# Final Scientific/Technical Report

**Project Title:** Residential Cold Climate Heat Pump (CCHP) w/Variable Speed Technology

**Award Number:** DE-EE0006107.000

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**Project Duration:** March 1, 2013 to February 29, 2016

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**Disclaimer**:



**Acknowledgement**:

This report is based upon the work supported by the Department of Energy under Award Number DE-EE0006107.000. We gratefully acknowledge the contribution made by Oakridge National Labs and Invention House, LLC.

## Executive Summary

The purpose of this project was to develop a residential, split-system, 3-5-ton, Cold Climate Heat Pump (CCHP) using *boosted compression* and technologies such as *variable speed compressors, advanced compact heat exchangers*, *electronic expansion valves* (EEV), *electronically commutated motors* (ECM) and microprocessor controls. The industry measures the efficiency as the Coefficient of Performance (COP), defined as the ratio of heat moved divided by the work to move it. Unico will attempt to meet the efficiency goals in Table 1 with a simple payback of less than 5 years…

**Table 1:** Project End Goals

| Outdoor Temperature, °F (°C) | Capacity, Btu/hr (kW) | COP (W/W) |
|---|---|---|
| +47 (+8.3) | +36000 (10.5) | 4.0 |
| +17 (-8.3) | 92% | 3.5 |
| -13(-25) | 75% | 3.0 |

Our proposal was based on the design used by Hallowell International whose patents and intellectual property is now held by their receivership bank. We also considered a competing technology developed by Purdue University. After careful consideration we decided that the Purdue design was more promising. We negotiated for the right to use their IP for small capacity residential products (single phase, less than 65,000 Btu/hr).

Heat pumps reduce energy use and carbon output, so the benefit to the general public is evident with the inherent conservation of energy and reduced negative impact on the environment. Residential CCHPs are defined as split-system units with capacities less than 65,000 Btu/hour. Unico proposed to develop a residential CCHP that would maintain capacity and efficiency at very low temperatures.

All air-source heat pumps exhibit a reduction in capacity as the outdoor temperature decreases. This reduction of capacity can be partially compensated for by oversizing a single stage compressor or by using multi-capacity (or variable speed) compressors. This is not the only problem; efficiency also decreases with decreasing outdoor temperature for existing heat pump designs. The fundamental thermodynamic process of a gas heating up as it is compressed creates two problems. First, extreme temperatures can damage the equipment or fluids; and second, it is a significant source of thermodynamic inefficiency.

In addition, existing compressors are not designed for the pressure ratios needed to raise the refrigerant temperature from the low outdoor temperature to the higher indoor temperatures. Various designs attempt to solve these problems, such as liquid injection compression, variable speed compressors, and variable capacity compressors. These solutions improve low temperature capacity but not efficiency and none are very effective at temperatures below 0°F. The reason is simple; they all employ a single compression stage (single compressor). What is needed is multi-stage compression, or “boosted” compression. This will provide the necessary pressure ratio (as high as 16:1); and with an economizer (intercooler), the thermodynamic efficiency is greatly improved.

Present day air-source heat pumps are by far the most prevalent type of heat pump used in the world today. Included in this category are: package terminal units, self-contained air conditioners, split system, ductless, air-to-water heating units, domestic hot water, dehumidification type, and many other varieties of residential systems. The simplest and most common heat pump design uses one single-speed compressor. The main problem is that the heat pump cannot adjust its capacity. It either does too much (short cycles when it is relatively warm outside) or it does too little (insufficient capacity when it is relatively cold outside). It is only perfectly sized at a single outdoor temperature.

Below the balance point, supplemental heating from some other source(s) of energy is required. The most prevalent supplemental source is large electric resistance heating coils located within the units. Electric heat elements allow the system to meet the load but are not efficient (COP = ~1.0). Even worse, when operating at the lower outdoor ambient temperatures, buildings heated by conventional heat pumps require just as much power input from the electric utility on very cold days as does a building heated only by electric resistance. In many localities, especially rural locations, the cost per btu from electricity is higher than that from fossil fuels. Thus, in cold climates, the use of electric resistance heat puts the typical consumer with an air-source heat pump at an economic disadvantage as compared with the other common sources of heating energy such as natural gas and oil.

The increased cost to operate with electric heat elements might be acceptable in mild climates, where they are not used as often. However, heat pumps have become so prevalent in mild, southern climates that electric utilities located there are now experiencing “winter peaking demand loads.” Therefore, even in the milder climates, the use of electric resistance heating must be reduced, providing an even greater market for the CCHP.

## Project Goals and Objectives Comparison to Actual Results

The objective of this project was to develop a high performing, efficient, and economically acceptable residential split-system CCHP using boosted compression technology. The goals are to maintain capacity throughout the operating range as constantly as possible while maintaining excellent COPs and to do this with a product that has a 5-year payback or less; see Table 2 for P4 testing results compared to goals.

**Table 2:** Discrete Project Targets for Capacity and Efficiency

| Criteria | **Goals** | | **P4** |
|---|---|---|---|
| | Final | Budget Period 2 | Testing Results |
| Capacity @ 47 °F | > 36,000 Btu/hr. | > 29,520 Btu/hr. | > 58,200 Btu/hr. |
| Capacity @ 17 °F | 92% of 47°F capacity | 82% of 47°F capacity | >100% of 47°F capacity |
| Capacity @ -13 °F | 75% of 47°F capacity | 69% of 47°F capacity | >100% of 47°F capacity |
| COP @ 47 °F | 4.0 | 3.28 | 3.55 |
| COP @ 17 °F | 3.5 | 2.87 | 2.75 |
| COP @ -13 °F | 3.0 | 2.46 | 2.45 |

Our actual test data in Table 2 (right column) shows that the capacity goals were being met. Efficiency, on the other hand, is more of a challenge as we knew it would be. The goals for the end of budget period two are to achieve 92% of capacity and 82% of COP. The results of the testing show that we are exceeding capacity goals and meeting all efficiency (COP) targets except for -13°F and narrowly missing the mark at 17°F.

## Summary of Project Activities

*Budget Period 1*

In order to facilitate testing of a CCHP, Unico designed and constructed a CCHP testing laboratory equipped with psychrometric test chambers with the capability to quickly reach and hold extreme sub-zero temperatures as low as -30°F. The cold chamber has two refrigeration units staged to bring the temperature down quickly. Initial results brought the chamber temperature down to -28°F in under 5 hours. This allowed us to change conditions quickly and save time during development and conduct cyclic testing.

The chamber temperatures and humidity levels are further maintained by three parallel air handlers with low temperature chilled water, electric heaters, and steam humidifiers.

Below are several photos (Figures 1-3) of the psychrometric test chambers as they appeared in mid-January 2014.

**Figure 1:** Dual-chamber environmental test cell (Chambers #2 and #3)

**Figure 2:** Outdoor test chamber (Chamber #3). Intermediate temperature refrigeration system is shown upper left. (Used to pull temperature down from 30°F to 10°F)

**Figure 3:** Backside of Chambers #2 and #3 showing the process equipment. The process control air handlers for the outdoor test cell are on the right. The chilled water storage tank is on the left.

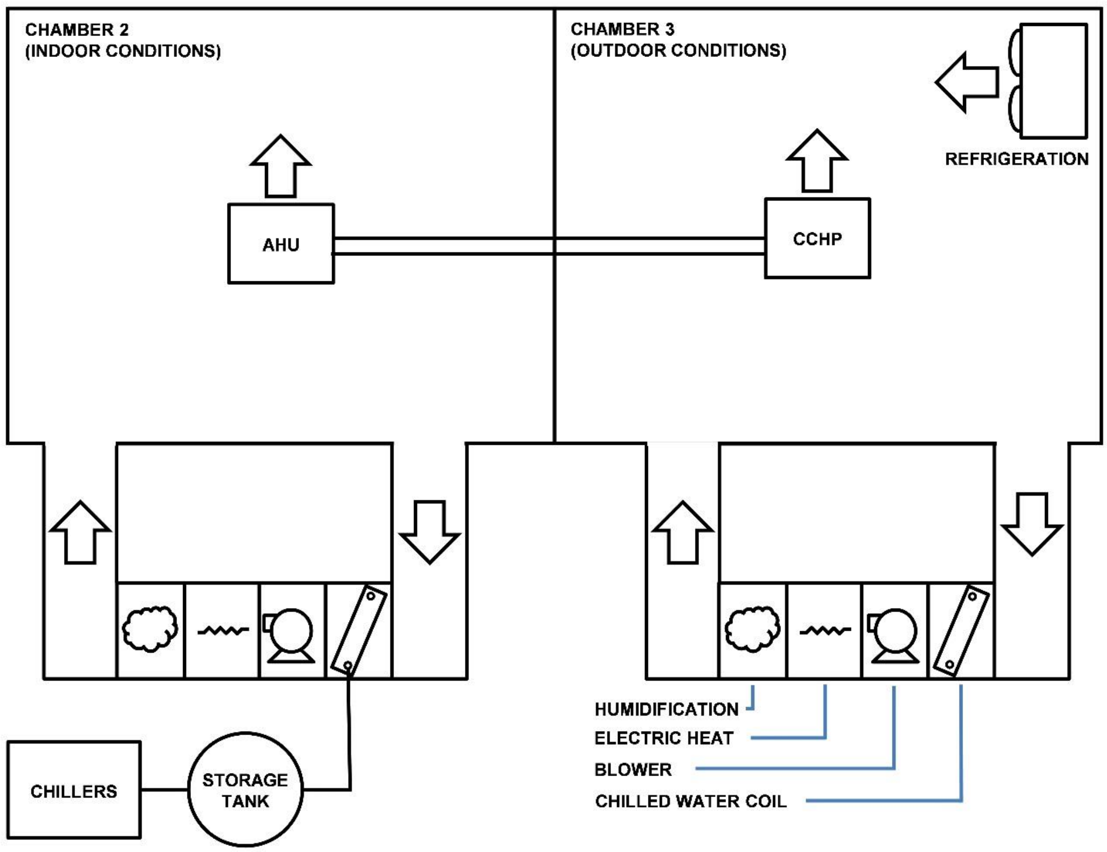



**Figure 4:** Function diagram of process equipment

In addition to the lab activities, the first phase of this project was to model various compressor configurations and choose the most promising. Oak Ridge National Labs performed the modeling using their simulation software with input data from the CCHP component vendors. The main component driving the performance of the system is the compressor. Both Danfoss and Emerson Copeland compressors were considered. Our proposal was based on variable speed compressors. The ORNL analysis showed that the Emerson compressors would perform better, in terms of both efficiency (COP) and capacity (Btu/hr). Therefore, Emerson Copeland was chosen.

