

Figure 7: Cold climate heat pump logic for the module developed within TRNSYS

The mode is set to single-stage if the ambient temperature is not below -16 ^oC (3 ^oF). The last decision by the controller is selecting the speed of the CCHP compressor. The temperature ranges shown in Table 1 are used to assign the RPM value. Both the RPM and ambient temperature are necessary to obtain a heating capacity and COP. The resulting capacity is compared to the calculated heating load for the building at the ambient condition. If the heating load is greater, the final output capacity is the capacity from the EES performance data. If the EES heating capacity is greater than the load, the final capacity is altered to match the building heating load to maintain a stable building temperature. An air-side energy balance is completed to find the air outlet temperature, T_{out}. The final capacity, specific heat of air, c_p, and values from the blower, the incoming temperature, T_{in}, and air mass flow rate, $\dot{m}$ are used. The air outlet temperature is read by the controller to prevent any extreme values. The supply air duct is sent both the air outlet temperature and air mass flow rate. A detailed depiction of the logic inside the CCHP device is shown in Figure 7 with the device boundaries shown as a dotted line box. If the final capacity is not sufficient to satisfy the building load, an auxiliary heater of 500 W (1.7 kBTU/hr) is turned on to supplement the CCHP output.

4. SIMULATION RESULTS

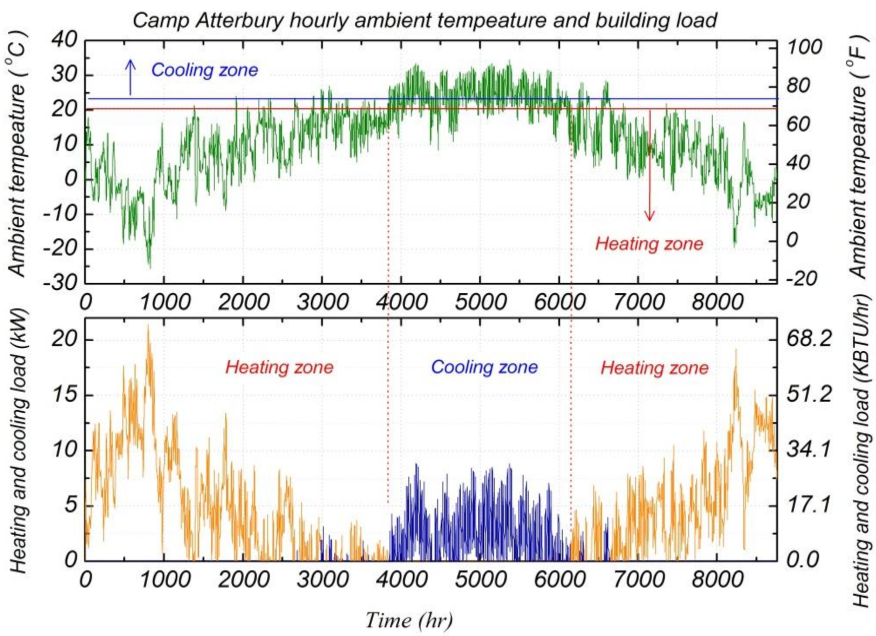

Figure 8: Ambient temperature and cooling or heating load at the military barracks

4.1 Military Barracks Heating and Cooling Loads

After running the TRNSYS simulation for an entire calendar year the ambient temperatures with the heating and cooling set points define the heating and cooling seasons. The heating season starts in the beginning of September and ends in April. For the majority of the heating season, the building load remains below the heat pump design point of 19 kW (64.8 kBTU/hr) at an ambient temperature of -20 ^oC (-4 ^oF). Figure 8 presents both the ambient temperature and the cooling or heating loads as a function of hours in a year.

4.2 Cold Climate Heat Pump Performance

As the ambient temperatures decreases during the heating season, the efficiency of the heat pump begins to decrease. Coupled with the decreasing temperatures, more capacity is needed to satisfy the building heating load while greater pressure ratios are encountered. The compressor speed is increased to reach higher capacities but this occurs with the cost of increased electricity consumption. At a certain point, the capacity is no longer sufficient and the heat pump enters two-staged compression. The COP is able to maintain a desirable value of approximately 2.4 during this transition with the aid of economizing. During the coldest part of the heating season, the COP is at a value of 3.5 or higher. Larger COP values close to 4 are reached with the warmer ambient temperatures during the beginning and end of the heating season. The heating season COP from hourly simulation results is 3.67 with a yearly heating

