ORNL/TM-2021/2029

# Cold Climate Heat Pump Evaluation


Jeffrey Munk
Tony Gehl

December 2022

OAK RIDGE NATIONAL LABORATORY
MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

**ORNL/TM-2021/2029**

Energy and Transportation Science Division

# COLD CLIMATE HEAT PUMP EVALUATION

Jeffrey Munk

Tony Gehl

Date Published: December 2022

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37831-6283
managed by
UT-Battelle, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

## CONTENTS

## LIST OF FIGURES

# LIST OF TABLES

## ABBREVIATIONS

| | |
|---|---|
| CFR | Code of Federal Regulations |
| ESP | external static pressure |
| ID | indoor |
| OD | outdoor |
| TXV | thermostatic expansion valve |
| UUT | unit under test |
| w.c. | water column |

**ACKNOWLEDGEMENTS**

Funding for this research was provided by the US Department of Energy, Office of Energy Efficiency and Renewable Energy. The authors would like to thank John Cymbalsky, manager for the Appliance and Equipment Standards Program, and technology managers, Ashley Armstrong and Tony Bouza, for their support of this work.

# 1. INTRODUCTION

The use of heat pumps in cold and very cold climates increased by 63% between 2009 and 2015 [1, 2]. This can be attributed to development of heat pumps with higher heating capacity at lower temperatures and increased efficiency. A primary technology driving this trend is the use of variable-speed compressors, which come with an increase in the level of controls in the equipment. Most units with variable-speed compressors also include electronic expansion valves and variable-speed indoor (ID) and outdoor (OD) fans. With the increased complexity of these systems, developing a test procedure that is representative of typical performance and not too burdensome on manufacturers presents a difficult challenge.

This study focuses on collecting laboratory data to help answer several key questions:

- Is load-based testing feasible and does it generate results that differ from the standard rating procedure?
- Does the equipment operate similarly in the standard rating test mode as it does under native controls?
- Does the current optional test point ($H4_2$) adequately capture the low-temperature performance of existing heat pumps?
- Does moisture uniformity in the OD chamber affect the performance of units with multi-faced OD coils?

# 2. LABORATORY SETUP

All equipment was tested in environmental chambers (ID and OD). The test setups followed the requirements specified in ASHRAE 37-2009 [3]. The primary capacity measurement for all systems was the ID air enthalpy method. The secondary test method was either the OD air enthalpy method or the refrigerant enthalpy method depending on the location of the cooling mode expansion valve. All the systems tested with the exception of the centrally ducted units had the cooling mode expansion valves in the OD units. This location results in inadequate subcooling in the liquid line required for the refrigerant enthalpy method, so the OD air enthalpy method was used.

Appendix A contains pictures to serve as examples of the equipment setups used during testing, and Table 1 lists the instrumentation used for measurements.

**Table 1. Instrumentation information**

| Measurement | Sensor description | Manufacturer | Model |
|---|---|---|---|
| ID unit power<br>ID unit voltage<br>ID unit frequency | Power meter and current transformers | Yokogawa<br>LEM | WT310e<br>IT-200-S Ultrastab |
| OD unit power<br>OD unit voltage<br>OD unit frequency | Power meter and current transformers (200:5) | Yokogawa<br>GE | WT333e<br>750X133311 |
| ID inlet dry-bulb temperature (DB)<br>ID inlet wet-bulb temperature (WB)<br>ID outlet DB<br>ID outlet WB<br>OD inlet DB<br>OD inlet WB<br>OD outlet DB<br>OD outlet WB | Platinum resistance temperature detector (RTD) | Rosemount | 0078N01N00N060AB6 |
| Comp. suction pressure<br>Service valve vapor line pressure<br>ID unit vapor line pressure | Pressure transducer 0–5000 psi | GE Druck | PTX5072-TA-A2 |
| Comp. discharge pressure<br>Service valve liquid line pressure<br>ID unit liquid line pressure | Pressure transducer 0–1,00 psi | GE Druck | PTX5072-TA-A2 |
| ID unit external static pressure | Differential pressure transducer (±2 in. water column [w.c.]) | Foxboro | IDP10S-T22B01FA-MIL1 |
| ID before nozzle pressure | Differential pressure transducer (±5 in. w.c.) | Foxboro | IDP10S-T22B01FA-MIL1 |
| ID nozzle delta pressure | Differential pressure transducer (0–4 in. w.c.) | Foxboro | IDP10S-T22B01FA-MIL1 |
| Barometric pressure | Pressure transducer | Vaisala | PTB110 |
| OD unit external static pressure | Differential pressure transducer (±0.1 in. w.c.) | Setra | 26510R1WB2BT1E |
| OD nozzle delta pressure | Differential pressure transducer (0–5 in. w.c.) | Setra | 2391005WD1F2C109NN |
| OD before nozzle pressure | Pressure transducer (800–1,060 mbar) | Omega | PX2780-800NB5V |
| Refrigerant temperatures<br>OD inlet grid<br>OD outlet grid<br>ID inlet grid<br>ID outlet grid | Type T thermocouple | | Special limits of error |
| Refrigerant mass flow | Coriolis mass flow meter | Emerson | CMF025M314N2GAEZZZ |

## 3. TEST PLAN

The test plan includes nine systems divided into two major categories of system types: mini-split and central-ducted split systems. A list of the system identifiers, nominal cooling capacity, and system description is provided in Table 2.

**Table 2. Test unit information**

| | | Capacity (Btu/h) | | |
|---|---|---|---|---|
| Manufacturer identifier | System identifier | Cooling | Heating | Description[a] |
| A | 1 | 22,000 | 27,000 | ND, HW, MS |
| B | 2 | 33,000 | 38,000 | CM, MS |
| B | 3 | 18,000 | 22,000 | ND, HW, MS |
| C | 4 | 34,000 | 33,000 | CD, SS |
| D | 5 | 22,000 | 27,000 | ND, HW, MS |
| E | 6 | 22,000 | 25,200 | ND, HW, MS |
| E | 7a | N/R | N/R | ND, HW, MS |
| E | 7b | 14,500 | 18,000 | ND, HW, MS |
| F | 8 | 34,800 | 34,400 | CD, SS |

[a] ND = non-ducted, CD = central-ducted, CM = ceiling-mount, MS = mini-split, SS = split system, HW = high wall, N/R = not rated

The test plan includes a combination of standard rating tests and non-standard tests. Test names and conditions are listed in Table 3 for mini-split systems and Table 4 for central-ducted systems.

All standard rating tests were conducted according to Appendix M1 to subpart B of 10 CFR (Code of Federal Regulations) 430 unless otherwise noted.

For dynamic tests, denoted with a subscript $x$, the equipment was operated without any controls override active (i.e., not in test mode). For the A, B, and H1 test conditions, the environmental chambers were operated with fixed cooling or heating output. The fixed heating and cooling output levels were identified by first running the equipment in its ratings test mode for the corresponding test condition. For example, the $B_1$ test was run immediately before the $B_{1x}$ test, and the ID chamber cooling and heating output were fixed for the $B_{1x}$ test at the levels used during the $B_1$ test. This results in the unit under test (UUT) responding to the load in the chamber to maintain the desired temperature. In most cases, the UUT set point had to be adjusted to achieve an ID inlet condition close to the target test temperature. If the unit capacity during the dynamic test did not match the capacity of the rating test, the cooling or heating output of the ID chamber was adjusted to try and achieve a match. If a similar capacity could not be achieved without the unit cycling on and off, then data were collected with the unit operating at a capacity as close as possible to the ratings test while running continuously (not cycling). For the dynamic tests at the H4, H5, and H6 test conditions, the ID and OD chambers were operated to control the temperature and humidity to the target values. The heating set point of the UUT was set to the highest possible value to cause it to run at its maximum available capacity.

Dynamic tests were subject to the tolerances shown in Tables 5 and 6 for cooling and heating modes, respectively.

**Table 3. Mini-split test plan**

| Test name | ID inlet (°F) Dry-bulb | ID inlet (°F) Wet-bulb | OD inlet (°F) Dry-bulb | OD inlet (°F) Wet-bulb | Compressor speed | Notes | Mode |
|---|---|---|---|---|---|---|---|
| $A_{2(FA)}$ | 80.0 | 67.0 | 95.0 | N/A | Cooling full | Test mode[a] | Cooling |
| $A_2$ | 80.0 | 67.0 | 95.0 | N/A | Cooling full | Test mode | Cooling |
| $A_{2(CD)}$ | 80.0 | 67.0 | 95.0 | N/A | Cooling full | Test mode[b] | Cooling |
| $A_{2x}$ | 80.0 | 67.0 | 95.0 | N/A | Dynamic | | Cooling |
| $B_1$ | 80.0 | 67.0 | 82.0 | N/A | Cooling minimum | Test mode | Cooling |
| $B_{1x}$ | 80.0 | 67.0 | 82.0 | N/A | Dynamic | | Cooling |
| $H0_1$ | 70.0 | ≤60.0 | 62.0 | 56.5 | Heating minimum | Test mode | Heating |
| $H1_N$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating full | Test mode[c] | Heating |
| $H1_{N(CD)}$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating full | Test mode[b,c] | Heating |
| $H1_2$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating full | Test mode[d,e] | Heating |
| $H1_{2x}$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Dynamic | | Heating |
| $H1_1$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating minimum | Test mode | Heating |
| $H1_{1x}$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Dynamic | | Heating |
| $H2_v$ | 70.0 | ≤60.0 | 35.0 | 33.0 | Heating intermediate | Test mode | Heating |
| $E_{Std}$ | 70.0 | ≤60.0 | 34.0 | 32.0 | Heating intermediate | Test mode | Heating |
| $E_{Marine}$ | 70.0 | ≤60.0 | 34.0 | 33.0 | Heating intermediate | Test mode | Heating |
| $H3_2$ | 70.0 | ≤60.0 | 17.0 | 15.0 | Heating full | Test mode | Heating |
| $H4_2$ | 70.0 | ≤60.0 | 5.0 | 4.0 | Heating full | Test mode | Heating |
| $H4_2$ | 70.0 | ≤60.0 | 5.0 | ≤3.0 | Heating full | Test mode | Heating |
| $H4_x$ | 70.0 | ≤60.0 | 5.0 | ≤3.0 | Dynamic | [f] | Heating |
| $H5_x$ | 70.0 | ≤60.0 | -5.0 | ≤-5.0 | Dynamic | [f] | Heating |
| $H6_x$ | 70.0 | ≤60.0 | -15.0 | ≤-15.0 | Dynamic | [f] | Heating |

[a] Free air test is run with neither the ID nor OD airflow apparatus connected to the unit.
[b] Closed duct test is run with both the ID and OD airflow apparatus connected to the unit.
[c] Maximum speed that the system controls would operate the compressor in normal operation at 47°F ambient.
[d] Maximum speed that the system controls would operate the compressor in normal operation at 17°F ambient.
[e] $H1_2$ test is not needed if the $H1_N$ test uses the same compressor speed.
[f] Run with heating set point set to maximum value and chambers controlling to target conditions.

**Table 4. Central-ducted test plan**

| Test name | ID inlet (°F) | | OD inlet (°F) | | Compressor speed | Notes | Mode |
|---|---|---|---|---|---|---|---|
| | Dry-bulb | Wet-bulb | Dry-bulb | Wet-bulb | | | |
| $A_2$ | 80.0 | 67.0 | 95.0 | N/A | Cooling full | Test mode | Cooling |
| $A_{2x}$ | 80.0 | 67.0 | 95.0 | N/A | Dynamic | | |
| $B_1$ | 80.0 | 67.0 | 82.0 | N/A | Cooling minimum | Test mode | |
| $B_{1x}$ | 80.0 | 67.0 | 82.0 | N/A | Dynamic | | |
| $H0_1$ | 70.0 | ≤60.0 | 62.0 | 56.5 | Heating minimum | Test mode | Heating |
| $H1_N$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating full | Test mode[a] | |
| $H1_2$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating full | Test mode[b,c] | |
| $H1_{2x}$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Dynamic | | |
| $H1_1$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Heating minimum | Test mode | |
| $H1_{1x}$ | 70.0 | ≤60.0 | 47.0 | 43.0 | Dynamic | | |
| $H2_v$ | 70.0 | ≤60.0 | 35.0 | 33.0 | Heating intermediate | Test mode | |
| $H2_v$ | 70.0 | ≤60.0 | 35.0 | 33.0[d] | Heating intermediate | Test mode | |
| $H2_v$ | 70.0 | ≤60.0 | 35.0 | 33.0[e] | Heating intermediate | Test mode | |
| $H2_v$ | 70.0 | ≤60.0 | 35.0 | 33.0[f] | Heating intermediate | Test mode | |
| $H3_2$ | 70.0 | ≤60.0 | 17.0 | 15.0 | Heating full | Test mode | |
| $H4_2$ | 70.0 | ≤60.0 | 5.0 | 4.0 | Heating full | Test mode | |
| $H4_2$ | 70.0 | ≤60.0 | 5.0 | ≤3.0 | Heating full | Test mode | |
| $H4_x$ | 70.0 | ≤60.0 | 5.0 | ≤3.0 | Dynamic | [g] | |
| $H5_x$ | 70.0 | ≤60.0 | -5.0 | ≤-5.0 | Dynamic | [g] | |
| $H6_x$ | 70.0 | ≤60.0 | -15.0 | ≤-15.0 | Dynamic | [g] | |

[a] Maximum speed that the system controls would operate the compressor in normal operation at 47°F ambient.

