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# Field Investigation of an Air-Source Cold Climate Heat Pump

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**Abstract**

In the U.S., there are approximately 2.6 million dwellings that use electricity for heating in cold and very cold regions with an annual energy consumption of 0.16 quads (0.17 EJ). A high performance cold climate heat pump (CCHP) would result in significant savings over current technologies (greater than 60% compared to electric resistance heating). We developed an air-source cold climate heat pump, which uses tandem compressors, with a single compressor rated for the building design cooling load, and running two compressors to provide, at -13°F (-25°C), 75% of rated heating capacity. The tandem compressors were optimized for heating operation and are able to tolerate discharge temperatures up to 280°F (138°C). A field investigation was conducted in the winter of 2015, in an occupied home in Ohio, USA. During the heating season, the seasonal COP was measured at 3.16, and the heat pump was able to operate down to -13°F (-25°C) and eliminate resistance heat use. The heat pump maintained an acceptable comfort level throughout the heating season. In comparison to a previous single-speed heat pump in the home, the CCHP demonstrated more than 40% energy savings in the peak heating load month. This paper illustrates the measured field performance, including compressor run time, frost/defrosting operations, distributions of building heating load and capacity delivery, comfort level, field measured COPs, etc.

## 1. Introduction

As described by Khowailed et al. [2], in the U. S., the primary target market for cold climate heat pumps (CCHP) is the 2.6 million U.S. homes using electric furnaces and conventional air-source heat pumps (ASHP) in the cold/very cold region, with an annual energy consumption of 0.16 quads (0.17 EJ). A high performance air-source CCHP would result in significant savings over current technologies (greater than 60% compared to electric resistance heating). It can result in an annual primary energy savings of 0.1 Quads (0.1055 EJ) when fully deployed, which is equivalent to 5.9 million tons (5.35 million MT) of annual $CO_2$ emissions reduction. In cold climate areas with limited access to natural gas, conventional electric ASHPs or electric resistance furnaces can be used to provide heating. During very cold periods, the ASHPs tend to use almost as much energy as the


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electric furnaces due to their severe capacity loss and efficiency degradation. Presently, technical and economic barriers limit market penetration of heat pumps in cold climates. R&D efforts should be employed to overcome these barriers and develop high performance CCHPs that minimize, or even eliminate, the need for backup strip heating.

A typical single-speed ASHP doesn't work well under cold outdoor temperature conditions typical of cold climate locations for three major reasons:

1. Too high discharge temperature: low suction pressures and high pressure ratios at low ambient temperatures cause significantly high compressor discharge temperatures, in excess of the maximum limit for many current compressors on the market. Furthermore, system charge of a heat pump is usually optimized in cooling mode, which leads to overcharge conditions in heating mode, further increasing the discharge temperature.
2. Insufficient heating capacity: heating capacity decreases with ambient temperature. The heating capacity at -13°F (-25°C) typically decreases to 20% to 40% of the rated heating capacity at 47°F (8.3°C) (~equivalent to the rated cooling capacity at 95°F (35°C)). As such, a single-speed ASHP, sized to match the building cooling load, is not able to provide adequate heating capacity to match the building heating load at low ambient temperatures, and supplemental resistance heat has to be used.
3. Low COP: heating COP degrades significantly at low ambient temperatures, due to the elevated temperature difference between the source side and demand side.

For the CCHP development, cost-effective solutions should be identified to tackle these three issues. US Department of Energy (DOE) has set stringent performance targets for CCHPs as follows: 1) maintain at least 75% of the rated space heating capacity at -13°F (-25°C), and 2) have a rated heating COP at 47°F (8.3°C) greater than 4.0. The 75% capacity criterion would result in a heat pump capacity approximately equal to the building heating load for a well-insulated home at -13°F (-25°C) in US climate Region V, for example, Minnesota (assumed to be the DHRmin load condition as defined by AHRI Standard 210/240 [1] for Region V), where the building heating load at -13°F (-25°C) is 80% of the building cooling design load at 95°F (35°C) ambient temperature.