As part of our analysis, we considered alternate compressor configurations. We considered over 10 different alternatives, two of which were especially promising and are the basis of our prototypes (see Table 3). The modeling data for *heating* is shown in the table.

Although the proposal was based on a 3-ton unit, after reviewing the compressors available, we believe that we needed to maximize the heating capacity to the greatest extent possible. The largest single phase compressors are nominally 5-ton capacity. The actual capacity is somewhat smaller and is shown in the modeling.

**Table 3:** Modeling results of the two top configurations for heating

| Compressor #1 | Compressor #2 | Modeling Results | | |
|---|---|---|---|---|
| | | Outside temp, °F | Btu/hr | COP |
| Variable speed 5-ton | Variable speed 5-ton | 47 | 59,590 | 4.0 |
| | | 17 | 54,054 | 3.01 |
| | | -13 | 44,407 | 2.11 |
| Dual-stage tandem 5-ton | Dual-stage 5-ton | 47 | 60,516 | 3.85 |
| | | 17 | 57,941 | 2.9 |
| | | -13 | 46,015 | 2.14 |

The first prototype built was a cooling only single compressor system. The purpose was to develop the basic control software for the compressor inverter drive. We developed the control software to manually change the speed of the compressor and expansion valves. We also developed an “auto” mode to control these devices in order to achieve a given setpoint. The photo in Figure 5 is the first prototype.

**Figure 5:** First prototype, cooling only, single variable speed compressor

To control the system we developed a laptop program to set the various functions and to display the internal data. Figure 6 is a screen shot of the first control software.

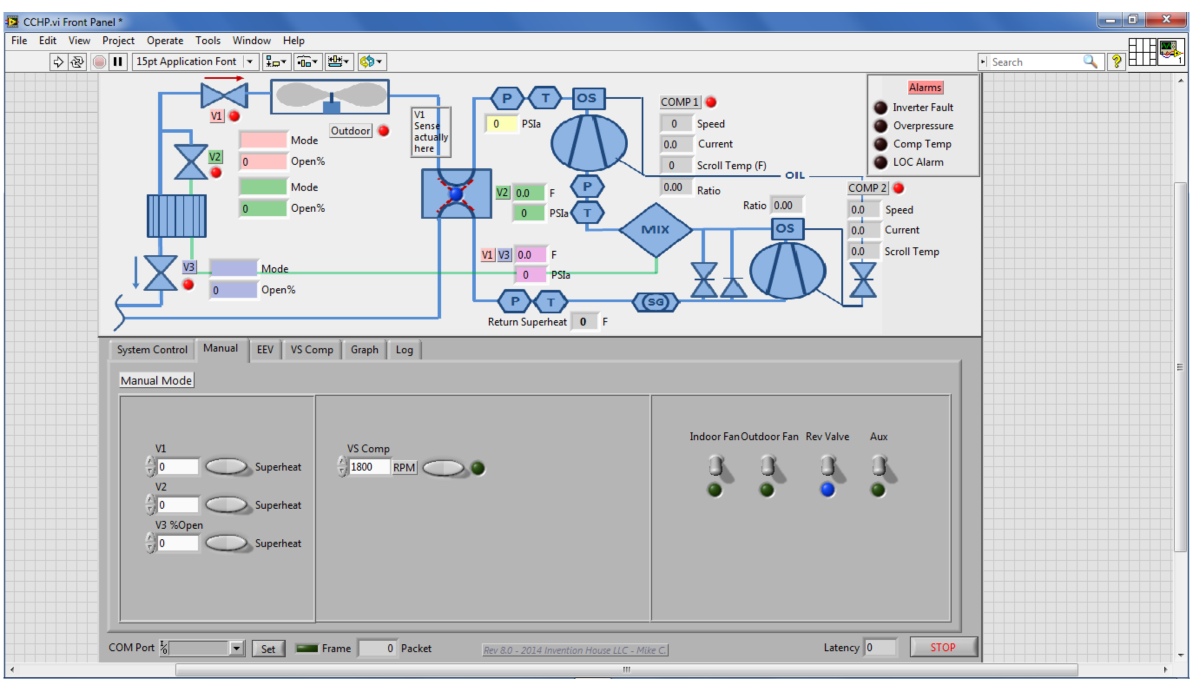



**Figure 6:** Initial Labview control program (screen shot)

After successfully controlling the devices, the prototype was rebuilt as a functioning dual compressor heat pump system. The first challenge was to communicate with two compressor inverter drives simultaneously. This required a new drive from Copeland to give us the option to change the communication address on the MODBUS system. We successfully ran the unit in cooling (one compressor), heating (one compressor), and heating (dual compressor).

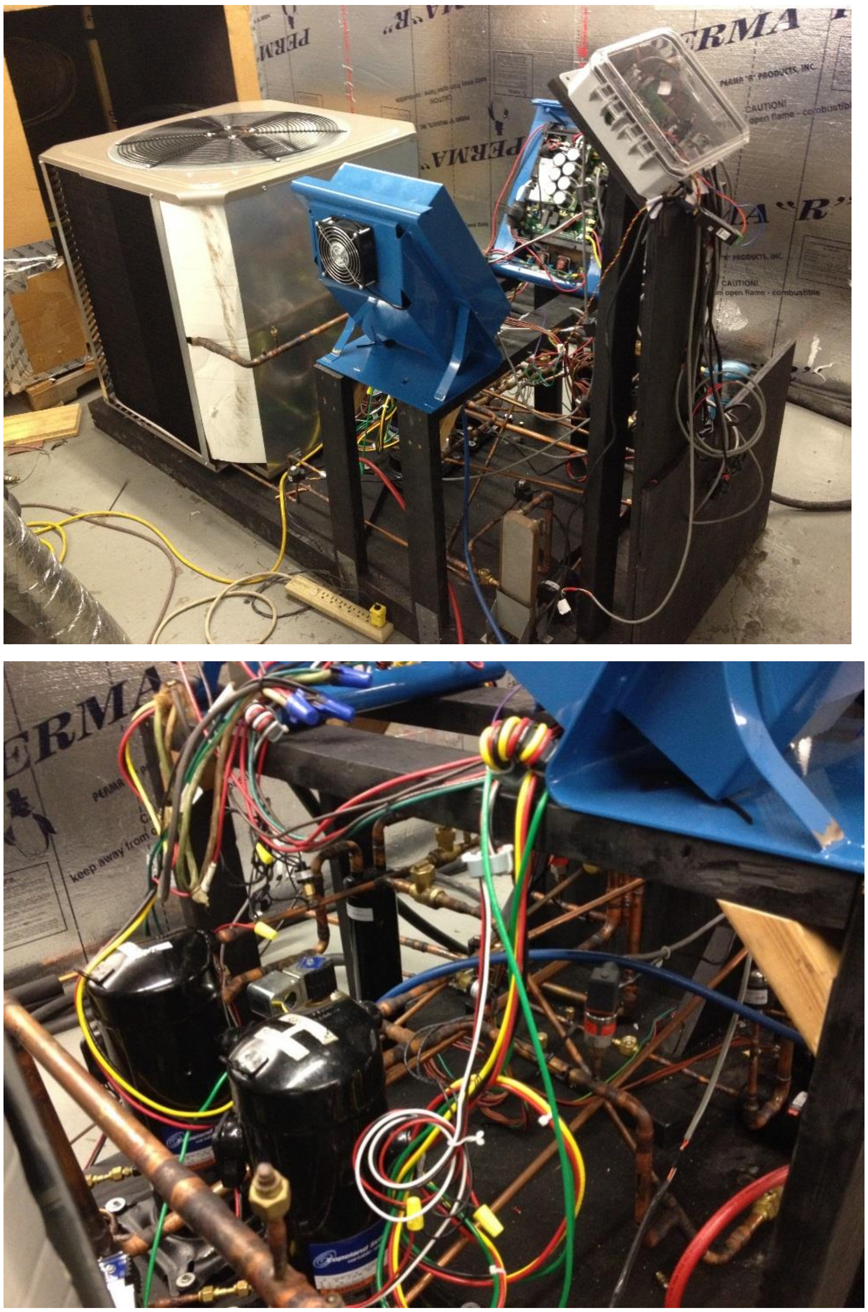


**Figure 7:** Dual compressor prototype

The auto-function allowed the system to maintain a fixed evaporating temperature in cooling and a fixed condensing temperature in heating. The user can manually change the setpoints.

*Budget Period 2*

Additional thermocouple channels were added to support coil circuit temperature measurement; this allowed us to identify if we were experiencing unequal temperature distribution between coil circuits. In addition, the data acquisition system (DAQ) was installed, programmed, and functioning accurately. See figure 8 below of the DAQ rack outside of the outdoor chamber.