[b] Maximum speed that the system controls would operate the compressor in normal operation at 17°F ambient.

[c] $H1_2$ test is not needed if the $H1_N$ test uses the same compressor speed.

[d] Moisture is introduced from one side of the OD coil. Wet-bulb temperature differences between the faces of the OD coil to have no tolerance and the mean average of the wet-bulb sensors shall be 33°F ± 0.3°F.

[e] Moisture is introduced from one side of the OD coil. Wet-bulb temperature differences between the faces of the OD coil controlled to 33.0°F ±1.0°F and the mean average of the wet-bulb sensors shall be 33°F ± 0.3°F.

[f] Moisture is introduced from one side of the OD coil. Wet-bulb temperature differences between the faces of the OD coil controlled to 33.0°F ± 0.5°F and the mean average of the wet-bulb sensors shall be 33°F ± 0.3°F.

[g] Run with heating set point set to maximum value and chambers controlling to target conditions.

**Table 5. Dynamic mode cooling tolerances**

| | Test operating tolerance | Test condition tolerance |
|---|---|---|
| ID entering dry-bulb temperature (°F) | 4.0 | 2.0 |
| ID entering wet-bulb temperature (°F) | 2.0 | 1.0 |
| OD entering dry-bulb temperature (°F) | 2.0 | 0.5 |
| External resistance to airflow (in. water column) | — | 0.02[a] |
| Airflow nozzle pressure difference (% of reading) | 8.0 | — |
| Electrical voltage (% of reading) | 2.0 | 1.5 |

[a] Applies to non-ducted units only

**Table 6. Dynamic mode heating tolerances**

| | Test operating tolerance | Test condition tolerance |
|---|---|---|
| ID entering dry-bulb temperature (°F) | 4.0 | 2.0 |
| ID leaving dry-bulb temperature (°F) | 4.0 | — |
| ID entering wet-bulb temperature (°F) | 2.0 | 1.0 |
| ID leaving wet-bulb temperature (°F) | 2.0 | — |
| OD entering dry-bulb temperature (°F) | 2.0 | 0.5 |
| External resistance to airflow (in. water column) | — | 0.02[a] |
| Airflow nozzle pressure difference (% of reading) | 8.0 | — |
| Electrical voltage (% of reading) | 2.0 | 1.5 |

[a] Applies to non-ducted units only

# 4. SYSTEM TESTING

## 4.1 Manufacturer A System 1

System 1 consists of a non-ducted, high-wall-mounted ID unit and OD unit. The test results for system 1 are summarized in Table 7. The system was placed into the test mode using the ID unit remote control. Once in the test mode, the UUT ran at a fixed compressor speed and airflow. Modifying the temperature set point while in test mode caused the unit to leave test mode.

The vertical airflow vane on the ID unit was set to the position closest to horizontal, and the horizontal airflow vane was set to the center for all tests. The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 5/8 in. diameter vapor line. When transitioning from the standard rating tests to the dynamic tests, the unit was removed from the test mode by changing the temperature set point on the remote control.

There was not a test mode for the $H4_2$ test, so the $H3_2$ test mode setup was used to try and run the $H4_2$ test. However, the unit would not enter the test mode at the $H4_2$ conditions, so only the dynamic test was run at this condition. For the $H4_x$ and $H5_x$ tests, the temperature set point on the remote control was set to its maximum value and the chamber was controlled to maintain the desired ID conditions. The lowest advertised operating temperature for the system was -4°F, so the $H5_x$ test was run at this temperature instead of -5°F, and the $H6_x$ test was omitted.

The airflow for all dynamic tests was set to the high setting, even for the $B_{1x}$ and $H1_{1x}$ tests. During the $B_{1x}$ test, the compressor power and capacity was not stable. At approximately 30 min intervals, the compressor power would spike for about 30 s as shown in Figure 1. Data from both a full 30 min test period and a shorter period capturing more stable data are included in Table 7. The power was also not stable during the $H1_{1x}$ test as seen in Figure 2, and we could not achieve as low of a capacity as that of the $H1_1$ test. During the $H5_x$ test, we could not achieve steady-state data because of frequent defrost cycles. Data were collected, including the defrost cycle, and analyzed like a frost/defrost test.

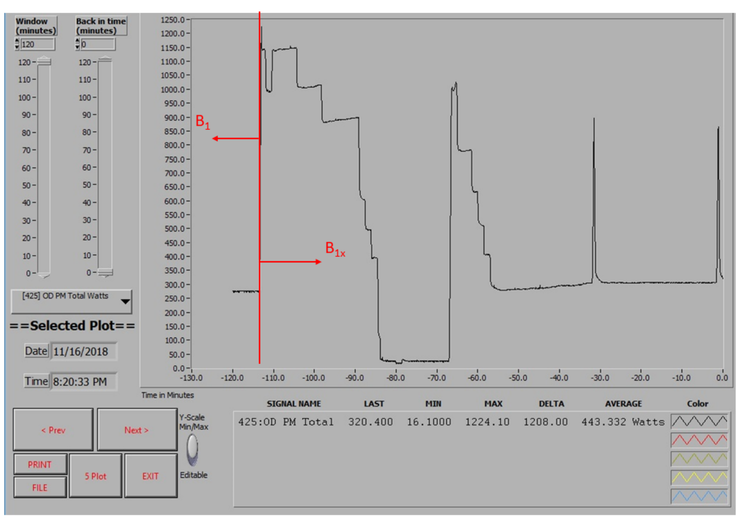



**Figure 1. Screen capture of power plot during the transition from $B_1$ test to $B_{1x}$ test**

**Table 7. System 1 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 22,286 | 1,710 | 13.03 | 891 | N/R | |
| $A_{2x}$ | 24,977 | 2,176 | 11.48 | 755 | N/R | |
| $B_1$ | 6,435 | 276 | 23.32 | 896 | N/R | |
| $B_{1x}$ | 6,835 | 308 | 22.19 | 761 | N/R | *a* |
| $B_{1x}$ | 6,775 | 306 | 22.15 | 760 | N/R | *a,b* |
| $H0_1$ | 11,072 | 653 | 4.97 | 488 | N/R | |
| $H1_N$ | 26,537 | 2,290 | 3.40 | 845 | N/R | |
| $H1_1$ | 9,969 | 672 | 4.35 | 887 | N/R | |
| $H1_{1x}$ | 14,720 | 1,140 | 3.78 | 719 | N/R | *a* |
| $H2_v$ | 12,960 | 1,170 | 3.25 | 894 | N/R | |
| $E_{std}$ | 12,773 | 1,164 | 3.22 | 895 | N/R | |
| $E_{marine}$ | 12,100 | 1,181 | 3.00 | 886 | N/R | |
| $H3_2$ | 17,465 | 2,026 | 2.53 | 880 | N/R | |
| $H4_x$ | 18,659 | 2,768 | 1.98 | 729 | N/R | *c* |
| $H5_x$ | 16,154 | 2,554 | 1.85 | 734 | N/R | *c,d* |

[a] Fan speed was set to high instead of low.
[b] Only 25.5 min of steady data was collected for this test.
[c] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.
[d] Unable to collect 30 min of stable data between defrost cycles, so transient data was collected and analyzed in similar fashion to the frost/defrost tests.

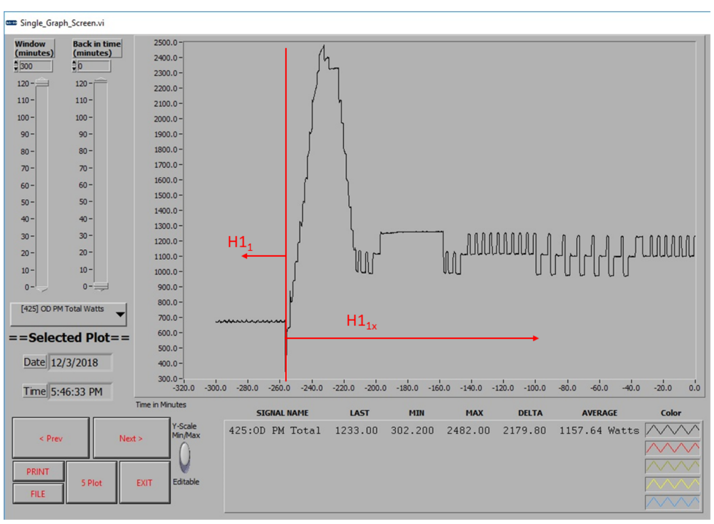



**Figure 2. Screen capture of power plot during the transition from $H1_1$ test to $H1_{1x}$ test**

## 4.2 Manufacturer B System 2 First Round of Testing

The first system from Manufacturer B consists of a ceiling-concealed, horizontal-ducted ID unit and OD unit. The test results for this system are summarized in Table 8. The system was controlled with a wired thermostat that was used to set the appropriate airflow for each test. Minimum compressor speed tests for both cooling and heating tests used the lowest airflow setting, the intermediate compressor speed test used the medium airflow setting, and all other tests used the high airflow setting.

The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 5/8 in. diameter vapor line. The ID unit was not installed with an inlet duct because of space constraints in the chamber. To achieve the proper airflow at the Appendix M1 external static pressure, the ID fan pressure setting was changed from the default of 50 Pa to 100 Pa through menu selections on the thermostat. When transitioning from the standard rating tests to the dynamic tests, power had to be removed from the unit. Midway through testing, we learned that the ID controller could be used to query the current compressor speed for the system. This was recorded with the test data when possible.

During the dynamic tests, $B_{1x}$ and $H1_{1x}$, we could not get the unit to run at as low of a capacity as measured during the standard rating tests at those conditions. The unit was cycling on and off, but at a capacity that was higher than the standard rating test. Data were collected at the lowest stable capacity achieved. Plots of the power and ID inlet dry-bulb temperature during the transition from the standard tests to dynamic tests are shown in Figures 3–6. The lowest advertised operating temperature for the system was -13°F, so the $H6_x$ test was run at this temperature instead of -15°F.

The $H4_x$ test results appear to disagree with the other test data. Therefore, this test was repeated during the second round of testing on this unit. Although no errors could be identified based on the test log and the data is accurate, we believed that this test was not set up properly to achieve maximum capacity from the unit. This is noted in a footnote in Table 8 and also indicated by struck-through text to ensure it is not overlooked.