Shen et al. [3] discussed the development of a cost-effective CCHP, using two equal, single-speed compressors (tandem) as shown in Figure 1. Note that the system is relatively simple and comparable to conventional ASHPs with the exception of having two compressors in parallel, thus it is considered to be relatively more cost-effective than more complex, variable-speed design approaches. The design considerations are summarized as below:

1. The two equal, single-speed compressors were provided with special "heating application" design features that allow the compressors to operate at higher discharge temperatures than most typical compressors (up to 280°F (138°C)). This enables the heat pump to operate at extremely low ambient temperatures.
2. Current two-speed heat pumps on the market use a single, two-stage compressor having a typical displacement volume split ratio of 100%-to-67%. In comparison, the tandem compressors have a volume split ratio of 100%-to-50%, which provides a larger extended-capacity potential, if the heat pump nominal COP and capacity ratings are established while running one compressor. That is the primary reason that the heat pump using the tandem compressors can reach greater than 75% capacity at -13°F (-25°C).
3. The CCHP is sized to match a 3-ton (10.6 kW) building cooling load using a single compressor. The system uses heat exchangers of a typical 5-ton (17.6 kW) heat pump. With a single compressor running (cooling mode and moderate temperatures in heating mode), the heat exchangers are under-loaded, and this provides higher system efficiency. That is the key that enabled the CCHP laboratory prototypes to reach a COP > 4.0 at 47°F (8.3°C).
4. The compressor(s) and discharge line are well insulated and placed outside the outdoor air flow stream, so as to minimize the shell heat loss. Insulating the compressors reduces the cooling performance slightly by increasing the heat rejection load on the condenser; however, its effect is negligible, since the condenser (outdoor heat exchanger) has been oversized for cooling mode.

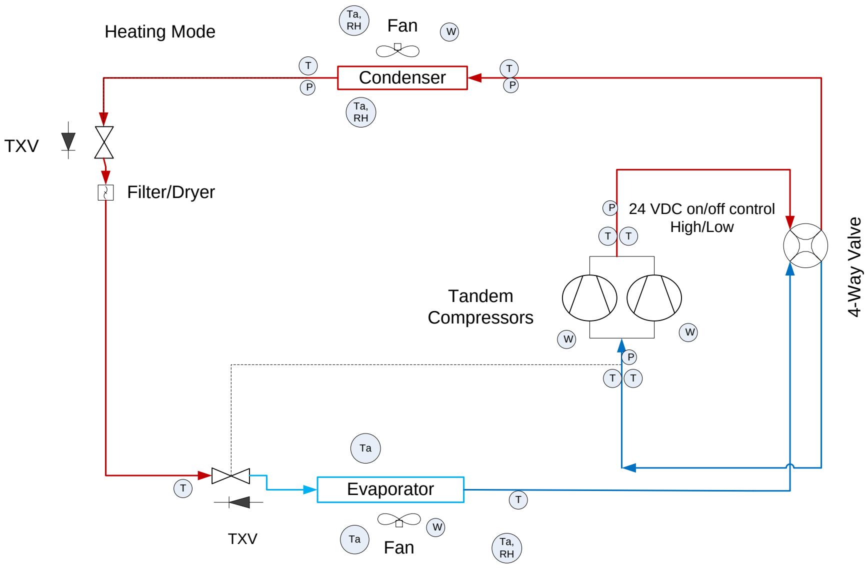


Figure 1: System diagram of field testing CCHP and instrumentation

## 2. Field Installation

We developed a CCHP laboratory prototype which achieved all the US DOE's performance goals, and proceeded to a field investigation. An occupied, single-story ranch home in Ohio, USA, was selected to host the field testing. One CCHP breadboard unit was used to replace a previous single-speed HP, having a 3.0-ton (10.6 kW) design cooling load. It's easier to configure the system control for tandem compressors than variable-speed compressors since we can utilize a regular 2-stage thermostat and controller. For the field testing unit, the control is a typical 2-speed ASHP control. We re-wired the 24 VAC signal with a relay to call the second compressor. We used a standard 2-stage thermostat. Its Y1 signal calls the first stage and the Y2 signal calls the second stage. Each stage corresponds to an individual indoor air flow rate, i.e. low or high air flow rate, but with the same outdoor air flow rate. The defrost control is an on-demand control. It measures the coil surface temperature and senses the temperature difference between the ambient air and the coil surface to start the defrosting cycle. During the defrosting cycle, it always runs two compressors, i.e. the high stage, to accelerate the defrosting with the indoor air flow on.