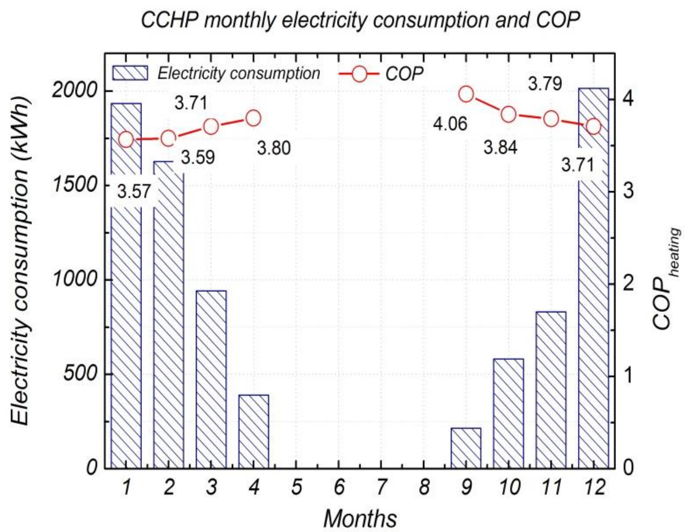

Figure 9: Monthly electric consumption and COP for the heating season

capacity of 30970 kWh (105,674 kBTU) and 8438.37 kWh (28,793 kBTU). The plot of the monthly electric consumption and COP is shown in Figure 9.

One interesting observation of the results is the fact that the monthly COP for January and February did not change much. The amount of electricity consumed between the two months different by almost 300 kWh (1.023 kBTU). The monthly COP did not increase by much even though the amount of electricity consumed decreased. When looking at Figure 8, the ambient temperatures reach extreme values below -20^{o}C (-4^{o}F). In this case, the CCHP needs to use the auxiliary electric heater. The COP for the overall system reaches low values close to 1.65 at these conditions. The monthly average COP is impacted by the low values resulting in little change from January to February.

The accuracy of the CCHP device used in TRNSYS is analyzed by looking at the conditions when the selected RPM gives a heating output larger than the building load. At these occurrences, the difference is calculated between the building load and the heating output. For the entire heating season the total difference is 17,721 kWh (60,466 kBTU/hr) which is 57% of the total heating load for the year. If the CCHP device did not change the capacity output to the building load energy rate, the actual heating output would become 48,629 kWh (165,929 kBTU) giving a larger amount of electricity used.

4.3 Percent Savings

After obtaining the heating output and electricity consumption of the CCHP for the entire heating season, the primary energy, cost and CO_2 emissions were calculated. The simulation results present an attractive percent savings for the CCHP compared to a natural gas furnace. Over 30% primary energy savings and CO_2 emissions and a 25% cost savings are predicted s shown in Table 2.

Table 2: Percent savings when comparing the CCHP to a natural gas furnace

	COP	Thermal energy demand for heating (kWh)	Electricity consumption for heating (kWh)	Thermal energy supply for heating (kWh)	Natural gas supply for heating (kWh)	Cost ($)	CO_2 emission (ton)
CCHP	3.67	30970	8438	--	24630	971	4.95
Furnace	0.85	30970	200	36435	37019	1295	7.44
Percent Savings	--	--	--	--	33.5%	25.0%	33.5%

5. CONCLUSIONS

The location of the military barracks is confirmed to be in a heating dominated climate based on the simulation results. Also, the predicted COPs show that the heat pump performs well in cold climates with maintaining values above 3.5 over the entire heating season. The available capacity of the heat pump is shown to provide for at least 95% of the expected heating requirements. The observed heating loads are for the majority, below the design point of the heat pump capacity. The results demonstrate the feasibility of deploying air-source heat pumps with two-stage compression and economizing in cold climates. Additionally, the percent savings of a cold climate heat pump versus a natural gas furnace are anticipated to be over 30% savings for primary energy consumption and CO_2 emissions with a 25% cost savings on yearly energy costs.

Two recommendations are given to enhance the accuracy of the simulation. One would be to consider a linear interpolation on compressor speeds to obtain heating capacity and COP values outside the results obtained from EES. Also, the percent savings calculation can be improved by considering the production of electricity from a mix of different sources that is consistent with the electric production used at Camp Atterbury.

REFERENCES

(1998). *AP 42, Fifth Edition, Compliation of Air Pollutant Emission Factors, Vol. 1, Ch 1.4.* EPA.
(1999). *Positive displacement refrigerant compressors and compressor units.* ANSI/ARI Standard 540.
(2010). *Energy Security Initiatives.* Department of Defense Security Task Force.
Average Annual Price. (2011). U.S. Energy Information Administration.
Deru, M., & Torcellini, P. (2007). *Source Energy and Emission Factors for Energy Use in Buildings.* National Renewable Energy Laboratory.
DOE Buildings Energy Data Book. (n.d.).
Kline, S. A. (n.d.). Engineering Equation Solver. http://fchart.com.
Lutz, J., Franco, V., Lekov, A., & Wong-Parodi, G. (2006). BPM Motors in Residential Gas Furnaces: What are the Savings? *ACEEE Conference.* Lawrence Berkeley National Laboratory.
Roth, K. W., Westphalen, D., Dieckmann, J., Hamilton, S. D., & Goetzler, W. (2002). *Energy Consumption Characteristics of Commericial Building HVAC Systems Volume III: Energy Savings Potential.* Cambridge: Building Technologies Program DOE.

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