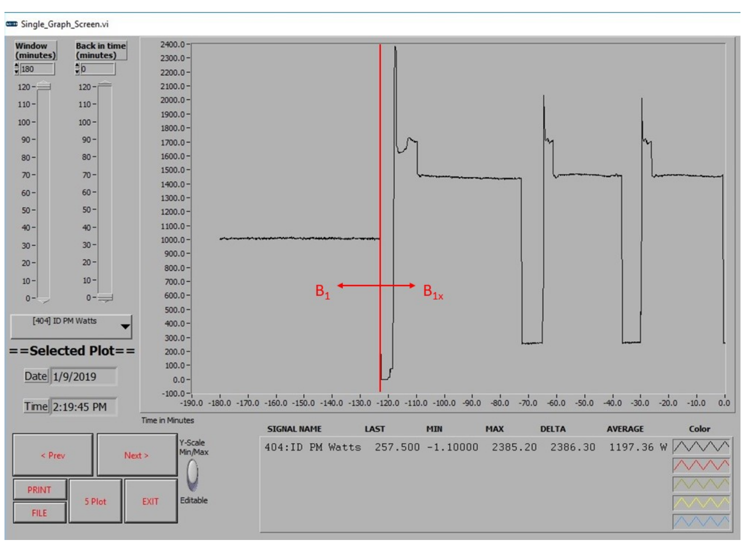



**Figure 3. Screen capture of power plot during the transition from $B_1$ test to $B_{1x}$ test**

**Table 8. System 2 test results: first round**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 32,546 | 2,604 | 12.50 | 1,075 | 47 | |
| $A_{2x}$ | 31,863 | 2,548 | 12.51 | 1,076 | N/R | |
| $B_1$ | 16,718 | 890 | 18.79 | 738 | 20 | |
| $B_{1x}$ | 22,852 | 1,273 | 17.95 | 736 | N/R | |
| $H0_1$ | 19,875 | 1,017 | 5.73 | 762 | 23 | |
| $H1_N$ | 39,697 | 3,222 | 3.61 | 1,120 | 62 | |
| $H1_{Nx}$ | 41,479 | 3,447 | 3.53 | 1,107 | 65 | |
| $H1_1$ | 16,168 | 971 | 4.88 | 765 | 23 | |
| $H1_{1x}$ | 20,058 | 1,390 | 4.23 | 761 | N/R | |
| $H2_v$ | 21,846 | 1,776 | 3.61 | 968 | 36 | |
| $E_{std}$ | 21,844 | 1,775 | 3.61 | 970 | 36 | |
| $E_{marine}$ | 20,300 | 1,758 | 3.38 | 973 | 36 | |
| $H3_2$ | 28,878 | 3,160 | 2.68 | 1,140 | 62 | |
| $H4_2$ | 23,189 | 3,070 | 2.21 | 1,147 | 62 | *a* |
| $H4_2$ | 18,713 | 2,780 | 1.97 | 1,138 | 62 | *a,b* |
| $\cancel{H4_x}$ | ~~18,774~~ | ~~2,698~~ | ~~2.04~~ | ~~1,137~~ | ~~61~~ | *c,d,f* |
| $H5_x$ | 35,553 | 5,950 | 1.75 | 1,132 | 105 | *c* |
| $H6_x$ | 32,479 | 5,950 | 1.60 | 1,134 | 107 | *c,e* |

[a] Test was run in the $H3_2$ test mode.
[b] Test was run with and OD inlet wet-bulb temperature of less than 3°F.
[c] ID remote was set to maximum temperature (83°F) in an attempt to force the unit to run at maximum capacity.
[d] Average OD wet-bulb temperature for the test was 3.73°F. No frost was visible on the OD coil.
[e] Test was run at an OD dry-bulb temperature of -13°F, which was the lowest advertised operating temperature for the system.
[f] While the test data is accurate, it is believed that this test was not setup properly to achieve maximum capacity from the unit. As such, the data has been struck through to help ensure that it is not mistakenly used.

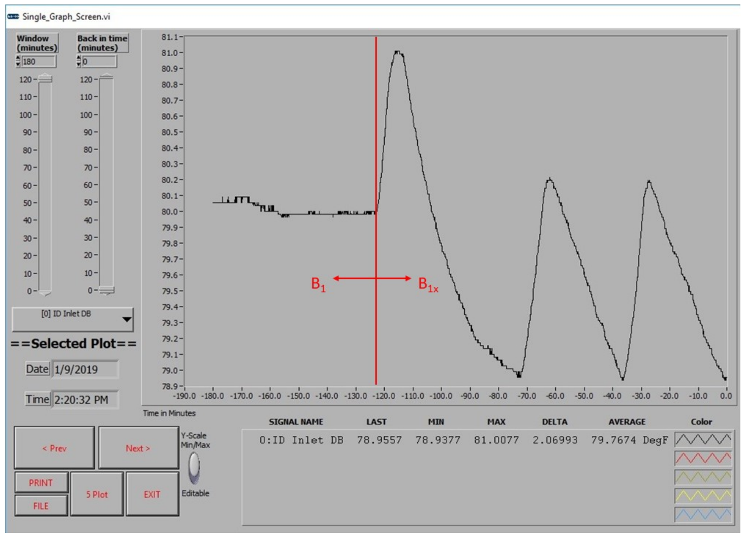



**Figure 4. Screen capture of ID inlet dry-bulb temperature plot during the transition from $B_1$ test to $B_{1x}$ test**

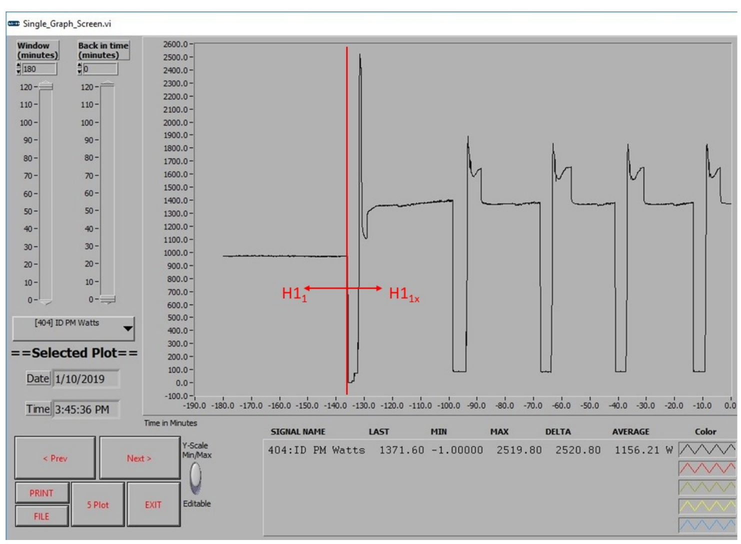



**Figure 5. Screen capture of power plot during the transition from $H1_1$ test to $H1_{1x}$ test**

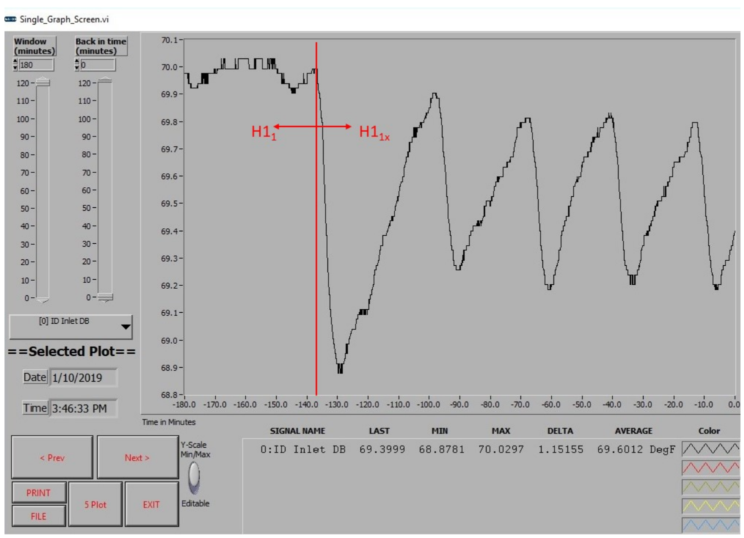



**Figure 6. Screen capture of ID inlet dry-bulb temperature plot during the transition from H1$_1$ test to H1$_{1x}$ test**

## 4.3 Manufacturer B System 2 Second Round of Testing

The second round of testing on this system focused on tests at the Appendix M external static pressure. The ID unit configuration for this test was different from the first round. The unit and code tester were set up in an under/over configuration and oriented differently to allow space for an inlet duct on the ID unit. The results from this round of tests are shown in Table 9.

As with the previous tests, on this unit, we could not achieve matching capacity on the $B_{1x}$ and $H1_{1x}$ tests. The unit was cycling on at a higher capacity than the standard tests and then cycling off. The ID chamber heating/cooling supply was adjusted to achieve stable operation at the lowest achievable capacity for these tests.

An additional $H4_x$ test was run at the Appendix M1 external static pressure. This was a repeat of a first-round test on this system because the initial test is believed to have had a setup error that did not allow it to run at maximum capacity. This test is marked in Table 9 with footnote *d* and should be used in place of the $H4_x$ text that is struck through in Table 8.

**Table 9. System 2 test results: second round per Appendix M (except where noted)**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | OD fan (rpm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|---|
| $A_2$ | 33,500 | 2,606 | 12.86 | 1,143 | 741 | 47 | |
| $B_1$ | 17,139 | 880 | 19.48 | 793 | 625 | 20 | |
| $B_{1x}$ | 23,231 | 1,260 | 18.44 | 739 | 469 | 30 | |
| $H0_1$ | 20,219 | 990 | 5.99 | 847 | 409 | 23 | |
| $H1_N$ | 40,223 | 3,130 | 3.77 | 1,276 | 705 | 62 | |
| $H1_{Nx}$ | 38,362 | 2,940 | 3.82 | 1,242 | 701 | 59 | |
| $H1_1$ | 16,270 | 946 | 5.04 | 849 | 415 | 23 | |
| $H1_{1x}$ | 20,133 | 1,380 | 4.28 | 836 | 705 | 30 | |
| $H2_v$ | 21,888 | 1,739 | 3.69 | 1,064 | 705 | 36 | |
| $E_{std}$ | 21,664 | 1,748 | 3.63 | 1,049 | 704 | 36 | |
| $E_{marine}$ | 20,124 | 1,720 | 3.43 | 1,040 | 703 | 36 | |
| $H3_2$ | 29,031 | 3,094 | 2.75 | 1,262 | 742 | 62 | |
| $H4_2$ | 18,588 | 2,738 | 1.99 | 1,267 | 745 | 62 | *a* |
| $H4_2$ | 18,609 | 2,738 | 1.99 | 1,262 | 745 | 62 | *a,b* |
| $H4_x$ | 39,959 | 5,990 | 1.96 | 1,264 | 720 | 105 | *b,c* |
| $H4_x$ | 38,568 | 5,920 | 1.91 | 1,170 | 722 | 104 | *b,c,d* |
| $H5_x$ | 35,205 | 5,870 | 1.76 | 1,267 | 721 | 105 | *c* |
| $H6_x$ | 32,620 | 6,030 | 1.59 | 1,251 | 720 | 111 | *c,e* |

[a] Test was run in the $H3_2$ test mode.
[b] Test was run with and OD inlet wet-bulb temperature of less than 3°F.
[c] ID remote was set to maximum temperature (83°F) in an attempt to force the unit to run at maximum capacity.
[d] Test was run at Appendix M1 external static pressure as a retest for the first round $H4_x$ test, which did not show the same trend as during the second round of tests.
[e] Test was run at an OD dry-bulb temperature of -13°F, which was the lowest advertised operating temperature for the system.

## 4.4 Manufacturer B System 3

This system consists of two non-ducted, high-wall–mounted ID units and an OD unit. Test results are shown in Table 10.

The two ID units were placed side by side, and a single supply plenum was used to connect them to the airflow measurement apparatus. A photograph of this setup is shown in Figure 7. This setup had the unintended side-effect of isolating the ID air temperature sensor, located on the lower section of the side of the unit, from the ambient air in the chamber. Because the dynamic tests require the unit to respond to changes in the chamber temperature as measured by this sensor, the dynamic tests were initially unsuccessful. To resolve this issue, the UUT temperature sensors were relocated to the return air inlet of the units by extending their wiring, also shown in Figure 7. This improved the response of the system during the dynamic tests, as well as the system's ability to match the chamber load.

The frost/defrost tests for this unit did not exactly follow the testing standard. There was no way to manually initiate a defrost cycle on the unit, and if the unit defrosted automatically, it would switch to running full compressor speed after the defrost cycle instead of intermediate compressor speed. Therefore, the $H2_v$ and $E_{std}$ test data consist of 6 h of operation at stable conditions without a defrost cycle prior to the start of the test data. During the $E_{marine}$ tests, the unit defrosted at just after 2.5 h (and subsequently switched to full compressor speed), so only the 2.5 h of stable data were used in the analysis.

During the $H5_x$ and $H6_x$ tests, the unit defrosted frequently enough that we could not collect 30 min of stable data. Therefore, data were collected from the end of the defrost cycle to the end of the following defrost cycle and analyzed as a frost/defrost test. For comparison, an $H4_x$ test was run in this fashion in addition to the regular $H4_x$ test.