As illustrated in Figure 1, a thermal expansion valve (TXV) was used to control the evaporator superheat degree around 10 R (5.6 K). It should be mentioned that the system charge was optimized for heating mode, i.e. 9% lower than the cooling mode optimized charge. The optimized charge led to 3 R (1.7 K) condenser subcooling at 82°F (27.8°C) in cooling mode when running one compressor and around 20 R (11.1 K) subcooling at 17°F (-8.3°C) in heating mode when running two compressors. Figure 2 shows the installed outdoor unit for field testing, where one can see the compressors were wrapped by a thermal insulation layer.

Figure 2: Outdoor unit of field test CCHP unit

The thermostat was located in the hallway. If the Delta-T between the temperature setting and the zone temperature is less than 1°F (0.6°C), the thermostat calls a single-compressor running (Y1); if it is greater than 1°F (0.6°C), it calls the second stage. When the Delta-T goes beyond 2°F (1.1°C), the supplemental resistance heat will be activated.

In the field, the air temperatures into and out of the outdoor coil were measured using T-type thermo-couples. The outdoor humidity was monitored using a relative humidity sensor. Three thermo-couples were evenly placed at the entrance of the indoor unit to measure the average return air temperature, and a RH sensor was used to measure the return RH. At the outlet of the indoor coil, and upstream of the blower, three thermocouples and a RH sensor were used to monitor the supply air state. T-type thermo-couples were soldered on tube wall to measure the refrigerant temperatures entering and leaving the indoor coil, and also, the suction and discharge temperatures of each compressor. Four pressure transducers were used to measure the refrigerant pressures entering and leaving the indoor coil, as well as entering and leaving the compressors. Four watt transducers were used to measure the power of the outdoor fan, indoor blower and two compressors, individually. In addition, one watt transducer was used to measure the total power consumption of the outdoor unit. The total outdoor power consumption was determined using the larger value between the total power measurement and sum of the individual power measurements. In particular, we put a thermocouple in the duct downstream of the indoor unit to sense when the electric supplemental heater was on or off. The supplemental heater was placed after the indoor blower. The data acquisition system scanned all the sensors and recorded the data every half minute.

The field testing was conducted in the occupied home with its existing ductwork. To minimize the interruption on the home owner, we didn't install an air flow monitor in the duct. Instead, we used a grid of pitot tubes to measure the air flow rates for one time during the heating season, respectively for the low and high air flow rates. The air flow rates were considered constant through the heating season, because the indoor blower uses an electronically commutated motor to control a constant airflow rate at each stage, regardless the duct pressure drop.

## 3. Field Heating Performance

The field testing in the 2015 heating season was monitored from the beginning of February to the end of April for three full months. We were able to capture the coldest condition in Ohio when the field temperature went down to -13°F (-25°C).

Figure 3 illustrates runtime fractions for both total compressor run time (running one and two compressors) and for high stage operation (running both compressors) vs. 5°F (2.8°C) ambient temperature bins. It can be seen that the second compressor operated more frequently at lower ambient temperatures. At -13°F (-25°C), the total compressor run time was 100%, but the second compressor still cycled with 80% running time, indicating that the CCHP system still had extra capacity capability even at this extreme cold condition.