**Figure 8:** Data Acquisition Rack

One of the outdoor test chamber refrigeration units failed in mid-2014 and needed to be repaired. This took several weeks of troubleshooting. Fortunately, this occurred mostly between the testing of prototypes. It pointed out the continued need to maintain the lab equipment because of the complexity of achieving very cold and stable operating conditions and temperatures.

The first half of budget period two centered around two promising prototype designs:

1) *two* variable speed compressors (prototypes P1 and P3)

2) *three* two-stage compressors (prototypes P2 and P4)

In July 2014, we chose to pursue the second design (see figure 9) with three compressors. The second design was less expensive overall (see Table 4) and less complex from a controls standpoint. In addition to cost and complexity, there were unforeseen challenges supplying the second generation variable speed compressors. Shipments were held back due to performance shortfalls experienced during supplier testing.

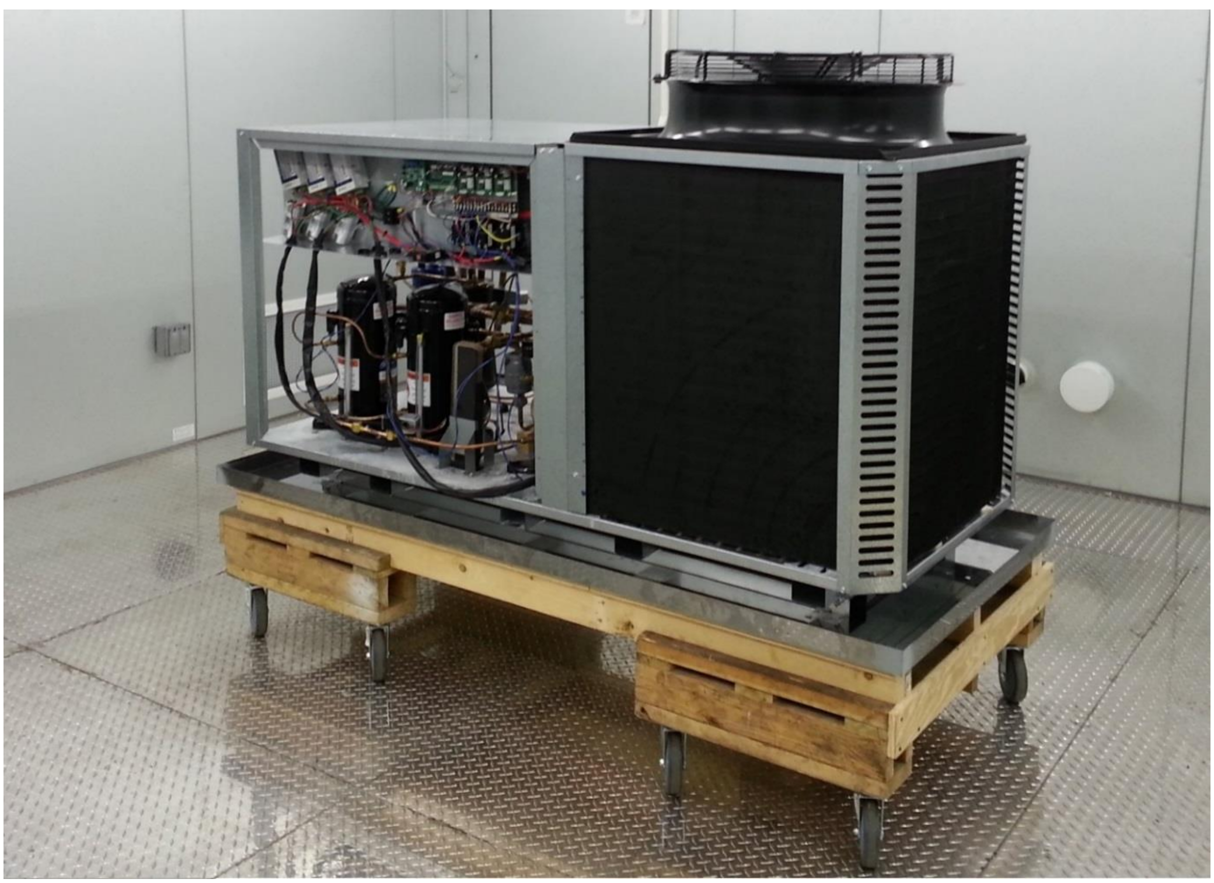


**Figure 9:** P2 prototype; built in second quarter 2014

**Table 4:** Estimated costs between Variable speed and Staged units

| Component | Component Details | VS Unit Cost Estimate | Staged Unit Cost Estimate |
|---|---|---|---|
| **Major components** | Compressors, Oil Separators, Accumulator, | $1,841 | $1,759 |
| **Coil** | | $636 | $636 |
| **Tubing** | | $461 | $500 |
| **Electrical** | (Controls, Valves, Sensors, wiring) | $2,009 | $840 |
| **Sheet metal** | | $383 | $411 |
| **Fan Module** | | $173 | $173 |
| | **TOTAL COST** | $5,503 | $4,319 |
| | **Cost for unit at 4x multiple** | $22,011 | $17,277 |
| | **Cost difference to consumer** | $4,734 | |

Unico Inc. negotiated for the right to use Purdue University's IP for the unit design which would be applied to small capacity residential products (single phase, less than 65,000 Btu/hr.). Our point of contact in Purdue's business office changed a few times, so final negotiations with Purdue were completed in May of 2014.

While testing the initial prototypes, we experienced uncertainty as to where, how much, and if any oil was traveling with the system's refrigerant charge. Unfortunately, the inherited oil management strategy made assumptions about system pressures and gravity which were not verified by inspection or fluid measurement. In order to monitor the oil level, we added sight tubes and oil level indicators with max and min levels marked on the clear sight tubes which were connected to the underside of the compressor.

Much of the year and budget was spent developing a complete electrical system controls package with excellent power factor correction, stand-alone software, and commission it for the P4 field trials. The controls consist of a main board, power relay board, EEV board, sensors, and a user interface display with buttons as seen below in figures 10 and 11.

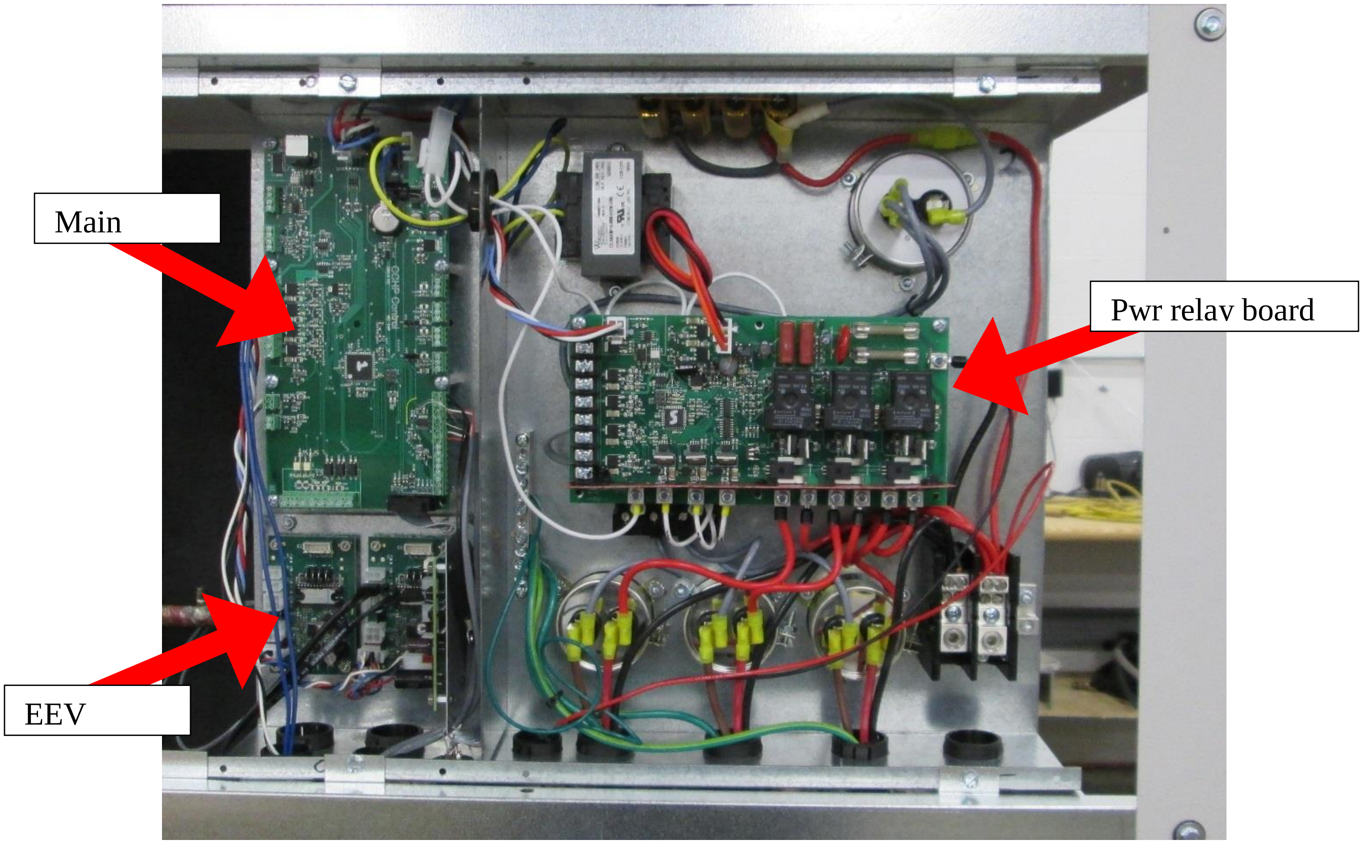



**Figure 10:** Control Box with Low voltage and High voltage PCBs


**Figure 11:** User interface for field technicians

A large portion of development went into testing and verifying the newly designed compressor control PCB and software. Initial challenges with start-up of the three compressors were overcome. We were successful in correcting the booster compressor motor power factor to greater than 0.92. Without the power factor correction, the power factor was less than 0.60 because the motors run at very low loads during the boosted compression cycle.