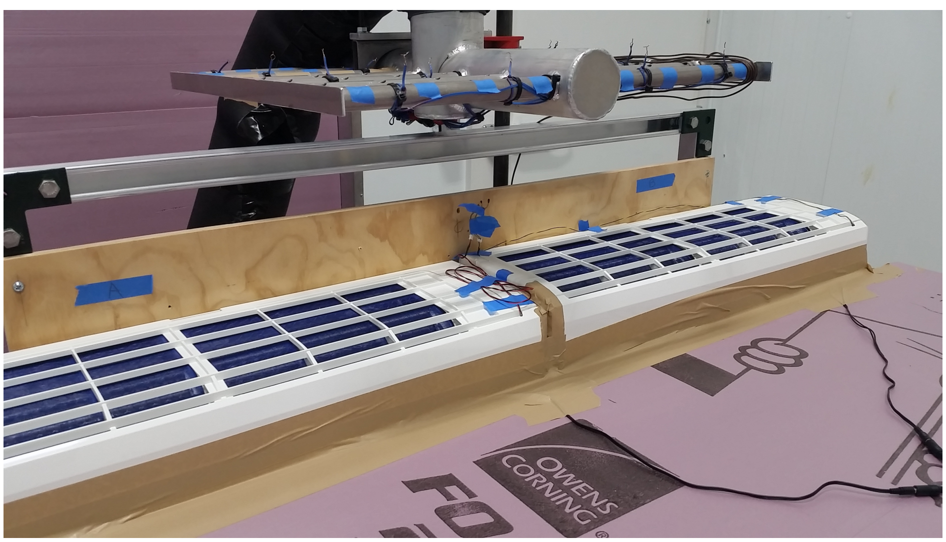



**Figure 7. Photograph of side-by-side ID unit setup and extended ID air temperature sensors**

**Table 10. System 3 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 18,264 | 1,244 | 14.68 | 725 | 26 | |
| $A_{2x}$ | 17,179 | 1,143 | 15.02 | 739 | 25.9 | |
| $B_1$ | 11,369 | 649 | 17.52 | 328 | 18 | |
| $B_{1x}$ | 12,232 | 778 | 15.73 | 326 | 22 | |
| $H0_1$ | 13,211 | 893 | 4.34 | 332 | 18 | |
| $H1_N$ | 23,263 | 1,603 | 4.25 | 686 | 35 | |
| $H1_{Nx}$ | 23,725 | 2,016 | 3.45 | 682 | 36 | |
| $H1_1$ | 9,719 | 788 | 3.61 | 335 | 18 | |
| $H1_{1x}$ | 10,486 | 1,242 | 2.47 | 335 | 20 | |
| $H2_v$ | 13,779 | 990 | 4.08 | 697 | 24 | *a,b* |
| $E_{std}$ | 13,447 | 986 | 4.00 | 691 | 24 | *a,b* |
| $E_{marine}$ | 13,012 | 987 | 3.87 | 692 | 24 | *a* |
| $H3_2$ | 13,865 | 1,494 | 2.72 | 698 | 35 | |
| $H4_2$ | 10,641 | 1,363 | 2.29 | 705 | 35 | *c,d* |
| $H4_2$ | 9,616 | 1,312 | 2.15 | 705 | 35 | *c,d,e* |
| $H4_x$ | 15,737 | 2,258 | 2.04 | 692 | 56.5 | *f* |
| $H4_x$ | 13,397 | 2,226 | 1.76 | 659 | 61 | *e,f* |
| $H5_x$ | 6,466 | 1,964 | 0.96 | 653 | 67.8 | *f,g* |
| $H6_x$ | 5,004 | 1,869 | 0.78 | 660 | 67.8 | *f,g,h* |

[a] Test did not begin with a defrost cycle.
[b] Unit ran for more than 6 h without defrosting, so no defrost cycle is included in these data.
[c] Test was run in the $H3_2$ test mode.
[d] Test was run with and OD inlet wet-bulb temperature of less than 3°F.
[e] Test was run as a frost/defrost test with data being collected from the end of a defrost cycle to the end of the next defrost cycle for comparison with the $H5_x$ and $H6_x$ tests.
[f] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.
[g] Unable to collect 30 min of stable data between defrost cycles, so transient data was collected and analyzed in similar fashion to the frost/defrost tests.
[h] Test was run at an OD dry-bulb temperature of -13°F, which was the lowest advertised operating temperature for the system.

## 4.5 Manufacturer C System 4

System 4 consists of a 36,000 Btu/h, central-ducted, fan-coil ID unit and a 36,000 Btu/h OD. The manufacturer provided a thermostat that allowed for the intermediate compressor speed tests to be run (the standard thermostat has a test mode that only allows minimum and maximum compressor speed tests to be run). The testing on this unit was exploratory and used a prototype inverter board provided by the manufacturer to power the compressor. The inverter board was intended to provide additional ability to "overspeed" the compressor for low ambient heating operation. The manufacturer also provided a hardware interface to connect a computer to the OD unit and software that provided information on the system operation and allowed for the compressor speed to be overridden. For the $H4_x$, $H5_x$, and $H6_x$ tests, the compressor speed was manually increased until the inverter began cutting back the compressor speed based on its internal protection limits. These tests reflect the maximum stable compressor speed achieved at each test condition using the non-factory inverter board provided by the manufacturer. The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 7/8 in. diameter vapor line. Test results for this system are shown in Table 11.

Several problems were encountered during testing related to refrigerant charge and refrigerant pressure stability. The unit was initially charged in cooling mode at $A_2$ test conditions. A leak was discovered on one of the pressure transducer lines after the $A_2$ test was run, so the unit was charged again in heating mode using the manufacturer's instructions. At this charge level, there was no subcooling during the $B_1$ test, so 26 oz. of refrigerant were added to establish subcooling. At this charge level, the unit would not run at the $H1_2$ test conditions because of high head pressure. To achieve 28°F subcooling at the $H1_2$ test condition and 400 psig at the liquid line service valve, 14 oz. of refrigerant were removed. After additional issues with the charge at other test conditions, we decided to replace the OD thermostatic expansion valve (TXV). The unit charge was recovered and the TXV was replaced with an adjustable TXV. After being placed under vacuum, the factory charge of 199 plus 10.8 oz. (for additional line set and the refrigerant flow meter) was added. The $H1_2$ test was rerun to check charge and an additional 16 oz. of refrigerant were added. This charge level was used for the remaining tests. During the $H3_2$ test, the system operated without superheat or subcooling. The $H1_2$ test was rerun after the OD TXV was adjusted to provide additional superheat to the compressor (12°F). The sensing bulb of the OD TXV was also insulated and a copper shim was placed between the line and the bulb to try and dampen its response and reduce pressure oscillations.

An additional set of $H4_x$ test data is included in the table at a lower than maximum compressor speed (90 Hz) for reference. The $H2_v$ tests that examined moisture uniformity around the OD coil were mistakenly run at lower external static pressure (ESP) than required by Appendix M1. This is indicated with a footnote in Table 11.

**Table 11. System 4 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 33,681 | 2,641 | 12.75 | 1,213 | 67 | a |
| $H0_1$ | 11,418 | 696 | 4.81 | 655 | 22 | |
| $H1_N$ | 31,682 | 2,913 | 3.19 | 1,159 | 67 | |
| $H1_2$ | 33,837 | 3,169 | 3.13 | 1,179 | 70 | |
| $H1_{2x}$ | 33,256 | 2,993 | 3.26 | 1,175 | 70 | |
| $H1_1$ | 9,540 | 698 | 4.01 | 642 | 22 | |
| $H1_{1x}$ | 9,736 | 720 | 3.96 | 641 | N/R | b |
| $H2_v$ | 18,026 | 1,733 | 3.05 | 1,028 | 51 | |
| $H2_v$ | 17,674 | 1,707 | 3.03 | 1,037 | 51 | c,d |
| $H2_v$ | 17,915 | 1,695 | 3.10 | 1,036 | 51 | c,e |
| $H2_v$ | 18,559 | 1,741 | 3.12 | 1,037 | 51 | c,f |
| $H3_2$ | 22,115 | 2,785 | 2.33 | 1,190 | 70 | |
| $H4_2$ | 18,260 | 2,624 | 2.04 | 1,192 | 70 | |
| $H4_x$ | 23,038 | 3,622 | 1.86 | 1,185 | 90 | g |
| $H4_x$ | 24,577 | 3,986 | 1.81 | 1,182 | 96 | g |
| $H5_x$ | 22,947 | 4,118 | 1.63 | 1,186 | 111 | |
| $H6_x$ | 19,299 | 4,056 | 1.39 | 1,192 | 120 | |

[a] Test was run prior to adjustment of OD TXV. This adjustment is not expected to affect this test.
[b] Compressor speed was not recorded during this test.
[c] Test was run at lower ESP (~0.26 in. water column) than required (0.37 in. water column).
[d] OD inlet dry-bulb and wet-bulb temperatures were measured independently on all four faces of the OD coil. Test was run while trying to minimize the difference between the wet-bulb temperatures on each face.
[e] OD inlet dry-bulb and wet-bulb temperatures were measured independently on all four faces of the OD coil. Test was run while trying to maintain a 1°F difference between the wet-bulb temperatures.
[f] OD inlet dry-bulb and wet-bulb temperatures were measured independently on all four faces of the OD coil. Test was run while trying to maintain the maximum difference possible between the wet-bulb temperatures. This difference was 1.9°F.
[g] Test was run at an OD inlet wet-bulb temperature of ~3.6°F. No significant frost was observed on the OD coil.

## 4.6 Manufacturer D System 5

System 5 consists of a non-ducted, high-wall-mounted ID unit and OD unit. The test results for this system are summarized in Table 12. The system was placed into the test mode using the ID unit remote control. Once in the test mode, the UUT ran at a fixed compressor speed and airflow.

The vertical airflow vane on the ID unit was set to the position closest to horizontal, and the horizontal vane was set to the center for all tests. The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 5/8 in. diameter vapor line.

The airflow for the $B_{1x}$ test was set to the high setting. The power during the $B_{1x}$ was also not stable. As seen in Figure 8, there were spikes in the power at approximately 30 min intervals. Three versions of the $H1_{1x}$ test were run; one with high airflow, one with low airflow, and one with auto airflow. During the low airflow and auto airflow tests, the unit did not run at a stable power as shown in Figure 9. During the high airflow test, the unit did run at a stable power, but at a higher capacity than the $H1_x$. We could not achieve a lower capacity without the unit cycling off during this test, so data were collected at the lowest stable capacity. There was not a test mode for the $H4_2$ test, so the $H3_2$ test mode setup was used to try and run the $H4_2$ test. However, the unit would not enter the test mode at the $H4_2$ conditions, so only the dynamic test was run. The temperature set point on the remote control was set to its maximum value and the chamber control maintained the desired ID conditions for the $H4_2$, $H5_2$, and $H6_2$ tests. The lowest advertised operating temperature for the system was -14°F, so the $H6_2$ test was run at this temperature instead of -15°F. These tests were run during the summer, and the OD chamber could not reach a wet-bulb temperature of 3°F for the $H4_2$ tests, so data were collected at the lowest achievable wet-bulb temperature. Immediately following the collection of the test data, the OD coil was inspected and no significant frost was visible.

The input frequency to the compressor was measured on this unit by using a multimeter to measure the frequency output of a current transformer attached to one of the three power wires leading to the compressor. These data are included in the table for tests when personnel were available to take this measurement.

