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Field tests of variable speed heat pumps to compare load-based and fixed-speed test and rating methods

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Abstract

The improved performance of variable-speed heat pump systems has resulted in their widespread adoption globally. Heat pump rating methods such as AHRI 210/240-2023 (2020) were originally developed to characterize fixed-speed heat pump performance. Recently, load-based testing methods such as CSA EXP07:19 have been developed as a response to evidence that such fixed-speed tests don't represent realistic performance of variable-speed systems. EXP07 laboratory test results differ significantly from AHRI 210/240 ratings, yet neither approach has been validated as representative of field performance. This paper describes a field test in which six heat pumps are installed in three unoccupied test houses in Lincoln, Nebraska (a climate with a wide range of weather conditions, with heating and cooling design temperatures of -17°C and 34°C respectively. Each house has separate ducted and ductless systems, instrumented to collect performance data; the systems alternate operation weekly. After field testing, the same units will be tested in a certified laboratory using both AHRI 210/240 and an updated load-based method, SPE-07:23 facilitating comparison between the laboratory and in-situ performance. This will allow a fair assessment of the representativeness of each rating method.

Keywords: Heat pump; rating; variable speed; load-based testing; field test; representativeness; EXP07; SPE07

1. Introduction

Interest in efficient, variable speed air conditioners and heat pumps has gained momentum around the world as efforts to decarbonize building systems increase and performance of variable-speed systems continues to improve. In many markets, however, air source heat pumps have a lingering negative reputation among contractors and consumers, particularly in cold climates where historically poor sizing and installation practices have led to low efficiency [1-2]. Inaccurate efficiency ratings increase the potential for poor product choices, disappointed customers, and unexpected high bills. When a contractor or consumer chooses a product based on an efficiency rating, they should reasonably expect that the ratings will realistically represent that product, especially when comparing efficiency ratings of similar products. The impact of unpredictable performance ratings can reinforce existing bias, reflect negatively on utility and other publicly supported market transformation programs, and offers little motivation for manufactures to improve product performance. A 2019 study conducted for the Northwest Energy Efficiency Alliance [3] suggests that when a test procedure is updated to better represent current equipment, new energy savings can be gained from "better characterizing the energy use of the product." The savings result from eliminating the less-reliable ratings that mischaracterize units as highly efficient, and the authors conclude that improving heat pump ratings (specifically the use of EXP07) is a high priority representing a high magnitude of new energy savings.

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In 2015 the Canadian Standards Association (CSA) formed a development committee to develop testing and rating procedures that would better represent installed performance of variable capacity heat pumps (VCHPs). The relevance of Heating Seasonal Performance Factor (HSPF) and Seasonal Energy Efficiency Ratio (SEER) ratings [4] as realistic performance metrics to represent savings was increasingly called into question, and concerns included substantial variations in equipment performance when installed in climates that differ substantially from those used for the ratings. Further, in-field monitoring consistently suggested that current ratings do not predict installed performance well [5-6-7-8]. Many utilities and state/provincial agencies are increasing their market transformation efforts and funding to promote efficient HVAC installations, and are increasingly motivated to find rating metrics that reduce investment risk and improve evaluation results.

The development committee focused on variable-speed equipment, which depends on on-board firmware to operate, and for which field-measured performance has appeared to vary the most from published ratings. The result of this effort is a test procedure that includes the effects of on-board control algorithms and a wide range of outdoor conditions, driving a performance metric across a wide range of climates. CSA published EXP07:19 [9] as a technical review document; the updated version of the procedure based on technical review comment resolution has been published by CSA as SPE-07:23 [10]. Significant improvements have been incorporated with the primary intention of improving repeatability and reproducibility, yet it is expected that results of testing using SPE-07:23 will be generally similar to those using EXP-07:19.