To control the system, a laptop Labview program was developed to set the various functions and to display the unit's internal operating data. See Figure 12 below.

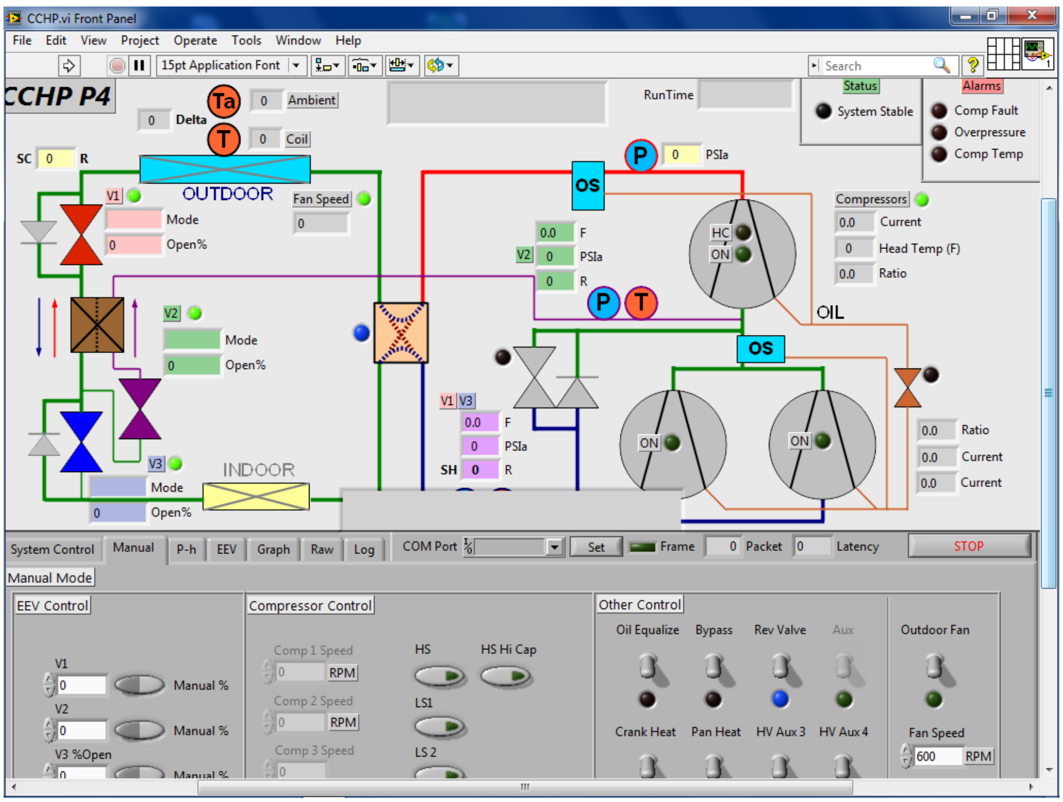



**Figure 12:** Final Labview control program (screen shot)

In the first half of 2014, ORNL completed Calorimeter Testing of a 3 Ton variable speed compressor supplied by Unico. The purpose of the testing was to validate compressor performance *beyond the existing compressor map data provided by the manufacturer*.

The results showed that previous ORNL simulations were a little conservative at low temperatures. It was estimated that volumetric efficiencies were around 70% at -13°F; however, the calorimeter testing concluded that volumetric efficiencies were greater than 90% at -13°F. Conversely, isentropic efficiency were slightly less than estimated. This new information prompted ORNL to modify the simulation logic accordingly which provided for more accurate predictions and better capacities at all temperatures as seen in Table 5 below.

**Table 5:** Modeling results comparison after calorimeter testing

| | | Modeling Results | | |
|---|---|---|---|---|
| | | Outside temp, °F | Btu/hr. | COP |
| Initial simulation results | (3) ZPS51 compressors | 47 | 59,590 | 4.0 |
| | | 17 | 54,054 | 3.01 |
| | | -13 | 44,407 | 2.11 |
| Simulation results after calorimeter testing information is known | (3) ZPS51 compressors | 47 | 60,516 | 3.85 |
| | | 17 | **59,492** | 2.87 |
| | | -13 | **58,000** | **2.3** |

The second budget period also saw continued modeling of additional compressors and configurations. Oak Ridge National Labs performed the modeling using their simulation software along with updated performance map data from calorimeter testing.

Since the low temperature modeling coefficients were updated and in conjunction with promising P2 test data, we asked ORNL to model several different compressor configurations. Based on ORNL simulation data, we chose to evaluate the next smaller size compressor in the Ultratech family for the high stage compressor (ZPS40) and three tandem sizes for the low stage compressor (ZPT108, ZPT114, and ZPT122). The tandem compressors were single speed compressors and provided larger displacement volume.

We also chose to evaluate both the ZPS51 and ZPS40 as the high stage, also called the main compressor, while in series with the ZPT122 tandem compressors.

In an effort to optimize the CCHP compressors for the P4 prototypes, we selected the ZPT122 as the low stage/booster compressor because it gave us better capacity and better COP at 17$^{o}$F and -13$^{o}$F, as seen in Table 6 below.

**Table 6: ORNL modeling results for additional compressor scenarios**

| COMPRESSOR SCENARIOS | Capacity at 47 F | COP at 47 F | Capacity at 17 F ZPS(H) | COP at 17 F ZPS(H) | Capacity at (-13 F) | COP at (-13 F) |
|---|---|---|---|---|---|---|
| ZPS51 (H) + ZPT122 (L) | 60516 | 3.85 | 53903 | 2.94 | 64936 | 2.23 |
| ZPS51 (H) + ZPT114 (L) | 60516 | 3.85 | 50897 | 2.96 | 61781 | 2.26 |
| ZPS40 (H) + ZPT122 (L) | 50873 | 4.18 | 51906 | 2.98 | 62356 | 2.21 |
| ZPS40 (H) + ZPT114 (L) | 50873 | 4.18 | 48975 | 3.02 | 58953 | 2.26 |

Following the July design decision to go with option #2 (three compressors), we focused our efforts on P4 development and refining the design for outdoor usage. We were able to reduce the overall weight by nearly 50 lbs. and shortened the overall length by almost 20 inches. In addition to the compressor configuration changes, we designed a fan assembly as opposed to P2 where we used a turnkey outdoor fan assembly from EBM Papst. The main driver to change the fan assembly was cost. Purchasing the individual components for the fan assembly was roughly half the cost of buying the turnkey assembly.

**Figure 13: P4 Prototype**

For P4, the coil circuiting pattern was changed to ensure a downward flow at all times, allowing gravity to act on the liquid-mass mixture and maintain even coil circuit temperatures. We added thermocouples to all the coil circuits in order to measure the temperature of each individual circuit during various stages of performance testing. We also attached a heater wire under the coil in order to maintain drainability. Preventing ice formation from the condensate that cascades off the coil during operation and defrost modes is crucial to system performance.

The Labview *auto function* allows the system to maintain a fixed evaporating temperature in cooling mode and a fixed condensing temperature in heating mode. The user can manually change these setpoints. In addition, the defrost function algorithm and logic were developed and implemented. The P4 unit has the ability to enter and leave the defrost mode automatically.

We also sent the exterior sheet metal parts to our powder coating supplier to be primered and powder painted. The coatings will allow the CCHP to endure outdoor conditions and be aesthetically pleasing as seen below in Figure 13.

Also in the second budget period, we performed a simple energy comparison analysis (see Table 7 below). From this analysis, we were able to determine that CCHP is favorable when compared to propane or fuel oil solutions, but not in natural gas markets.