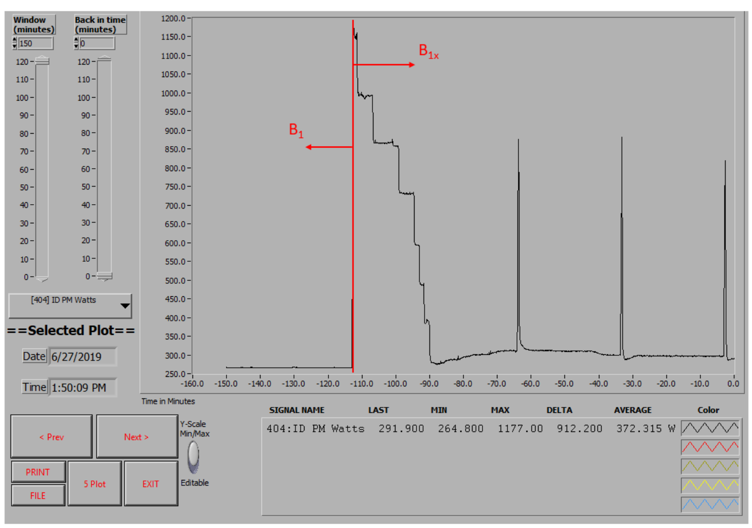



**Figure 8. Screen capture of power plot during the transition from $B_1$ test to $B_{1x}$ test**

**Table 12. System 5 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 22,353 | 1,680 | 13.31 | 815 | 45.2 | |
| $A_{2x}$ | 21,182 | 1,556 | 13.61 | 696 | N/R | |
| $B_1$ | 6,208 | 266 | 23.34 | 802 | 10.0 | |
| $B_{1x}$ | 6,920 | 302 | 22.91 | 685 | 10.0 | [b] |
| $H0_1$ | 10,864 | 680 | 4.68 | 402 | 22.0 | |
| $H1_N$ | 26,789 | 2,290 | 3.43 | 769 | 67.3 | |
| $H1_{Nx}$ | 27,669 | 2,652 | 3.06 | 599 | 71.3 | |
| $H1_1$ | 9,960 | 656 | 4.45 | 783 | 22.0 | |
| $H1_{1x}$ | 9,903 | 774 | 3.75 | 404 | 22.1–27.1 | |
| $H1_{1x}$ | 10,780 | 808 | 3.91 | 507 | 22.1–27.1 | [a] |
| $H1_{1x}$ | 13,756 | 1,072 | 3.76 | 634 | 34.5 | [b] |
| $H2_v$ | 12,634 | 1,162 | 3.19 | 780 | 37.2 | |
| $E_{std}$ | 12,554 | 1,159 | 3.17 | 781 | 37.2 | |
| $E_{marine}$ | 12,279 | 1,169 | 3.08 | 781 | 37.2 | |
| $H3_2$ | 17,875 | 1,990 | 2.63 | 769 | 67.3 | |
| $H4_x$ | 18,109 | 2,688 | 1.97 | 626 | 92.3 | [c] |
| $H4_x$ | 19,086 | 2,760 | 2.03 | 624 | 96.3 | [c,d] |
| $H5_x$ | 16,180 | 2,580 | 1.84 | 628 | 96.3 | [c,e] |
| $H6_x$ | 13,225 | 2,364 | 1.64 | 645 | 96.5 | [c,f] |

[a] Test was run with auto fan mode selected.
[b] Test was run with high fan mode selected.
[c] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.
[d] Test was run with an OD inlet wet-bulb temperature of 3.6°F.
[e] Test was run with an OD inlet dry-bulb temperature of -4.0°F.
[f] Test was run with an OD inlet dry-bulb temperature of -14.0°F.

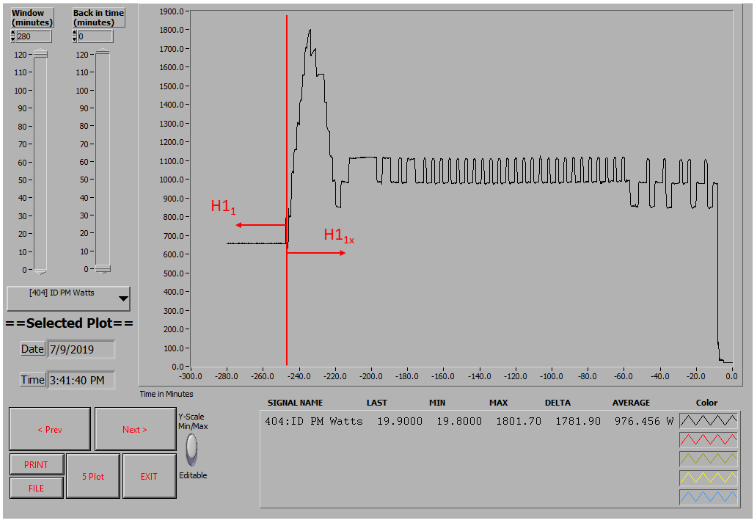



**Figure 9. Screen capture of power plot during the transition from H1$_1$ test to H1$_{1x}$ test**

## 4.7 Manufacturer E System 6

The first system tested from Manufacturer E consists of a 24,000 Btu/h, non-ducted, high-wall–mounted ID unit and a 24,000 Btu/h OD unit. The ID unit remote was used to put the system in test mode. The vertical airflow vane on the ID unit was set to position 3, and the horizontal vane was set to the center prior entering the test mode. However, when entering the test mode, the vertical vanes closed completely and then reopened to approximately position 2 for cooling. This was observed when running the $A_2$ free-air test. The vane position for the heating test mode was not observed because the ID duct work obstructed the view of the vanes during the tests. The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 5/8 in. diameter vapor line.

Test results are summarized in Table 13. The $B1_x$ test was run at both high and low airflow settings on the ID unit remote control. During the $H1_{1x}$ tests, the system cycled frequently and did not run at as low of a capacity as the standard $H1_1$ test. This can be seen from the power and capacity plots in Figures 10 and 11. During the $H6_x$ test, steady-state conditions could not be attained between defrost cycles, so data were collected from the end of a defrost cycle to the end of the next defrost cycle. The performance of the system was calculated in the same fashion the frost/defrost tests (e.g., $H2_v$).

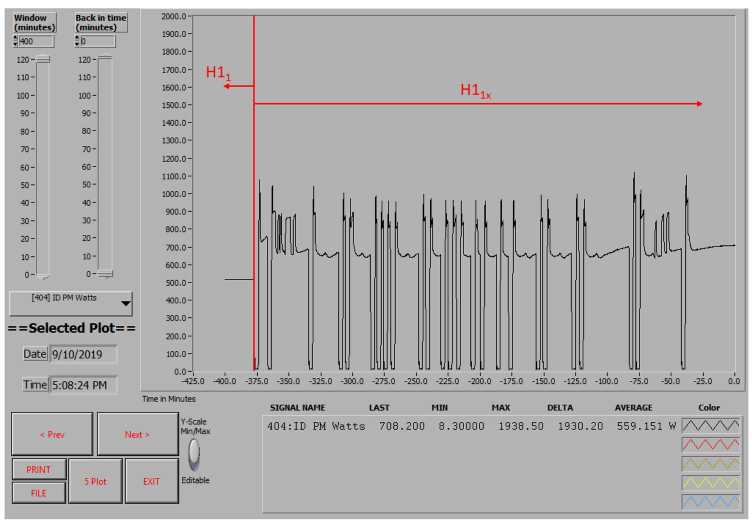



**Figure 10. Screen capture of power plot during the transition from $H1_1$ test to $H1_{1x}$ test**

**Table 13. System 6 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 21,668 | 1,708 | 12.69 | 587 | 38.0 | |
| $A_{2x}$ | 22,734 | 1,884 | 12.07 | 586 | 41.3 | |
| $B_1$ | 11,124 | 536 | 20.75 | 502 | 16.1 | |
| $B_{1x}$ | 11,018 | 586 | 18.80 | 482 | 18.2 | [a] |
| $B_{1x}$ | 11,528 | 654 | 17.63 | 389 | N/R | |
| $H0_1$ | 11,654 | 520 | 6.57 | 497 | 16.1 | |
| $H1_N$ | 24,239 | 1,924 | 3.69 | 604 | 43.4 | |
| $H1_{Nx}$ | 24,728 | 2,064 | 3.51 | 609 | 46.4 | |
| $H1_1$ | 9,197 | 518 | 5.20 | 500 | 16.2 | |
| $H1_{1x}$ | 10,934 | 710 | 4.51 | 392 | 20.1 | |
| $H2_v$ | 12,255 | 964 | 3.73 | 626 | 24.2 | [b] |
| $E_{std}$ | 12,133 | 961 | 3.70 | 621 | 24.2 | [b] |
| $E_{marine}$ | 11,269 | 950 | 3.48 | 625 | 24.2 | |
| $H3_2$ | 15,996 | 1,606 | 2.92 | 625 | 43.4 | |
| $H4_2$ | 13,174 | 1,492 | 2.59 | 660 | 43.4 | [c] |
| $H4_2$ | 12,992 | 1,484 | 2.57 | 662 | 43.4 | [c,d] |
| $H4_x$ | 22,441 | 3,466 | 1.90 | 549 | 85.8 | [d,e] |
| $H5_x$ | 19,718 | 3,342 | 1.73 | 506 | 88.2 | [e] |
| $H6_x$ | 11,878 | 2,711 | 1.28 | 433 | 88.3 | [e,f] |

[a] Fan speed was set to high instead of low.
[b] Unit ran for more than 6 h without defrosting, so no defrost cycle is included in these data.
[c] Test was run using the $H3_2$ test mode.
[d] Test was run with an OD inlet wet-bulb temperature of less than 3°F.
[e] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.
[f] Unable to collect 30 min of stable data between defrost cycles, so transient data was collected and analyzed in similar fashion to the frost/defrost tests.

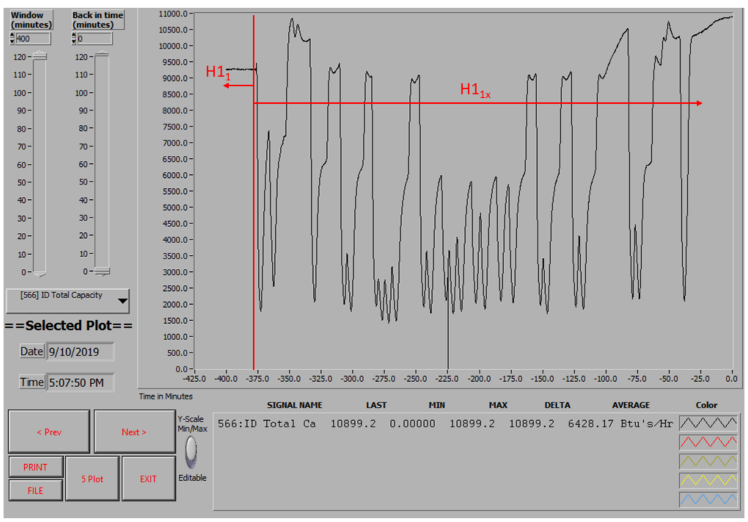


**Figure 11. Screen capture of capacity plot during the transition from H1$_1$ test to H1$_{1x}$ test**

## 4.8 Manufacturer E System 7a

The second system tested from Manufacturer E uses the same 24,000 Btu/h, non-ducted, high-wall–mounted ID unit as the system 6 unit but is paired with a 15,000 Btu/h OD unit. Because this system used the same ID unit as the system in Section 4.7, the process of running the tests was identical. The system was installed with 25 ft of line set having a 3/8 in. diameter liquid line and 5/8 in. diameter vapor line according to the ID unit installation instructions. The OD unit installation instructions did not provide guidance on the line set sizing.

The system was initially tested with the factory charge of 3 lb 1 oz. of refrigerant. The cooling tests and $H1_N$ tests were run with this charge. The measured capacity during the $H1_N$ test was lower than expected, and we believed that the refrigerant charge was low because of the pairing of a larger ID unit and refrigerant lines than what was intended for the OD unit. An additional 12 oz. of refrigerant was added to the system. $H1_N$ capacity was still slightly lower than expected, but the $A_2$ test indicated very little superheat, indicating near overcharge condition. The majority of the heating tests were run, with the exception of the dynamic tests and frost/defrost tests. The unit would not reliably run the $H1_1$ or $H1_{1x}$ tests because it would often not establish subcooling or superheat and shutdown. Therefore, the matching ID unit was ordered, installed, and tested in place of the existing ID unit.

Test results are summarized for the system in Table 14.

**Table 14. System 7a test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 15,662 | 1,074 | 14.59 | 588 | 50.1 | a |
| $A_{2x}$ | 15,369 | 1,022 | 15.04 | 586 | 48.4 | a |
| $B_1$ | 4,753 | 162 | 29.42 | 413 | 12.2 | a |
| $B_{1x}$ | 6,687 | 265 | 25.21 | 420 | 19.2 | a |
| $H1_N$ | 16,213 | 1,108 | 4.29 | 626 | 61.5 | a |
| $A_2$ | 15,775 | 1,146 | 13.76 | 573 | 50.1 | b |
| $H1_N$ | 17,499 | 1,150 | 4.46 | 624 | 61.5 | b |
| $H1_1$ | 3,335 | 196 | 4.99 | 507 | 12.2 | b |
| $H3_2$ | 11,300 | 1,064 | 3.11 | 636 | 61.5 | b |
| $H4_2$ | 9,196 | 1,008 | 2.67 | 640 | 61.5 | b,c,d |
| $H4_x$ | 19,649 | 2,706 | 2.13 | 515 | 131.4 | b,d,e |
| $H5_x$ | 16,884 | 2,446 | 2.02 | 525 | N/R | b,e,f |
| $H5_x$ | 16,279 | 2,562 | 1.86 | 414 | 131.4 | b,e |
| $H6_x$ | 13,893 | 2,310 | 1.76 | 426 | 131.5 | b,e |

[a] Factory charge of 3 lbs. 1 oz.
[b] Factory charge plus 12 oz. of refrigerant. Total charge = 3 lbs. 13 oz.
[c] Unit was run using the $H3_2$ test mode.
[d] Test was run with an OD inlet wet-bulb temperature of less than 3°F.
[e] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.
[f] Compressor speed was not recorded

## 4.9 Manufacturer E System 7b

The third system tested from Manufacturer E uses a 15,000 Btu/h, non-ducted, high-wall–mounted ID unit paired with the 15,000 Btu/h OD unit. The test mode was initiated using the ID remote. As with the other system from Manufacturer E, the vertical airflow vane on the ID unit was set to position 3 and the horizontal vane was set to the center. However, when entering the test mode, the vertical vanes closed completely and then reopened to approximately position 2 for cooling. This was observed when running the $A_2$ free-air test. The vane position for the heating test mode was not observed because the ID duct work obstructed the view of the vanes during the tests. The system was installed with 25 ft of line set having a 1/4 in. diameter liquid line and 1/2 in. diameter vapor line.