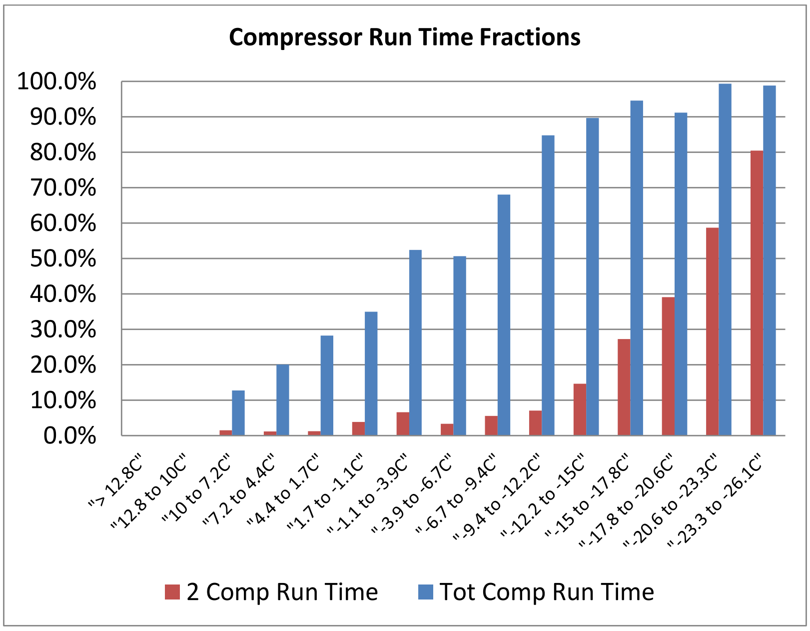


Figure 3: Compressor run time fractions

Figure 4 shows the delivered space heating capacities of the CCHP with one compressor and with two, in comparison to the total house heating load line. It can be seen that the second compressor was needed when the ambient temperature went below 10°F (-12.2°C). At -13°F (-25°C), running two compressors delivered 30,416 Btu/h (8.9 kW), which is 75% of the rated capacity of 39,717 Btu/h (11.6 kW).

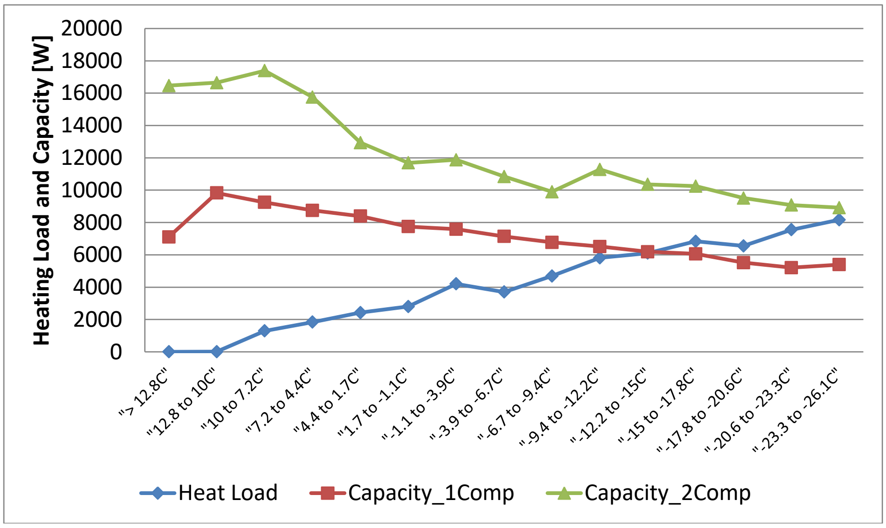

Figure 4: Delivered heat capacities and measured building heating load line

Figure 5 illustrates the fraction of CCHP energy use due to the supplemental resistance heaters in each temperature bin, relative to the total energy use in each bin. The overall resistance heat use was negligible and primarily caused by control issues. At -13°F (-25°C), the resistance energy use was 3.2% even though the second compressor still cycled by 80% (extra capacity was available). This means that the CCHP responded more slowly to the increased heat demand than required causing the thermostat to reach the 2 R (1.1 K) dead band level and trigger the back-up heaters first. A change in the control approach to prevent running a single compressor below a certain ambient temperature could have eliminated most, if not all, of the supplemental heater use during the test period. It is interesting to see that there was some supplemental resistance heat use even at moderate ambient temperatures between 45°F and 20°F (7.2°C to -6.7°C). This was mainly due to the home owner's action to reduce the thermostat setpoint when leaving the house. Upon returning home the home owner would increase the thermostat temperature setting, often to a level where the difference between the new set point and the room temperature exceed the 2 R (1.1 K) dead band level, causing some supplemental heat use.