**Table 7: Breakeven analysis of primary fuel/energy options**

| | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Natural gas | | Fuel Oil | | Propane | | | Furnace efficency | | COP, Break Even fuel only | | | | | Wisconsin 2013 | Maine 2013 | NYst 2013 | | | | | |
| | | | BTU/gal | | BTU/gal | | | | | | | | | | | | | | | | |
| | | | 140,000 | | 91,502 | | 80% | | Electricity Cost | | | | | | | | | | | | |
| | | Eenrgy cost | | Energy cost | | | | cents/kw | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | |
| $/1000cu-ft | $/Mbtu | $/Gal | $/btu | $/gal | $/btu | $/Kw-hr | $/Kw-hr | $/kw-hr | 0.08 | 0.09 | 0.1 | 0.11 | 0.12 | 0.13 | 0.14 | 0.15 | 0.16 | 0.17 | 0.18 | 0.19 | |
| 4.08 | 4 | | | | | 0.014 | 0.017 | | 4.7 | 5.3 | 5.9 | 6.4 | 7.0 | 7.6 | 8.2 | 8.8 | 9.4 | 10.0 | 10.6 | 11.1 | |
| 5.1 | 5 | | | | | 0.017 | 0.021 | | 3.8 | 4.2 | 4.7 | 5.2 | 5.6 | 6.1 | 6.6 | 7.0 | 7.5 | 8.0 | 8.4 | 8.9 | |
| 6.12 | 6 | | | | | 0.020 | 0.026 | | 3.1 | 3.5 | 3.9 | 4.3 | 4.7 | 5.1 | 5.5 | 5.9 | 6.3 | 6.6 | 7.0 | 7.4 | |
| 7.14 | 7 | | | | | 0.024 | 0.030 | | 2.7 | 3.0 | 3.3 | 3.7 | 4.0 | 4.4 | 4.7 | 5.0 | 5.4 | 5.7 | 6.0 | 6.4 | |
| 8.16 | 8 | | | | | 0.027 | 0.034 | | 2.3 | 2.6 | 2.9 | 3.2 | 3.5 | 3.8 | 4.1 | 4.4 | 4.7 | 5.0 | 5.3 | 5.6 | |
| 9.18 | 9 | | | | | 0.031 | 0.038 | | 2.1 | 2.3 | 2.6 | 2.9 | 3.1 | 3.4 | 3.6 | 3.9 | 4.2 | 4.4 | 4.7 | 4.9 | |
| 10.2 | 10 | | | | | 0.034 | 0.043 | | 1.9 | 2.1 | 2.3 | 2.6 | 2.8 | 3.0 | 3.3 | 3.5 | 3.8 | 4.0 | 4.2 | 4.5 | |
| 11.22 | 11 | | | | | 0.038 | 0.047 | | 1.7 | 1.9 | 2.1 | 2.3 | 2.6 | 2.8 | 3.0 | 3.2 | 3.4 | 3.6 | 3.8 | 4.0 | |
| 12.24 | 12 | | | | | 0.041 | 0.051 | | 1.6 | 1.8 | 2.0 | 2.1 | 2.3 | 2.5 | 2.7 | 2.9 | 3.1 | 3.3 | 3.5 | 3.7 | |
| 13.26 | 13 | | | | | 0.044 | 0.055 | | 1.4 | 1.6 | 1.8 | 2.0 | 2.2 | 2.3 | 2.5 | 2.7 | 2.9 | 3.1 | 3.2 | 3.4 | |
| 14.28 | 14 | 2 | 1.42857E-05 | | | 0.049 | 0.061 | | 1.3 | 1.5 | 1.6 | 1.8 | 2.0 | 2.1 | 2.3 | 2.5 | 2.6 | 2.8 | 3.0 | 3.1 | |
| 15.3 | 15 | 2.1 | 0.000015 | | | 0.051 | 0.064 | | 1.3 | 1.4 | 1.6 | 1.7 | 1.9 | 2.0 | 2.2 | 2.3 | 2.5 | 2.7 | 2.8 | 3.0 | |
| 16.32 | 16 | 2.2 | 1.57143E-05 | | | 0.054 | 0.067 | | 1.2 | 1.3 | 1.5 | 1.6 | 1.8 | 1.9 | 2.1 | 2.2 | 2.4 | 2.5 | 2.7 | 2.8 | |
| | | 2.3 | 1.64286E-05 | | | 0.056 | 0.070 | | 1.1 | 1.3 | 1.4 | 1.6 | 1.7 | 1.9 | 2.0 | 2.1 | 2.3 | 2.4 | 2.6 | 2.7 | |
| | | 2.4 | 1.71429E-05 | | | 0.058 | 0.073 | | 1.1 | 1.2 | 1.4 | 1.5 | 1.6 | 1.8 | 1.9 | 2.1 | 2.2 | 2.3 | 2.5 | 2.6 | |
| | | 2.5 | 1.78571E-05 | | | 0.061 | 0.076 | | 1.1 | 1.2 | 1.3 | 1.4 | 1.6 | 1.7 | 1.8 | 2.0 | 2.1 | 2.2 | 2.4 | 2.5 | |
| | | 2.6 | 1.85714E-05 | | | 0.063 | 0.079 | | 1.0 | 1.1 | 1.3 | 1.4 | 1.5 | 1.6 | 1.8 | 1.9 | 2.0 | 2.1 | 2.3 | 2.4 | |
| | | 2.7 | 1.92857E-05 | | | 0.066 | 0.082 | | 1.0 | 1.1 | 1.2 | 1.3 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | 2.1 | 2.2 | 2.3 | |
| | | 2.8 | 0.00002 | | | 0.068 | 0.085 | | 0.9 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.8 | 1.9 | 2.0 | 2.1 | 2.2 | |
| | | 2.9 | 2.07143E-05 | | | 0.071 | 0.088 | | 0.9 | 1.0 | 1.1 | 1.2 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | 2.0 | 2.2 | Range of CCHP test results to-date |
| | | 3 | 2.14286E-05 | | | 0.073 | 0.091 | | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.8 | 1.9 | 2.0 | 2.1 | |
| | | 3.1 | 2.21429E-05 | | | 0.076 | 0.094 | | 0.8 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | 2.0 | |
| | | 3.2 | 2.28571E-05 | | | 0.078 | 0.097 | | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | |
| | | 3.3 | 2.35714E-05 | | | 0.080 | 0.101 | | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | |
| | | 3.4 | 2.42857E-05 | | | 0.083 | 0.104 | | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | |
| | | 3.5 | 0.000025 | | | 0.085 | 0.107 | | 0.8 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | |
| | | 3.6 | 2.57143E-05 | | | 0.088 | 0.110 | | 0.7 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.6 | 1.7 | |
| | | 3.7 | 2.64286E-05 | | | 0.090 | 0.113 | | 0.7 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | |
| | | 3.8 | 2.71429E-05 | | | 0.093 | 0.116 | | 0.7 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.6 | |
| | | 3.9 | 2.78571E-05 | 2.5 | 2.7322E-05 | 0.093 | 0.117 | | 0.7 | 0.8 | 0.9 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.5 | 1.6 | |
| | | 4 | 2.85714E-05 | 2.6 | 2.8415E-05 | 0.097 | 0.121 | | 0.7 | 0.7 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | |
| | | 4.1 | 2.92857E-05 | 2.7 | 2.9508E-05 | 0.101 | 0.126 | | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.4 | 1.5 | |
| | | 4.2 | 0.00003 | 2.8 | 3.06E-05 | 0.104 | 0.131 | | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.1 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | |
| | | | | 2.9 | 3.1693E-05 | 0.108 | 0.135 | | 0.6 | 0.7 | 0.7 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.3 | 1.3 | 1.4 | |
| | | | | 3 | 3.2786E-05 | 0.112 | 0.140 | | 0.6 | 0.6 | 0.7 | 0.8 | 0.9 | 0.9 | 1.0 | 1.1 | 1.1 | 1.2 | 1.3 | 1.4 | |
| | | | | 3.1 | 3.3879E-05 | 0.116 | 0.144 | | 0.6 | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.2 | 1.3 | |
| | | | | 3.2 | 3.4972E-05 | 0.119 | 0.149 | | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 | 0.9 | 0.9 | 1.0 | 1.1 | 1.1 | 1.2 | 1.3 | |
| | | | | 3.3 | 3.6065E-05 | 0.123 | 0.154 | | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.2 | |
| | | | | 3.4 | 3.7158E-05 | 0.127 | 0.158 | | 0.5 | 0.6 | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 0.9 | 1.0 | 1.1 | 1.1 | 1.2 | |
| | | | | 3.5 | 3.8251E-05 | 0.131 | 0.163 | | 0.5 | 0.6 | 0.6 | 0.7 | 0.7 | 0.8 | 0.9 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | |
| | | | | 3.6 | 3.9343E-05 | 0.134 | 0.168 | | 0.5 | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.1 | |
| | | | | 3.7 | 4.0436E-05 | 0.138 | 0.172 | | 0.5 | 0.5 | 0.6 | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 0.9 | 1.0 | 1.0 | 1.1 | |
| | | | | 3.8 | 4.1529E-05 | 0.142 | 0.177 | | 0.5 | 0.5 | 0.6 | 0.6 | 0.7 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | |

We surveyed our internal sales group on the marketability of the CCHP in its current form (prototype P4). We determined that the challenges are size, weight, cost, and servicer complexity.

Toward the end of 2014, we came to the conclusion that the goals of the SOPO were not achievable with existing compressor technologies and modified the Statement of Project Objectives (SOPO) accordingly to support the project goals of performance, efficiency, and cost.

Although we were pleased and encouraged by the “best in industry” heat pump performance and COP testing results, the following factors contributed to our conclusions: extensive R&D required, high initial unit cost, product complexity, schedule duration, and high warranty exposure risk.

In order to facilitate a single compressor solution and based on the reasons noted above Unico approached budget period 3 with a plan to develop a CCHP compressor specification and find a compressor manufacturer who had the desire and capability to design a compressor around the CCHP needs like a greater volumetric efficiency and excellent performance at sub-zero temperatures.

*Budget Period 3*

In the modified SOPO for 2015, we reported that we would begin baseline testing an existing heat pump designed to operate in cold climates. We started the testing and completed all of the cooling mode and some of the heating mode tests. Testing was put on hold at the end of January 2016 because we had to move the lab to our manufacturing facility.

CCHP Lab development continued throughout the project. The air sampling trees were designed and built in 2015. The following efforts will allow us to measure the entering/leaving enthalpy at the indoor coil more accurately. In addition, these advancements will improve overall capacity measurements, increase accuracy of input measurements, and increase efficiency of data collection.

Psychrometer boxes were fabricated and constructed as seen below in Figure 14. The associated instrumentation for data gathering was also installed and tested to support test data acquisition. Integration of the air sampling trees and psychrometers into the DAQ system was completed and tested. Additionally, DAQ programming was improved by implementing automated data logging.