Initial results of lab testing on numerous models using EXP07:19 have varied significantly from those units AHRI ratings of SEER and HSPF [11-12], with large variability in the load-based test results even among models with identical AHRI ratings, and very different relative rankings in relative efficiency levels for both heating and cooling performance. The load-based testing inherently includes the built-in control algorithms of variable-speed systems, whereas conventional tests require the use of "locked" or fixed-speed compressor and fan modes. The built-in controls appear to play a large role in the discrepancies in at least some cases. Although the load-based rating procedure has primarily been promoted for voluntary use (e.g. for qualified product lists and to differentiate high-performing products), stakeholders have suggested that better understanding of test-to-test repeatability, lab-to-lab reproducibility, and lab-to-field representativeness is needed to fully characterize the differences between load-based and fixed-speed testing.

Fixed-speed testing has an advantage in the rating process because steady-state operation makes measurements easier and reduces uncertainty. Load-based testing is conducted under "quasi-steady state" conditions, and depends on the behavior of built-in control algorithms, so uncertainty in the results will likely always be higher and repeatability lower. The high degree of lab familiarity with traditional fixed-speed tests (and by contrast, unfamiliarity with load-based methods) increases the potential for misunderstandings or diverse interpretations of testing procedures that can lead to discrepancies in results of load-based tests. An initial evaluation of repeatability and reproducibility [13] has shown that there is significant room for improvement. At the same time, many of the concerns raised by participants and stakeholders have been addressed during the comment resolution process leading to the updated SPE-07:23. Further study of repeatability and reproducibility using SPE-07:23 will be an important step in understanding these improvements and characterizing the differences between SPE-07 and more traditional test and rating methods. The focus of this paper is a further study, currently underway, that will focus on the representativeness of load-based and fixed-speed test methodologies to *in situ* field performance.

The authors of this paper are unaware of any systematic field validation of this nature that have previously been undertaken on any past versions of AHRI 210/240 and its precursors over several decades. Several studies have examined factors that introduce bias in the SEER and HSPF ratings [7-14-15]. Field studies and program evaluations generally cannot be taken as a direct comparison to standardized efficiency ratings. Many have found performance that varies from rated values, but such results can be influenced by numerous external factors including occupant behavior, indoor air conditions, duct leakage, load differences, and installation practices such as air flow and refrigerant charge. Recently, with the 2023 edition of AHRI 210/240 [16], an effort has been made to correct assumptions regarding the heating load lines used in different climates (related to both building thermal performance and equipment sizing assumptions) for both single and variable-speed equipment, and the assumed static pressure of duct systems has been increased from an unrealistically low 50 Pa (0.2 in.w.c) to 125 Pa (0.5 in.w.c) for most common ducted system types. These changes will result in new metrics of HSPF2 and SEER2, that will generally be numerically lower than those of HSPF and SEER. Both of these changes should reduce systematic bias in the new metrics, making them incrementally more representative of field conditions. However, the continued use of fixed-speed testing that ignores built-in equipment control algorithms is likely to miss some important performance variations.

Although the lab tests using CSA EXP07 revealed substantial differences between the fixed-speed ratings and load-based tests [12] and provided some evidence that these differences were largely driven by the behavior of built-in equipment control algorithms in the load-based testing, to date there has been no clear proof that load based testing results in ratings that better approximate real-world performance. This paper describes an effort to address this gap.

2. Representativeness Study Overview

The study is divided into two phases, a field segment (Phase I, currently underway) and a laboratory segment (Phase II). Phase I studies the performance of six heat pumps installed the field in a simulated occupancy home environment. Phase II will test the same heat pumps in an accredited lab, using both SPE-07:23 and AHRI 210/240: 2023 [16]. Comparing the test and rating results will provide insight into which method (if either) more closely matches the performance of the actual systems in the field. Equally important, the detailed field monitoring and lab test results may provide insight into what aspects of performance drive the differences between the rating methods, as well as differences between both methods and the field performance. The intent is to examine the differences between the rating methods and provide insight into how either or both methods might be improved to represent real-world performance more closely. Ratings are always models based on limited testing and assumed applications, and as such are inherently simplifications; they are typically used for purposes such as comparing different products, roughly predicting product performance, and demonstrating regulatory and/or voluntary program compliance. But systematic biases and other significant errors in rating methodologies have the potential to dilute and confuse those purposes.