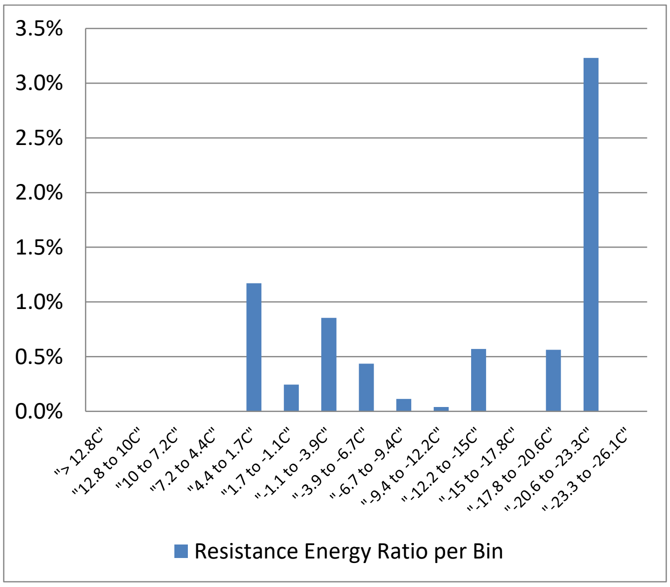


Figure 5: Supplemental resistance heat uses

Figure 6 shows the return and supply air temperatures. The return temperature profile indicates that the home owner set the thermostat at 68°F during the majority of time when the heat pump was operating. This caused the return air temperature to change from 66°F to 70°F (18.9°C to 21.1°C). At -13°F (-25°C), the CCHP was able to deliver the supply air at 86°F (30°C) out of the indoor blower and before the resistance heater, with the high second stage air flow rate of 1350 CFM (0.64 $m^3/s$). Recall that the field test system controller changed the indoor blower speed from low to high when the system went from first to second stage. There was no attempt during the field test to modulate the indoor blower speed based on the outdoor temperature to modulate the supply air temperature.

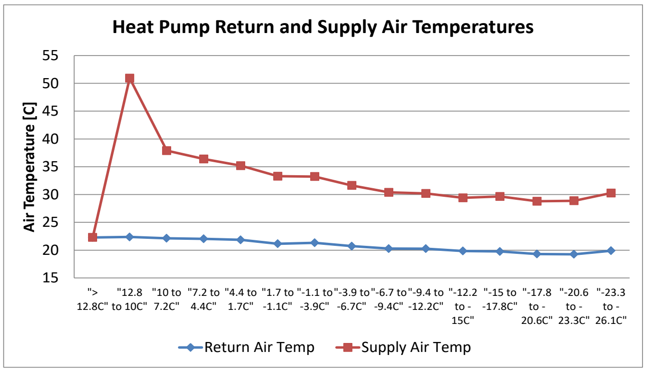


Figure 6: Return and supply air temperatures

Figure 7 presents loads due to defrosting operation, compared to the total heating energy delivered in each temperature bin. The defrost loads were calculated as the temperature decrease across the indoor air handler multiplied by the indoor air flow rate, accumulated during the defrosting cycles in each temperature bin. It is clear that defrost operation and resultant energy losses were minimal for the CCHP, for two reasons: 1) at the temperature range most prone to frost growth (roughly +5 to -10 °C) frost formation was slow because only one compressor was running most of the time and outdoor HX was relatively oversized leading to higher evaporating temperature than with a typical ASHP; 2). When two compressors were needed at lower ambient temperatures, the humidity level was very low and hardly any moisture condensed on the outdoor coil.