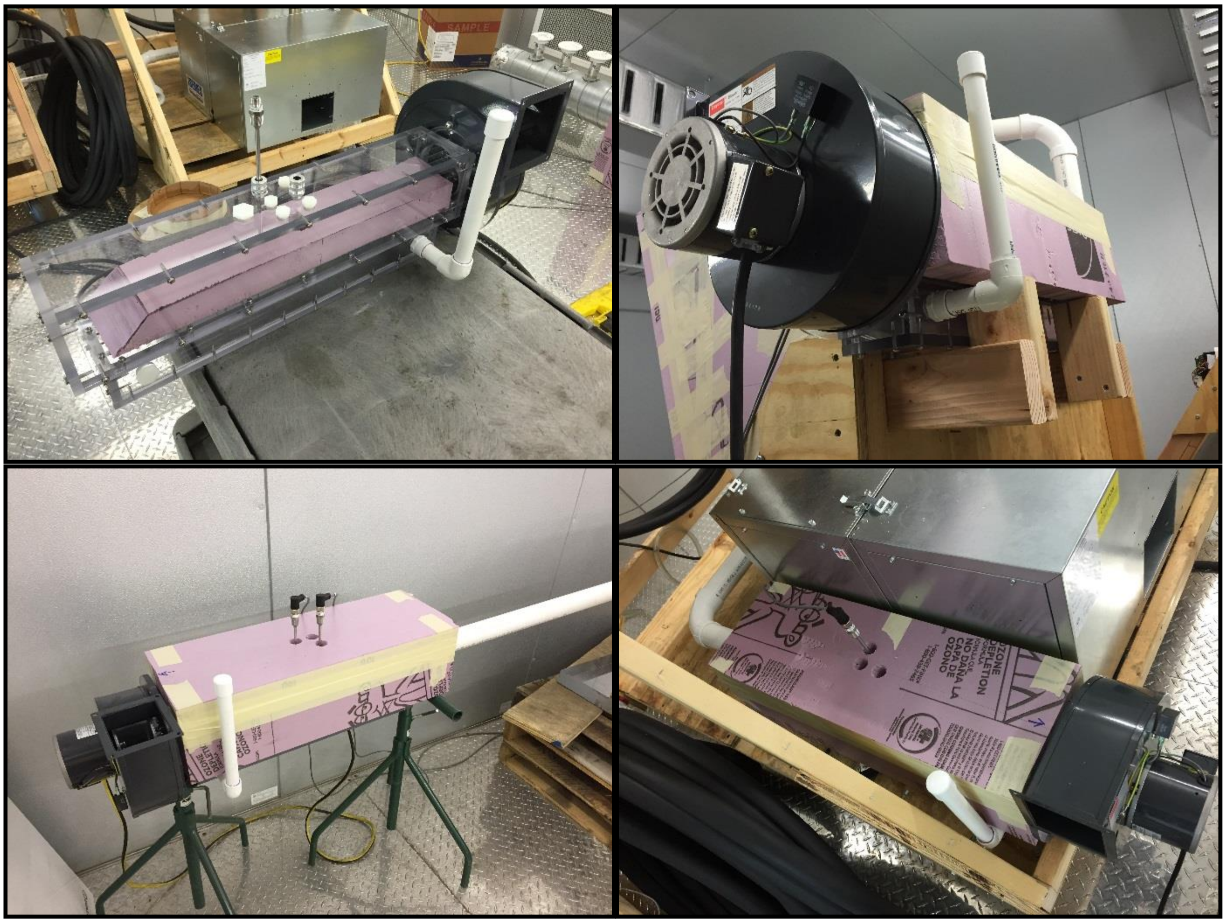


**Figure 14: Psychrometer Boxes**

Construction and calibration of precision pressure measurement boxes was also completed. These will be used in conjunction with a nozzle box method of measuring airflow. We integrated AHRI 210/240 provisions and ASHRAE fundamentals equations into the DAQ program; these steps will improve data accuracy.

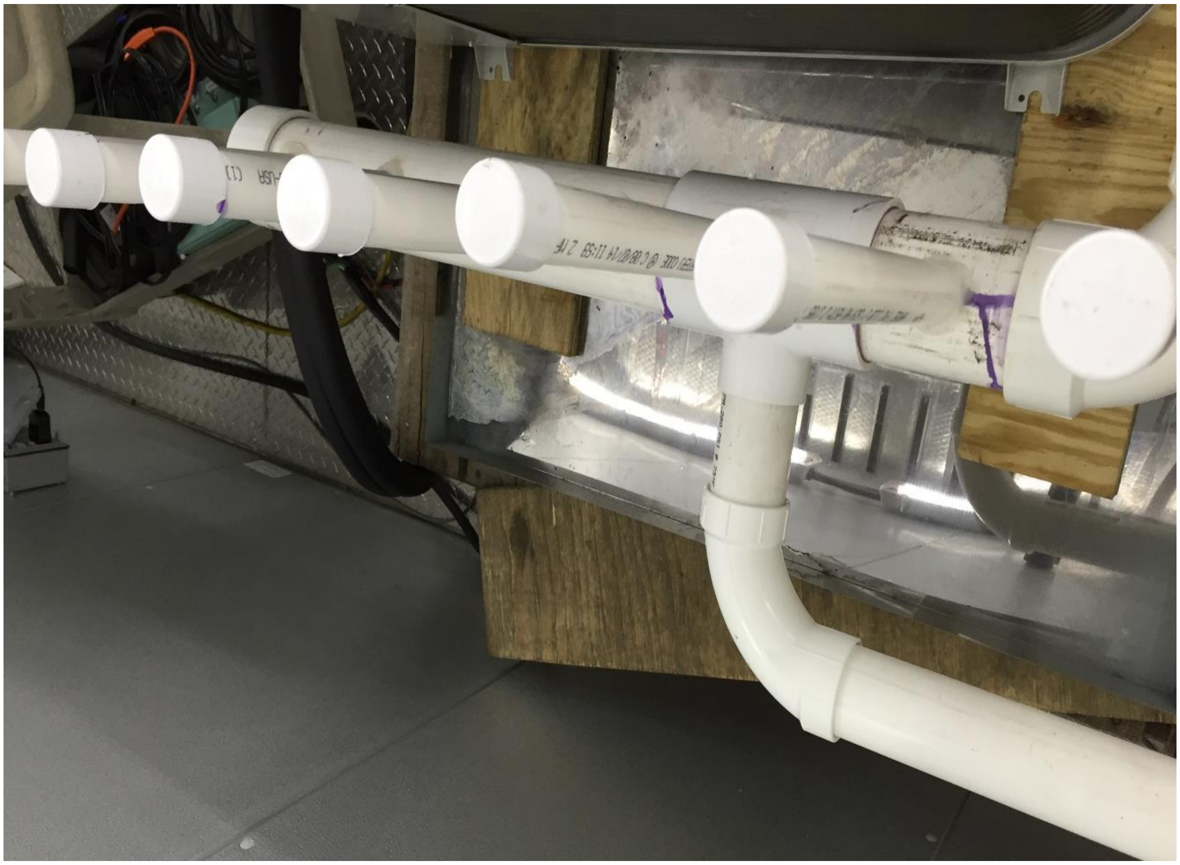


**Figure 15: Air sampling tree**

Another focus was to improve testing accuracy by including a secondary energy check and calculating an energy balance. Since the first tests were cooling mode, we were able to achieve an energy balance of less than 5% by performing a condensate balance check. This is not appropriate for heating, so we added refrigerant flow meters to perform a refrigerant-side energy balance. The flow meters were found to be out of calibration so we returned them to the manufacturer for repairs. They were returned to us in January but not in time for the few heating tests we performed. We will use them when the lab is operational again.

We made a lot of progress in setting up the capability to test at cold temperatures and we produced several working prototypes that provided excellent capacity (meeting the goals easily) but did not meet the efficiency goals (although still reasonable). We had a lot of lessons learned along the way. The most important were the significant risks mentioned above concerning oil management, unit weight, and expected retail costs to the end-user; which did not justify commercialization of our original CCHP design. At that time, we decided to pursue other designs of similar concept in addition to modifying the SOPO with the intent to seek an extension of time to the contract

The most promising alternate CCHP design utilizes a concept being developed by Oak Ridge National Laboratory (ORNL) using vapor injection with dual compressors in parallel. They performed extensive lab testing and were just beginning limited field trials in a couple of locations. Performance was good. The next step would be to develop a unit designed for manufacturability and perform expanded long-term field trials to verify reliability. We are intrigued by this technology and believe it holds promise but we felt the time to market was much longer than the scope that the CCHP project allowed.

We also explored the possibility of extending the CCHP to an existing product that we recently introduced. During, and even before, the start of the CCHP project, we had a parallel development project for a variable capacity heat pump developed by Argoclima in Italy, a Unico Inc. product line called “iSeries”. The focus of the iSeries project has been to develop a variable speed heat pump that can match to our small duct high velocity (SDHV) indoor units. This product was introduced to the market in June 2014.

Although the iSeries was not specifically a “cold climate” heat pump, Argoclima designed it to work at very low temperatures (see Table 8). At the lowest temperatures, the units provide heat at roughly 50% of nominal rated capacities. Thus, the iSeries could be thought of as a “colder” climate heat pump. Unico has been marketing the iSeries in the northeastern states (MA, ME, VT, NH, NY, RI, CT) with good success.