Phase I is being conducted in Lincoln, NE in a residential neighborhood using three identical, unoccupied mobile homes as test houses. Each test house is outfitted with two heat pumps, one ducted and the other ductless, that are being used in alternate weeks to heat or cool each house. The heat pumps were installed and set up to run in cooling mode, with data collection beginning in mid-August 2022, and were subsequently changed over to heating mode in late October. In March 2023 the tested systems will be uninstalled and shipped to the laboratory for Phase II. This time span will include a wide range of load conditions for both heating and cooling operation.

During Phase II, the laboratory will conduct the test procedure using a modified version of SPE-07:23. The modifications will be to approximate both the load line and thermal mass capacitance of the actual houses as measured during Phase I, so that the test method can be calibrated to the buildings used in the study. The laboratory will also conduct a set of heating and cooling tests from AHRI 210/240-2023, with manufacturer cooperation to access any required proprietary test modes. The lab testing may include additional testing such as the optional H_{4Full} test [16] at -15 °C (5 °F), the Min/Mild and/or the H4_Max/Cold tests as detailed in the DOE cold climate challenge specification [17].

3. Field Study Method

3.1. General approach

Each test house has two heat pumps installed; operation is alternated between the two systems approximately once each week, to capture a range of operating conditions for each. The test houses have been calibrated to approximately match the heat transfer and thermal mass characteristics of SPE-07. The test houses are operated with internal gains, and latent internal gains during cooling, to approximate occupied conditions. Detailed measurements of heat pump operating parameters are recorded, as well as conditions in the house, and outdoor conditions at the site, allowing measurements of equipment capacity and input power over the entire range of operating conditions at a 1-second time interval. All the recorded data is uploaded to a server for access and analysis.

3.2. Test house description and characterization

The test houses consist of three manufactured homes of identical design, built in 2021 at the same factory, sited in the same neighborhood and orientation to the compass. Each house has 113 m² floor area on one story (Figure 1), with low-slope vaulted ceilings and are insulated to current code requirements for manufactured homes. They are set on concrete blocks, raised off the ground level approximately 1 m. The existing heating system is an electric forced-air furnace of 15 kW nominal capacity, with a supply duct running the length of the "belly" (enclosed joist/truss area) and a single branch duct running to each room; a single return register is located in the utility room.

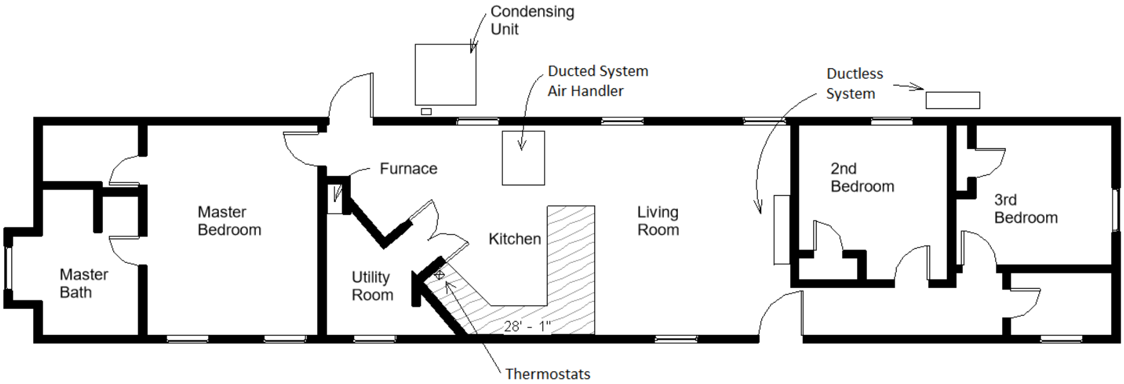

Fig 1: Plan View of Test House.