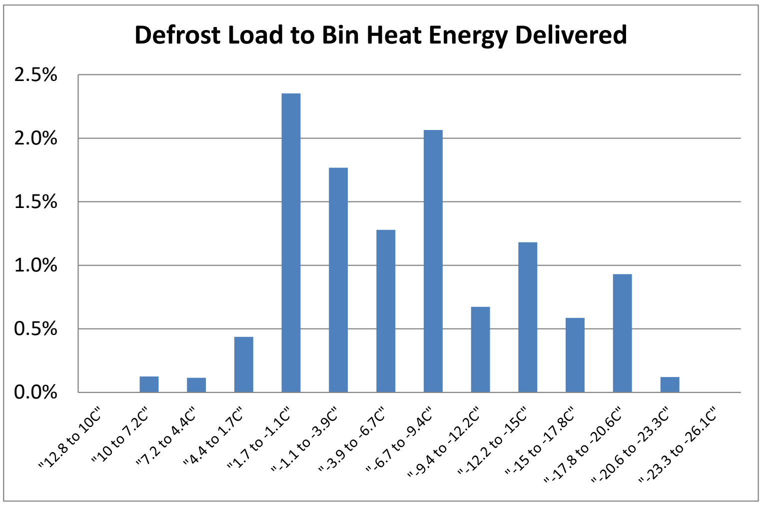


Figure 7: Defrost load relative to capacity delivered in each bin

Figure 8 shows the heating COP vs. outdoor temperature bin for single- and two-compressor operation along with total COP. The total COP was calculated as the total energy delivery divided by the total energy consumed for each bin, including the effects of cyclic losses, supplemental resistance heat use, frosting/defrosting losses, and switching between running one compressor and two compressors. It can be seen, from 45°F to 50°F (7.2°C to 10°C), the average COP for single compressor operation is 4.05. The average total COP for the same bin is 3.83; lower than the one-compressor COP due to cyclic losses and occasionally running the second compressor. Adjustment of the control system prevent two-compressor operation at moderately temperatures would allow the total COP to more closely follow the one-compressor COP curve. It is encouraging to see that, at -13°F (-25°C), the total COP was 2.2 i.e. 120% more efficient than resistance heating. The overall seasonal average heating COP for the test period was measured as 3.16.

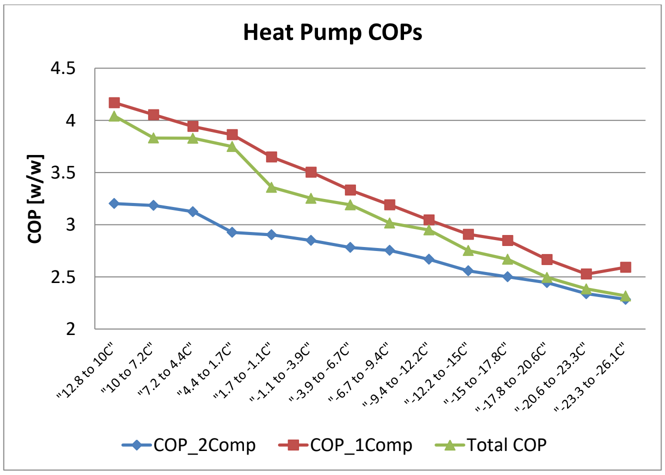


Figure 8: Field COPs in heating mode

## 4. Electric Bills Before and After Installing the CCHP

We collected the electric bills of the field testing home in the past three years. As shown in Figure 9, in comparison to the previous single-speed ASHP in the same house, >40% energy reduction was achieved during the coldest months with similar average temperatures around 20°F (-6.7°C).

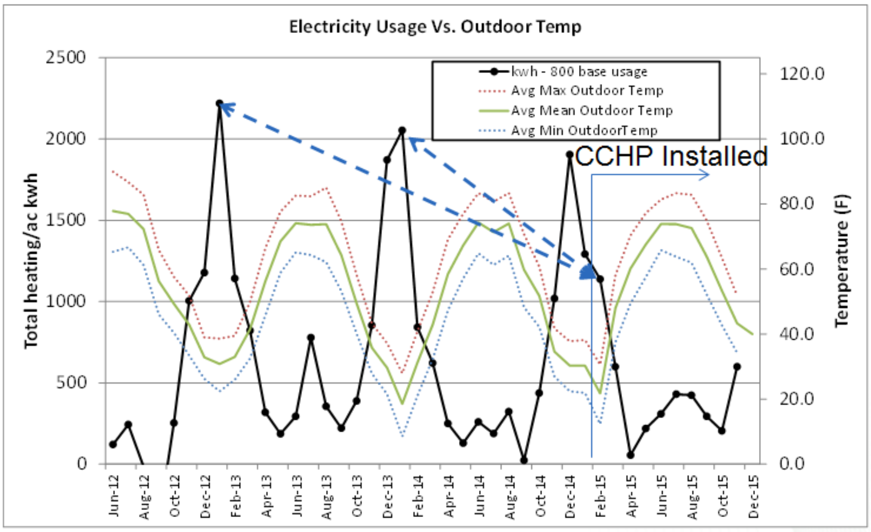


Figure 9: Comparing electric bills of the field testing home before/after installing the CCHP with tandem single-speed compressors.