Test results are summarized for the system in Table 15. All testing went smoothly with the exception of the $E_{marine}$ test. During this test, the cooling coils for the OD chamber kept frosting over. This reduced the airflow enough that the safety switches for the electric heat and steam generator would trip, causing the conditions in the OD chamber to go out of tolerance. About 3.3 h of data were collected for the $E_{marine}$ test prior to issues with the OD coils, and these were used for the test data. The test was initiated with a defrost cycle but did not terminate with a defrost cycle and did not reach the 6 h limit. During the $B_{1x}$ and $H1_{1x}$ tests, we did not achieve as low of a capacity as the corresponding rating test without the unit cycling off and on as seen in Figures 12 and 13. For these tests, the unit was run at lowest achievable stable capacity.

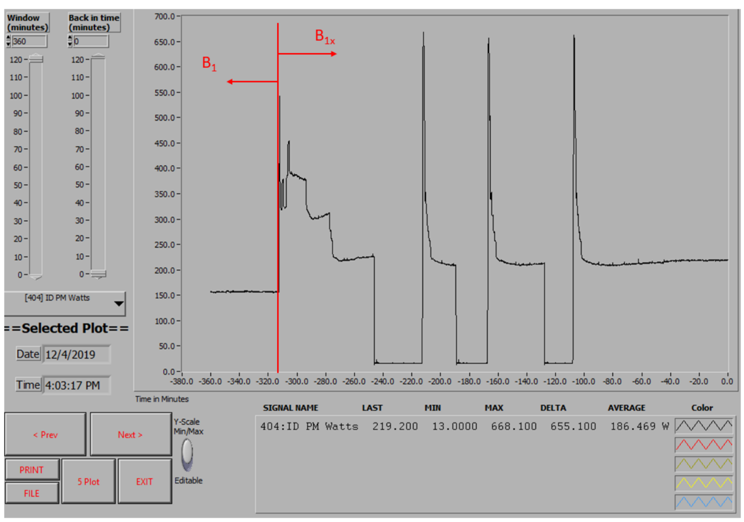



**Figure 12. Screen capture of power plot during the transition from $B_1$ test to $B_{1x}$ test**

**Table 15. System 7b test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | Notes |
|---|---|---|---|---|---|---|
| $A_2$ | 15,216 | 1,078 | 14.11 | 499 | 50.1 | |
| $A_{2x}$ | 14,797 | 1,022 | 14.48 | 461 | 48.4 | |
| $B_1$ | 4,752 | 157 | 30.27 | 340 | 12.2 | |
| $B_{1x}$ | 5,832 | 219 | 26.65 | 342 | 16.1 | |
| $H0_1$ | 4,311 | 154 | 8.22 | 415 | 12.2 | |
| $H1_N$ | 18,278 | 1,198 | 4.47 | 500 | 61.5 | |
| $H1_{Nx}$ | 17,815 | 1,232 | 4.24 | 492 | 63.5 | |
| $H1_1$ | 3,644 | 176 | 6.07 | 429 | 12.2 | |
| $H1_{1x}$ | 5,053 | 286 | 5.18 | 337 | 16.2 | |
| $H2_v$ | 7,152 | 498 | 4.21 | 513 | 28.6 | [a] |
| $E_{std}$ | 6,910 | 498 | 4.07 | 520 | 28.6 | [a] |
| $E_{marine}$ | 6,369 | 502 | 3.71 | 506 | 28.6 | [b] |
| $H3_2$ | 11,888 | 1,064 | 3.27 | 516 | 61.5 | |
| $H4_2$ | 9,750 | 998 | 2.86 | 517 | 61.5 | [c] |
| $H4_2$ | 9,707 | 996 | 2.86 | 520 | 61.5 | [c,d] |
| $H4_x$ | 20,509 | 2,688 | 2.24 | 488 | 131.4 | [d,e] |
| $H5_x$ | 17,776 | 2,458 | 2.12 | 492 | 131.4 | [e] |
| $H6_x$ | 14,656 | 2,268 | 1.89 | 444 | 131.5 | [e] |

[a] Unit ran for more than 6 h without defrosting, so no defrost cycle is included in this data.
[b] There were issues maintaining the test chamber conditions without the cooling coils frosting over. The data are for ~3.3 h of operation after an initial defrost.
[c] Test was run using the $H3_2$ test mode.
[d] Test was run with an OD inlet wet-bulb temperature of less than 3°F.
[e] ID remote was set to maximum temperature (88°F) in an attempt to force the unit to run at maximum capacity.

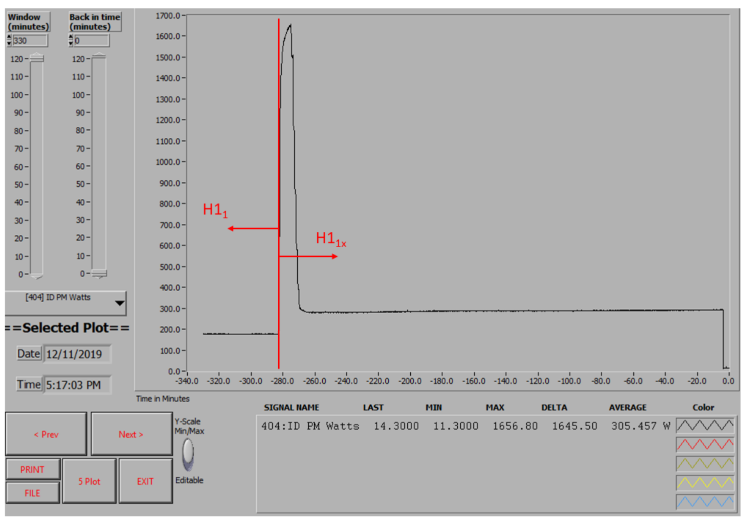



**Figure 13. Screen capture of power plot during the transition from H1$_1$ test to H1$_{1x}$ test**

## 4.10 Manufacturer F System 8

System 8 consists of a 36,000 Btu/h, central-ducted fan coil unit and a 36,000 Btu/h OD unit. The system was installed with 25 ft of refrigerant lines connecting the ID and OD unit with a 3/8 in. diameter liquid line and 7/8 in. diameter vapor line. Although this system uses an variable frequency drive compressor, it is programmed to run at discrete speeds based on inputs from a conventional 2-stage thermostat. The staging is based on thermostat inputs as shown in Table 16.

The control of staging for a specific set of thermostat inputs is based on runtime duration of the current cycle, the stage of the previous cycle, and OD air temperature. If the unit satisfies the call for cooling or heating quickly (in less than 20 min), the unit will stage down for the next cycle. If during a cycle the unit runs for over 45 min in cooling or 30 min in heating, the unit will stage up. If the unit staged up during the prior cycle, the entire next cycle will be at the stage of the prior cycle. With this type of control, the unit had no direct feedback on the ID air temperature or set point; it is only provided with requests for high or low stage operation. Therefore, obtaining near steady-state data during a load-based test is difficult or impossible. As a workaround, the dynamic ($_x$) tests for cooling were run using low or high stage thermostat inputs with the ID and OD chambers controlled to the desired conditions. The unit was cycled off prior to staging up, and this initial cycle was used to precondition the chamber and allow it to stabilize. For cooling, the next cycle was used for data collection with the last 30 min of the 45 min runtime in the desired stage (the unit staged up after 45 min) being used for calculations. Because the stage up time for heating was only 30 min, a different strategy was needed. For the nominal speed test, the Y2 (high stage compressor signal) was briefly de-energized, which reset the 30 min stage-up timer and was brief enough to not significantly affect the steady-state operation of the unit. For the minimum speed test, a method for controlling the unit to provide 30 min of steady-state data was not identified. The data for the $H1_{1x}$ test reflect a shorter data collection interval as noted in Table 17.

When trying to achieve the 0.58 in. water column (w.c.) ESP for the initial $A_2$ test, the behavior of the ID blower motor made reaching the desired static pressure impossible. As shown in Figure 14, the blower motor apparently changed speeds in response to changes in the system pressure. When the blower motor and code tester were turned on initially, the ESP was too high. As the code tester speed was increased to decrease the ESP, the blower power and airflow increased, indicating a constant speed programming. At approximately data point 140, there is a drastic drop in blower power, ESP, and airflow, indicating that the blower motor changed speed. Because the ESP was near 0 in. w.c., the code tester was slowed to increase the ESP again to try and achieve 0.58 in. w.c. As the code tester speed was decreased, the ESP gradually increased to around 0.26 in. w.c. However, as the code tester speed was decreased further, the ESP, blower power, and airflow began to rise sharply from data point 313 and on. Results were similar with different blower dip switch settings, with the system not being able to run in the middle of the ESP range. To overcome this issue, an external signal generator was used to provide the pulse width modulation signal to the blower motor directly, bypassing the ID units blower control board.

Additional ID fan-only tests were run on this unit to map the fan performance. For these tests, the power measurement was made on the supply power wires running directly to the blower motor. The pulse width modulation control signal was provided by a separate signal generator, so the power use of the onboard control typically used to generate this control signal was not included in the power measurement. Data from these tests are summarized in Table 18.

**Table 16. Compressor staging based on thermostat inputs**

| Thermostat signal | Available compressor stages |
|---|---|
| Y1 + O | Cooling low |
| Y1 + O | Cooling intermediate |
| Y1 + O | Cooling low high |
| Y1 + Y2 + O | Cooling nominal high |
| Y1 + Y2 + O | Cooling boost |
| Y1 | Heating low |
| Y1 | Heating intermediate |
| Y1 | Heating low high |
| Y1 + Y2 | Heating nominal high |
| Y1 + Y2 | Heating boost |