**Table 8: iSeries Heat Pump Performance (matched to non-ducted indoor units)**

| | **Capacity** | | | | **COP** | | | |
|---|---|---|---|---|---|---|---|---|
| **Model** | **G50** | **G65** | **G80** | **G110** | **G50** | **G65** | **G80** | **G110** |
| **+7°C (44.6°F)** | 20472 | 29684 | 38214 | 46062 | 6.7 | 4.8 | 5.0 | 5.5 |
| **+2°C (35°F)** | - | - | - | - | 3.4 | - | 4.3 | - |
| **-7°C (19.4°F)** | 13307 | 20131 | 23202 | 28320 | 2.9 | 3.1 | 2.5 | 2.4 |
| **-10°C (14°F)** | 12624 | 18084 | 22178 | 25590 | - | - | - | - |
| **-22°C (-7.6°F)** | 11260 | 14672 | 16719 | 20472 | - | - | - | - |
| **-32°C (-25.6°F)** | - | - | - | - | 0.98 | 0.98 | 0.98 | 0.98 |
| **SCOP** | - | - | - | - | 4.01 | 4.01 | 4.07 | 4.12 |
| **Note:** This data applies to outdoor units matched to wall-mount indoor units, and has not been certified by AHRI. Source: Argoclima product literature. | | | | | | | | |

We approached Argoclima about the possibility of adding vapor injection to the product as an attempt to increase the cold temperature capacity. They already performed some work on this, but were not currently interested in commercializing the product.

In anticipation of positioning the existing iSeries as an effective cold climate heat pump, we needed to test it in our chambers. This would give us a baseline for further development against our other CCHP prototypes and provide performance data for use in commercializing the iSeries today.

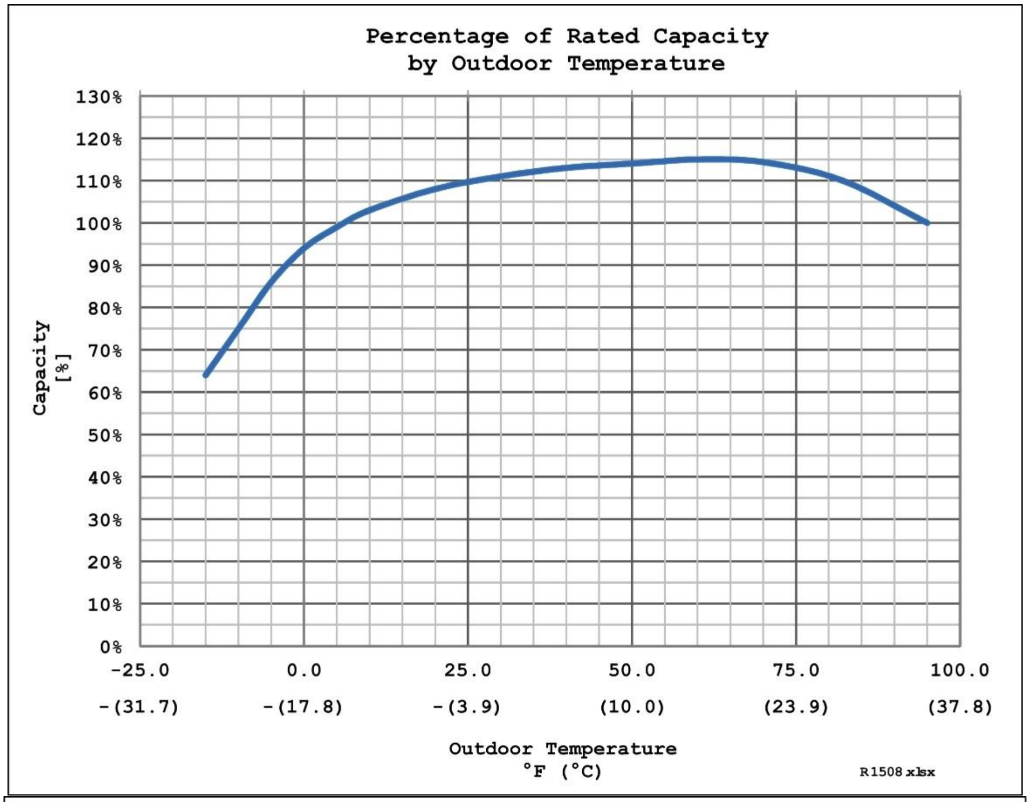
Argoclima product literature and initial testing showed that the iSeries heating capacity decreases to nearly half of the rated capacity at low temperatures, so we added the ability of the unit to switchover to boiler heat. This is a unique design feature unmatched by any other mini-/multi-split product on the market and was driven by the needs of our customers, mostly in the New England region of the U.S.  The introduction of the iSeries brings heat pump technology to a region of the country that traditionally uses gas or oil for heating, even if the heat pump is unable to meet the full demand of the home.

Our first priority was to be sure that the iSeries satisfy DOE and AHRI requirements for minimum efficiency and capacity. This required test data for both heating and cooling. We already had data from Intertek for the SEER and HSPF testing which was used to establish our ratings. This testing was only for a few conditions but we needed data for other conditions.

Since our focus is cold climates, we first performed functional testing at very low temperatures to be sure that the system program was working properly. This testing allowed us to diagnose several problems during defrost mode and then implement corrective actions. One problem was associated with the grill protecting the fins (figure 16). We discovered that the frost was accumulating to such an extent that it was attaching to the grill and forming a block of ice that could not be removed. Extra coil sensors were added to force the defrost mode to initiate sooner and to completely clear the coil. These changes also reduced the length of the defrost cycle while improving heating capacity.

**Figure 16: Defrost with fin protector screen (left) and without (right)**

The second step was to perform testing (capacity and efficiency) at a large range of conditions. We tested the unit in both cooling and heating. The cooling tests were conducted at normal cooling conditions and at very cold temperatures. We published cold temperature performance data in a technical bulletin, *Unico Technote 130*, which are shown in figure 17.



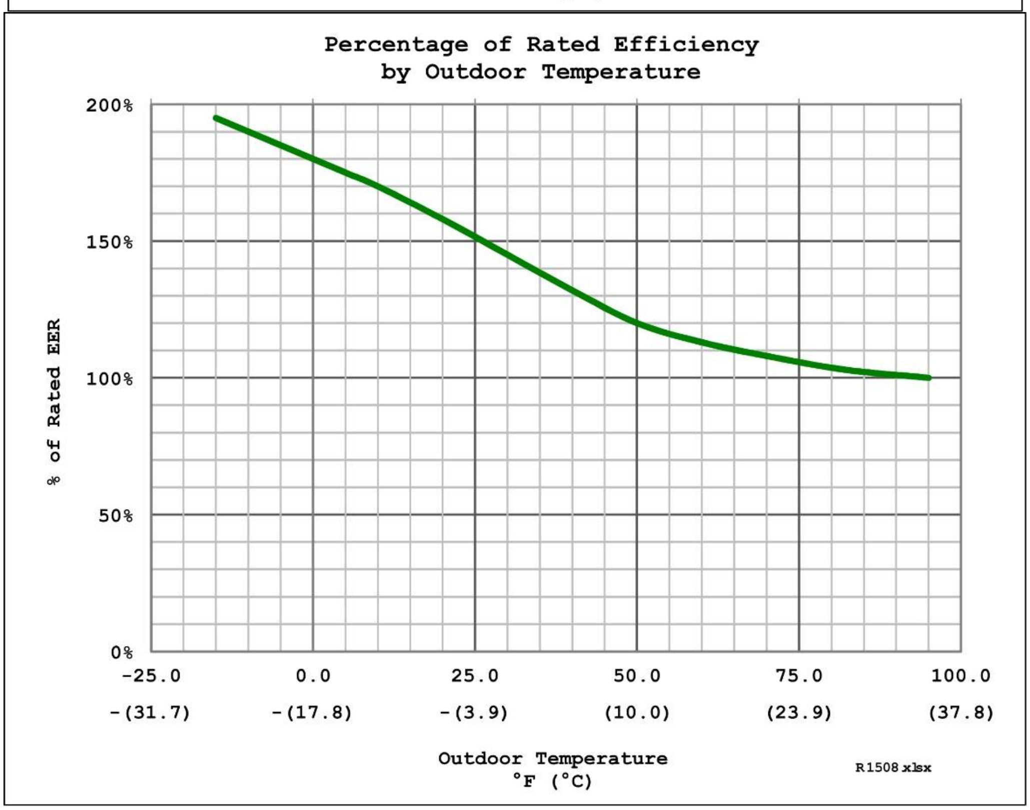




**Figure 17: iSeries Cold temperature cooling performance**

This information from actual testing results allowed our sales team to market the iSeries as a viable heat pump in very cold climates, highlighting one of the real benefits of the CCHP project.

The baseline cooling performance was conducted in the last quarter of 2015; ending at the end of January 2016 when all testing was put on hold for the chamber move. The baseline testing covered the full range of operation under all load conditions. The plan is to continue with the heating tests once we are able to resume testing. The test data so far will be used to map the cooling performance for building load software which will add to our ability to commercialize a cold climate heat pump and compare performance on energy and economic bases versus traditional heat pumps and fossil fuels.

We performed a large number of cooling tests: (52) for the 3-ton unit and (10) for the 2-ton unit. The 3-ton was exhaustively tested in order to establish coefficients that could be used to predict system performance based on indoor and outdoor operating conditions. The reduced set of data collected for the 2-ton unit showed that the shape of the capacity and efficiency curves was different for the 2- and 3-ton units and that more data was needed to establish accurate coefficients for the 2-ton unit. This data and the resulting coefficients can be used to compare the iSeries with SDHV versus traditional HVAC systems and further increase market penetration of the product.