Figure 10 overlays the field measured average daily energy use vs. outdoor temperature (kwh/day), i.e. total measured kWh at a certain ambient temperature divided by the total time at the temperature, with the electricity bills, as a function of the average ambient temperature. It can be seen that the field data is very close to the CCHP electricity bills, which indicates good field measurement accuracy.

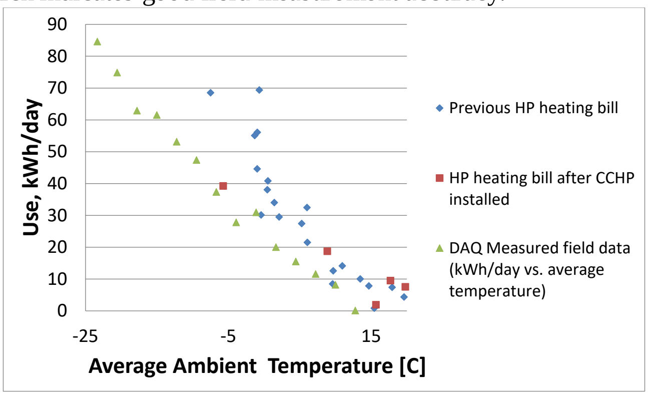


Figure 10: Overlay of the field testing energy use data with the electricity bills

## 5. Conclusions

We developed and tested a high efficiency CCHP in an occupied residential home in Ohio, in the central U.S. The CCHP is a cost-effective design, very similar in concept to conventional single-speed ASHPs. The primary design enhancement is the use of two, identical, parallel single-speed compressors (optimized for heating operation) along with the added control features to regulate the compressor staging. It uses one compressor to meet the building cooling load and heating load at moderate ambient temperatures, and turns on the second compressor at low ambient temperatures to augment the heating capacity. The field investigation demonstrated advantages of the CCHP as below:

1. It works with widely available two-stage unit controls and thermostats, which can be set up easily in the field without needing a manufacturer specific variable-speed control.
2. It operated down to -13°F (-25°C) ambient temperature, and provided adequate heating capacity (>75% rated capacity), without violating the compressor discharge temperature limit. It demonstrated the feasibility to eliminate the need of supplemental resistance heating at the test location.

3. It had minimum frost/defrosting loss, mainly because running a single compressor with an outdoor heat exchanger sized for two compressors caused slow frost growth in the outdoor temperature range most prone to outdoor HX frosting (e.g., from 20°F (-6.7°C) to 40°F (6.7°C)).
4. It achieved a field measured seasonal heating COP >3.0. At -13°F (-25°C), the heat pump COP was larger than 2.0. At the 47°F (8.3°C) rated temperature, the field COP, including cyclic loss, was 3.8.
5. In comparison to a previous, conventional single-speed ASHP in the test home, the CCHP achieved >40% energy saving during the coldest month.

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## References

[1] ANSI/AHRI, 2012. *Standard 210/240 with Addenda 1 and 2, 2008 Standard for Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment*, Air Conditioning, Heating, and Refrigeration Institute, Arlington, VA, USA[2]

[2] Khowailed, G., K. Sikes, and O. A. Abdelaziz., 2011. *Preliminary Market Assessment for Cold Climate Heat Pumps*, ORNL/TM-2011/422, Oak Ridge National Laboratory, August.

[3] Shen, B., Omar Abdelaziz, Keith Rice, Van Baxter and Hung Pham, 2016. "Cold Climate Heat Pumps Using Tandem Compressor", Conference Paper in 2016 ASHRAE Winter Conference, Orlando, FL.

[4] Shen B., Rice, C. K., Abdelaziz O., "Compressor Selection and Equipment Sizing for Cold Climate Heat Pumps." Proc. 11th IEA Heat Pump Conference, May 2014, Montreal, Canada

[5] Bertsch, S. S. (2005). "Theoretical and experimental investigation of a two stage heat pump cycle for nordic climates" (Doctoral dissertation, Mechanical Engineering, Herrick Labs 2005-13P, Report). Purdue University, West Lafayette, IN, USA

[6] Bertsch, S. S., and Groll, E. A. (2008). Two-stage air-source heat pump for residential heating and cooling applications in northern US climates. International Journal of Refrigeration, 31(7), 1282-1292.