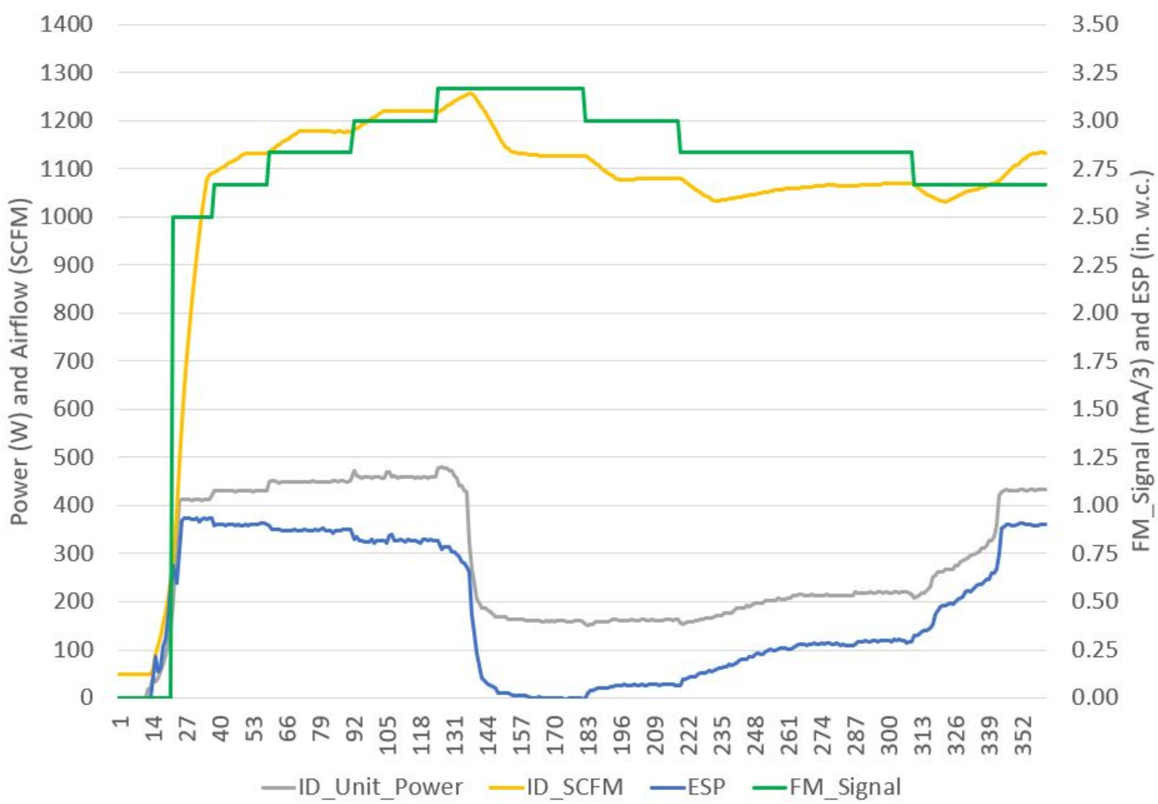



**Figure 14. Recorded data while trying to achieve 0.58 in. w.c. ESP using the system's blower control board**

**Table 17. System 8 test results**

| Test name | Capacity (Btu/h) | Power (W) | EER/COP (Btu/Wh or W/W) | ID airflow (scfm) | Compressor speed (Hz) | OD fan rpm | ESP [a] | Notes |
|---|---|---|---|---|---|---|---|---|
| $A_2$ | 34,984 | 2,795 | 12.52 | 1,203 | 48.4 | 1,055 | M | |
| $A_2$ | 34,516 | 2,902 | 11.89 | 1,205 | 48.4 | 1,055 | M1 | |
| $A_{2x}$ | 34,235 | 2,906 | 11.78 | 1,211 | 48.4 | 1,057 | M1 | |
| $E_v$ | 26,365 | 1,440 | 18.31 | 806 | 33.3 | 791 | M | |
| $E_v$ | 26,219 | 1,470 | 17.84 | 806 | 33.3 | 792 | M1 | |
| $E_v$ | 26,518 | 1,511 | 17.55 | 999 | 33.3 | 792 | M | b |
| $E_v$ | 26,286 | 1,566 | 16.78 | 999 | 33.3 | 792 | M1 | b |
| $B_2$ | 37,982 | 2,364 | 16.07 | 1,204 | 48.4 | 1,055 | M | |
| $B_2$ | 37,438 | 2,467 | 15.18 | 1,207 | 48.4 | 1,055 | M1 | |
| $B_1$ | 21,216 | 965 | 21.99 | 801 | 25.2 | 601 | M | |
| $B_1$ | 21,088 | 993 | 21.24 | 802 | 25.2 | 602 | M1 | |
| $B_{1x}$ | 21,019 | 998 | 21.06 | 802 | 25.2 | 603 | M1 | |
| $F_1$ | 23,257 | 708 | 32.84 | 804 | 25.2 | 603 | M | |
| $F_1$ | 23,122 | 736 | 31.42 | 804 | 25.2 | 602 | M1 | |
| $H0_1$ | 21,533 | 1,067 | 5.92 | 801 | 25.2 | 607 | M | |
| $H0_1$ | 21,596 | 1,098 | 5.76 | 801 | 25.2 | 607 | M1 | |
| $H1_N$ | 35,326 | 2,715 | 3.81 | 1,200 | 51.3 | 1,059 | M | |
| $H1_N$ | 35,592 | 2,819 | 3.70 | 1,199 | 51.4 | 1,059 | M1 | |
| $H1_{Nx}$ | 35,737 | 2,821 | 3.71 | 1,194 | 51.4 | 1,058 | M1 | c |
| $H1_1$ | 17,518 | 1,050 | 4.89 | 803 | 25.2 | 604 | M | |
| $H1_1$ | 17,544 | 1,081 | 4.76 | 803 | 25.2 | 605 | M1 | |
| $H1_{1x}$ | 17,581 | 1,075 | 4.80 | 810 | 25.2 | 605 | M1 | d |
| $H2_v$ | 18,974 | 1,498 | 3.71 | 812 | 34.2 | 791 | M | |
| $H2_v$ | 18,800 | 1,527 | 3.61 | 816 | 34.2 | 794 | M1 | |
| $H2_v$ | 18,621 | 1,485 | 3.67 | 1,005 | 34.2 | 791 | M | b |
| $H2_v$ | 18,850 | 1,546 | 3.57 | 1,005 | 34.2 | 791 | M1 | b |
| $H3_2$ | 22,549 | 2,385 | 2.77 | 1,198 | 51.4 | 1,058 | M | |
| $H3_2$ | 22,902 | 2,494 | 2.69 | 1,197 | 51.4 | 1,058 | M1 | |
| $H4_2$ | 18,520 | 2,257 | 2.40 | 1,204 | 51.4 | 1,058 | M | |
| $H4_2$ | 18,818 | 2,360 | 2.34 | 1,204 | 51.4 | 1,058 | M1 | |
| $H4_x$ | 21,169 | 2,695 | 2.30 | 1,199 | 58.4 | 1,095 | M1 | e |

[a] Indicates whether the external static pressure was set according to Appendix M or Appendix M1.

[b] The system was run at a higher intermediate airflow for these tests. The airflow target was 1,000 scfm instead of 805 scfm.

[c] In heating mode, the system stages up from nominal to boost after 30 min of operation. To record 30 min of near steady-state data, the Y2 (high stage compressor) call was briefly de-energized, resetting the 30 min timer but not upsetting the operation of the unit.

[d] There was no way to operate the unit so that it stayed in low speed operation for more than 30 min. The data recorded reflects 23.5 min of data at the end of a 30 min cycle.

[e] The system was running at "boost" compressor speed for this test.

**Table 18. System 8 ID fan test results**

| Reference ESP (in. w.c.) | Airflow (scfm) | ESP (in. w.c.) | Fan power (W) |
|---|---|---|---|
| 0.23 | 1,202.4 | 0.231 | 230.4 |
| | 1,003.4 | 0.161 | 139.2 |
| | 804.8 | 0.104 | 75.6 |
| 0.38 | 1,201.9 | 0.382 | 274.8 |
| | 1,000.8 | 0.264 | 163.2 |
| | 805.9 | 0.173 | 88.8 |
| 0.48 | 1,200.2 | 0.483 | 304.8 |
| | 1,003.6 | 0.334 | 182.4 |
| | 807.7 | 0.218 | 97.2 |
| 0.58 | 1,199.4 | 0.583 | 334.8 |
| | 1,000.5 | 0.404 | 199.2 |
| | 803.2 | 0.263 | 104.4 |

# 5. REFERENCES

## References

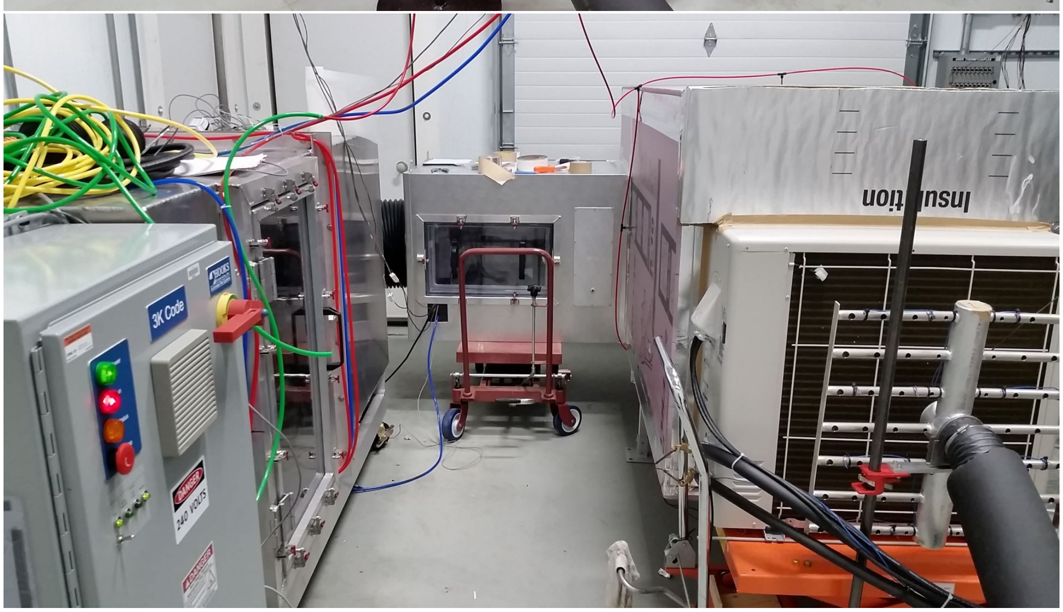

[1] Energy Information Administration. 2009 residential energy consumption survey, 2013.

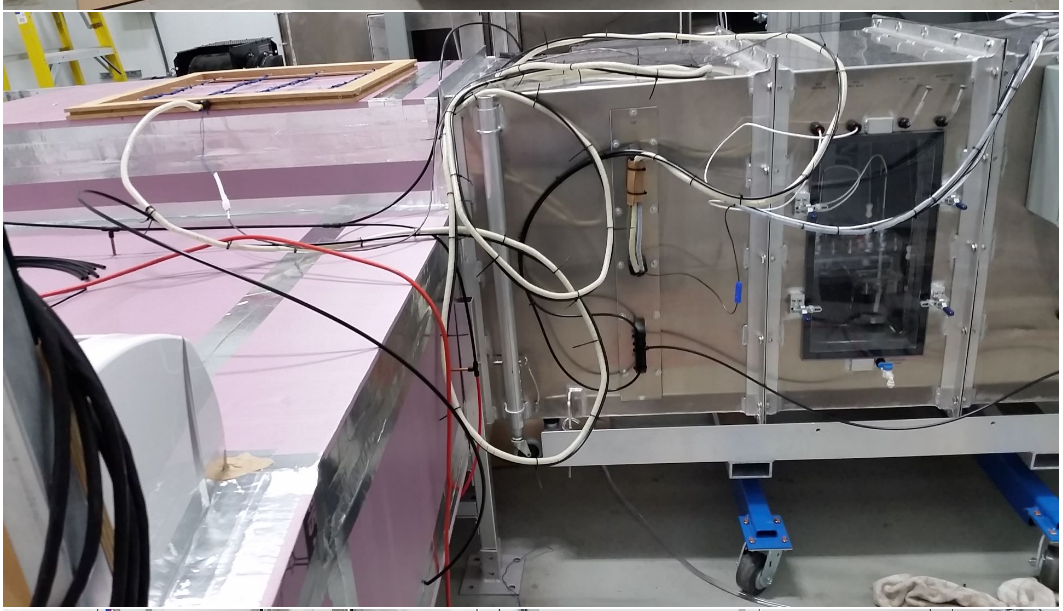
[2] Energy Information Administration. 2015 residential energy consumption survey, 2018.

[3] American Society of Heating, Refrigerating, and Air-Conditioning Engineers. *Standard 37-2009: Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment*. ASHRAE, Atlanta, Georgia, 2009.