The results from this testing are summarized below in figures 18 and 19. The first graph, figure 18, shows the capacity of the system. There are five distinct vertical clusters of test points at the five outdoor temperature test conditions. The distribution of points is a function of the data collected. Compressor speed, indoor airflow, indoor dry bulb and indoor and outdoor relative humidity were all varied. It can be seen that the iSeries has a capacity turndown ratio of between 4.7 at 62°F and 3.3 at 110°F. Capacities were verified using a condensate balance.

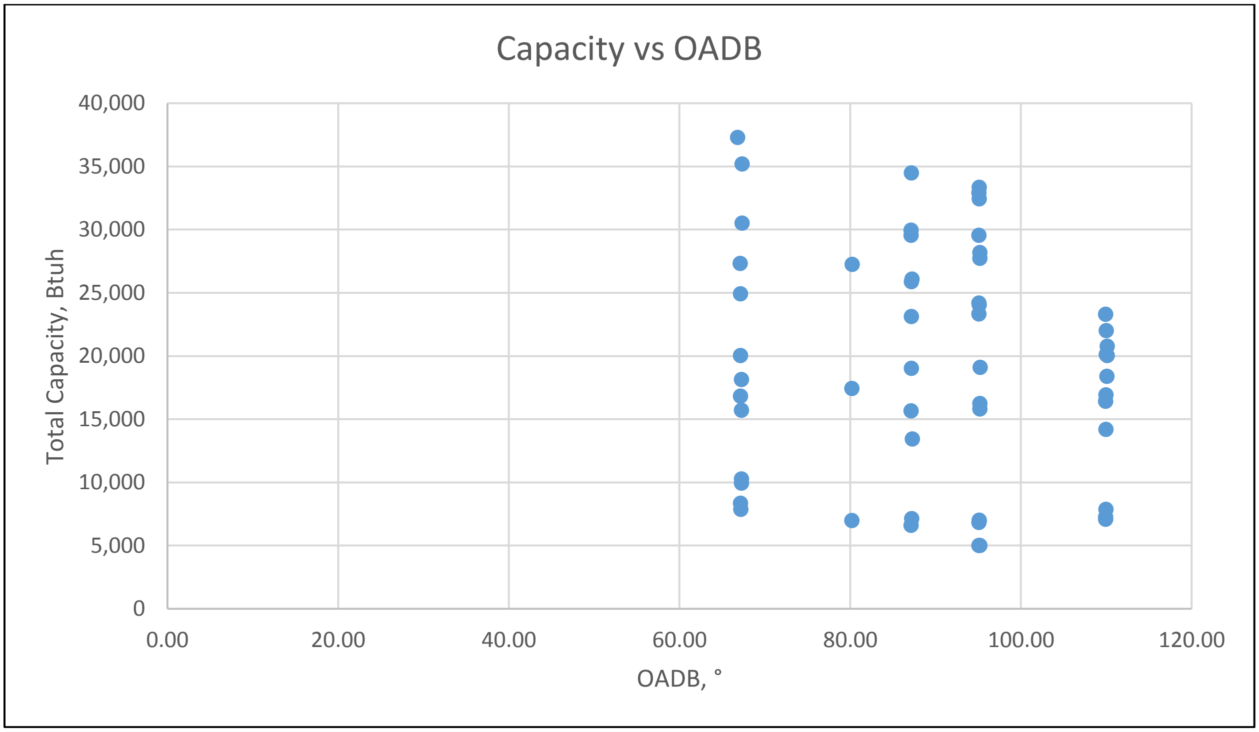



**Figure 18. IS36G110 + M3036 Cooling Performance**

In the next graph, figure 19, shows the efficiency of the system. As expected, the efficiency of the system in cooling mode increases as the outdoor temperature decreases. The COP varies by 0.5 at 110°F between the minimum and maximum and by 5 points at 62°F. This shows the large range of tunability of the system, and highlights the advantage of variable speed compressors, where target efficiency can be achieved as much through software controls on operating parameters as do the materials of construction.

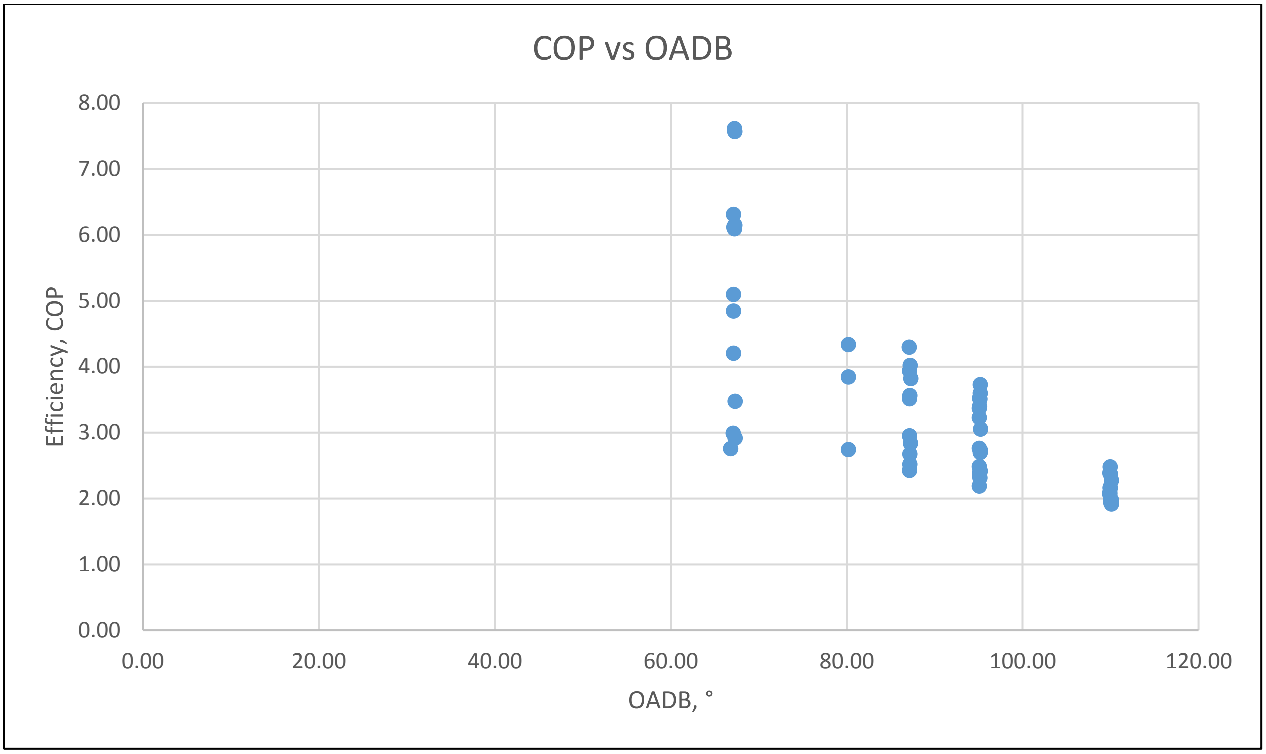



**Figure 19. IS36G110 + M3036 Cooling Performance**

Due to our refrigerant flowmeter fouling, we were not able to achieve a heat balance for our heating tests. Initial tests repeating AHRI certification data showed that our capacity and efficiency lined up with third-party test labs. We were unable to conduct extensive testing prior to our lab relocation.

As mentioned above, we were forced to put all testing on hold at the end of January 2016 in order to dismantle the chambers and move them to our factory location in Arnold, Missouri. As of March 17, 2016, our main office and lab in St. Louis are located at 1120 Intagliata Drive, Arnold MO. Testing is expected to resume in August. In the meantime, we are putting our effort into the rebuilding the chambers.

Thus far, we have built the lab walls and doors, installed main power (2000 kVA), installed lighting, re-assembled the control booth, installed network wiring, and erected the chambers. By end of the second quarter, we expect to have all the conditioning equipment and instrumentation installed. The month of July is reserved for lab calibration. The functionality of the lab remains the same although the footprint has been reduced from about 8000 sq. ft. to roughly 4500 sq. ft.

The most recent photos of the lab (March 2016) are as follows:

Figure 20
Entrance to lab

Figure 21
Entrance to middle chamber (#2)

Figure 22
All three chambers starting from right (#1, #2, and #3). Chambers #1 and #3 are low temperature outdoor chambers. Chamber #2 is indoor-only. Chamber #1 can also be indoor.

Figure 23
Backside of chambers.
Electrical power installation to right and left and spiral ducting installed at the ceiling.
Ready for air handlers (lab conditioning equipment).

Figure 24:
Inside of chamber 3.
Refrigeration equipment installed.

Figure 25
Control booth with Heat Pump installed.

Figure 26
Outside of Lab.
Racks and equipment: outdoor heat pumps and condensing units to condition the chambers and lab work space.

Relocating the lab gave us the rare opportunity to act on some of the lessons learned when we first designed and installed the chambers. One of the lessons is to add more cooling capacity for the low temperature chambers to maintain water temperature stability. We did this in two ways. First we added additional chillers and second we fully insulated the surrounding (exterior) walls of the lab space. To provide more functionality, we also redesigned the chilled water piping to allow the extra chillers to be switched between the lab chambers. With three chambers operational, the switching allows us to avoid duplication of chiller equipment. We can shift the capacity from one chamber to the next.

## Conclusion

Sometimes product development takes unusual twists and turns. This project had its share. When we first proposed our design, we intended to use the technology developed by Hallowell International. We were unable to reach a licensing agreement with the owner of the patent rights. Therefore, we decided to pursue a slightly different design developed at Purdue, for which we obtained the necessary licensing. Unfortunately we discovered that this design was not fully developed and we were very concerned about its reliability which forced us to look at different technologies, all of which were farther from commercialization than we wanted.