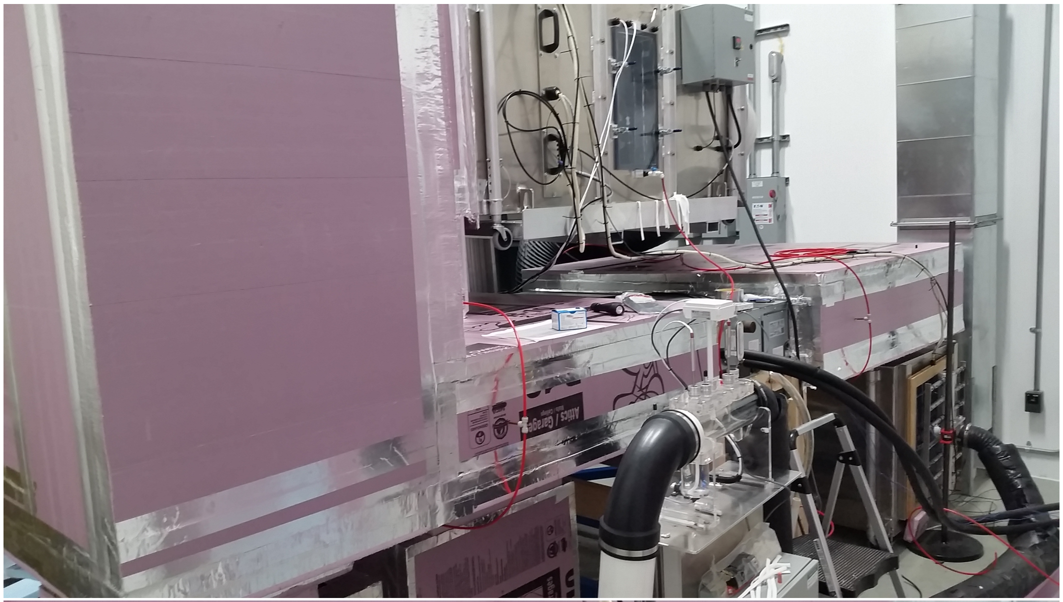
# APPENDIX A. PICTURES OF EQUIPMENT SETUPS

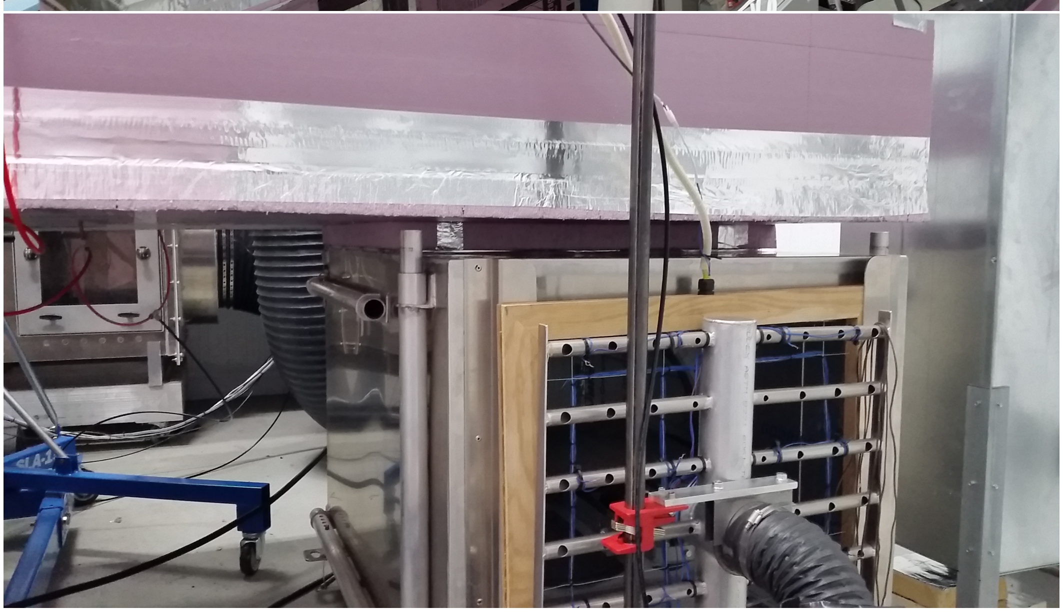

Streamline
LINE SETS

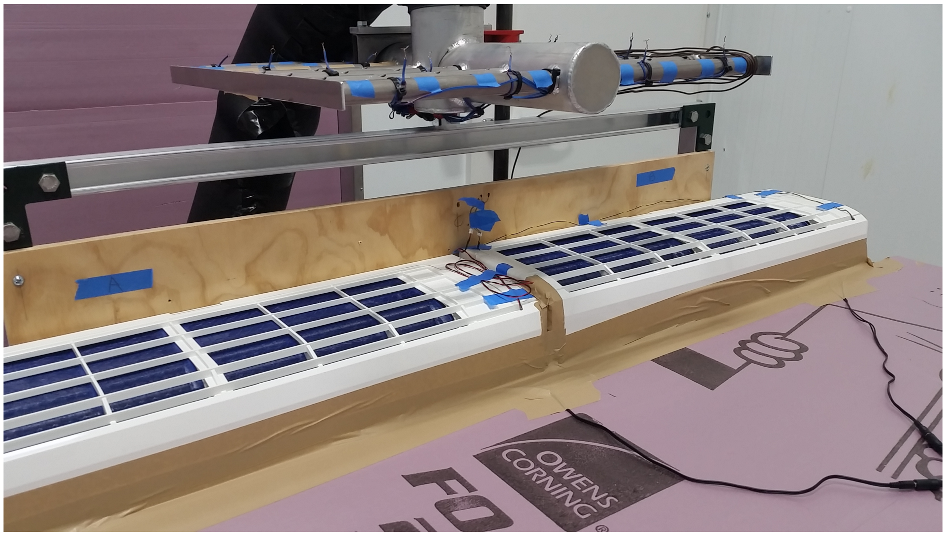

3K Code
DANGER
240 VOLTS



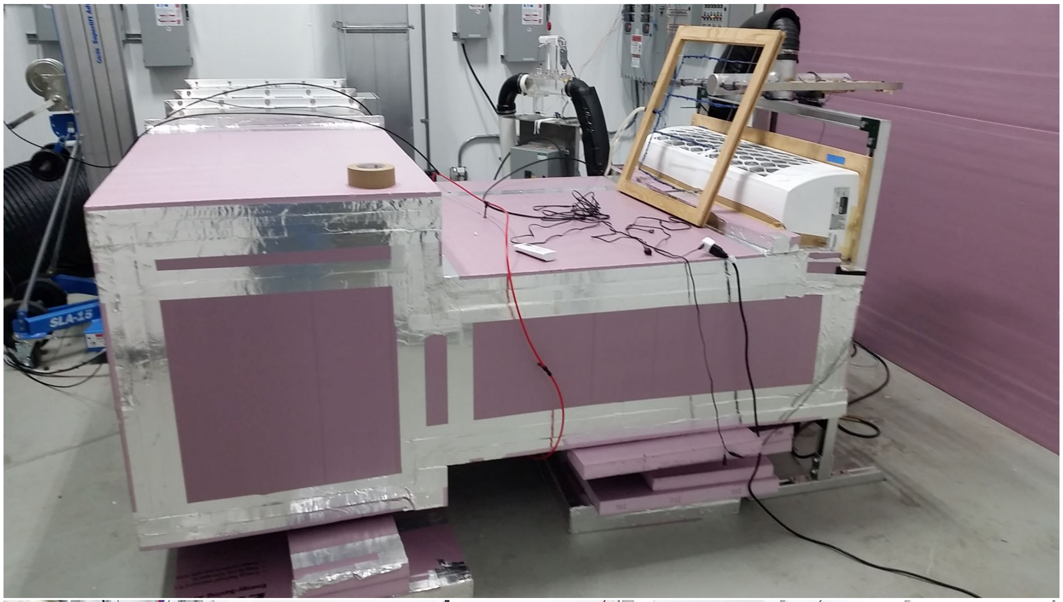

Insulation

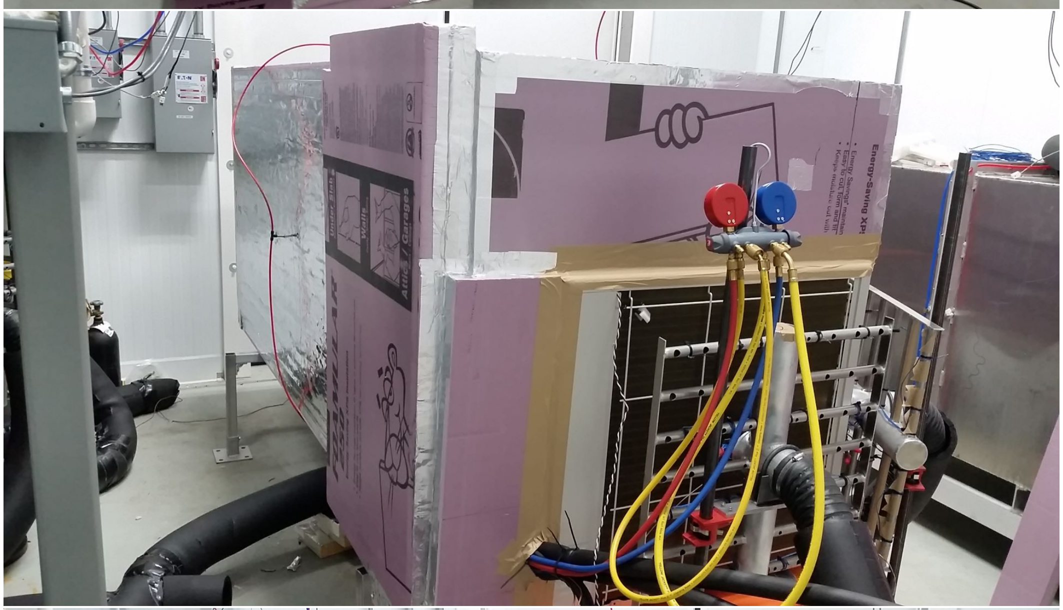

Attics / Garages

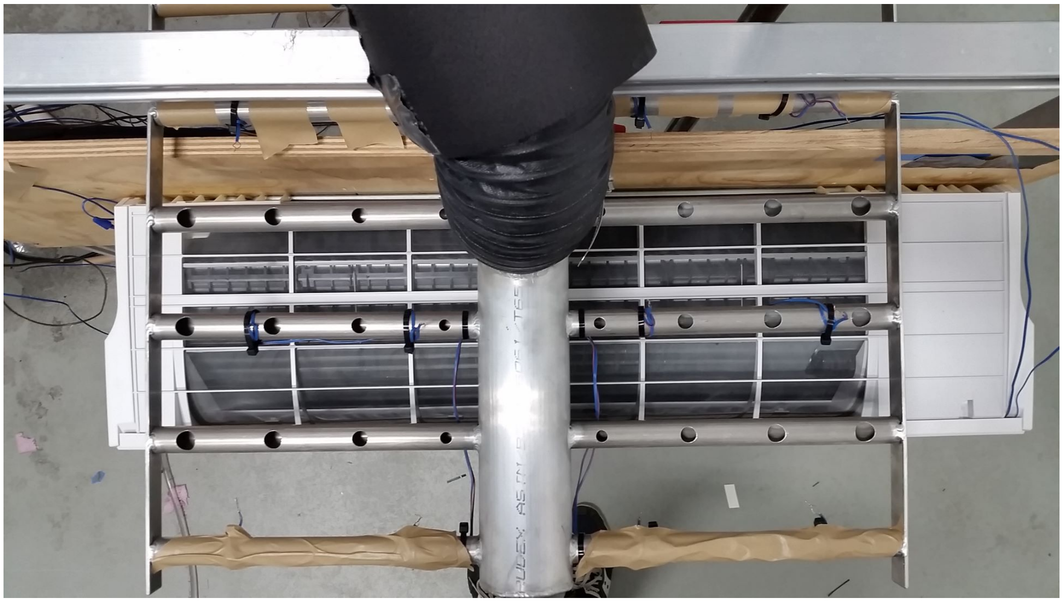


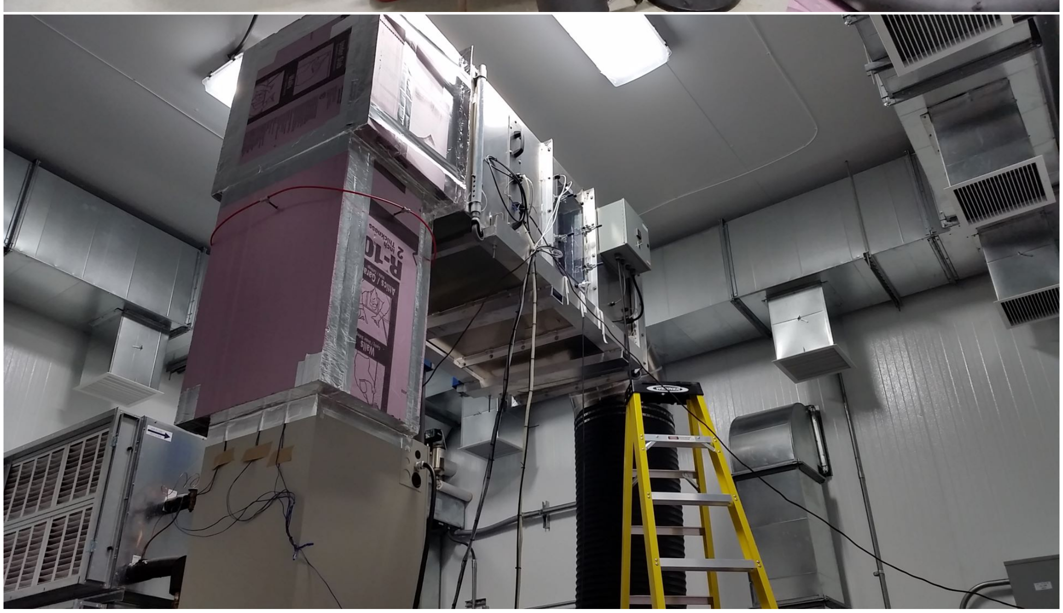

OWENS
CORNING


BOHN










Walls
Attics / Garages
R-10