The houses were tested and calibrated prior to the test period, with enclosure thermal transfer (UA) and mass capacitance measured, as well as enclosure and duct leakage. Each home has one ducted and one ductless heat pump installed. The ductless heat pumps are high-wall mounted. The ducted heat pump air handlers are mounted off the floor, with return air at the bottom and a single supply duct that terminates inside the service closet near the return grille of the central system, as shown in Figure 2.

Fig. 2. Ducted Indoor Unit.

Each heat pump is controlled by a wall-mounted thermostat on an interior kitchen wall, near the entrance to the utility room. During the test period, the air handler fan of the pre-existing central furnace remains on, to promote air mixing. In addition, there are two ceiling fans, four box fans and two stand fans to mix the air throughout the house.

3.3. Test house calibration

The test method of SPE-07 relies on a model of the building load and shallow thermal mass typical of a residential building. The model is scaled to equipment capacity. The laboratory test mimics the thermal response of the modeled building as the equipment capacity varies during each load-based test condition. Measurements and modifications were made to bring the UA and mass capacitance of the test homes close to the SPE-07 lab test values. The test method of AHRI 210/140 has no built-in assumptions about either parameter, so no conflict results from targeting those of SPE-07.

The UA can be defined as:

$$UA = \frac{Q_{heat\ loss} - c}{\Delta T} \tag{1}$$

Where:

UA is the U-value times the area of the building in W/K

$Q_{heat\ loss}$ is the heat loss of the house in W,

ΔT is the difference between indoor and outdoor temperature in K, and

c is the temperature-independent heat loss (internal gains)

To estimate the UA of the test houses, the houses were heated with electric resistance heaters at night to reduce transients and solar effects. By measuring the energy input and indoor/outdoor temperature differences, the UA can be estimated as the slope of a linear regression of those two variables. Figure 3 shows an example of a UA calculation for one house that is tested over a period of 26 nights.

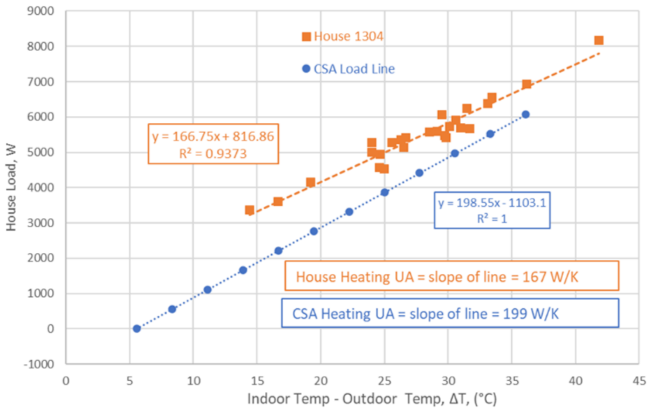

Fig. 3. Plot of Target CSA UA and Measured House UA.

The target UA for SPE-07 based on equipment nominal cooling capacity of 5.2 kW is 199 W/K. Measurements of the three houses ranged from 160 to 169 W/K, so modifications were made to increase the heat transfer. Some insulation and air barrier materials were removed from the belly (crawlspace) of the homes, and in each house four double-low-e glazing panels were replaced with sheet galvanized steel.

The target thermal capacitance for SPE-07 for a 5.2 kW capacity heat pump is 317 Wh/K. The capacitance of the houses was estimated by using the preexisting electric furnace and measuring the cycling rate of the furnace under a range of heating load conditions. The capacitance [18] is then calculated using:

$$C_s = \frac{Q_{h,s,D}}{4 \cdot N_{max} \cdot \Delta T_{db}} \tag{2}$$

Where:

C_s is the thermal capacitance in Wh/K,

$Q_{h,s,D}$ is the heater capacity in W,

N_{max} is the maximum cycling rate in hours^{-1}, and

ΔT_{db} is the thermostat deadband in K

N_{max} can be determined using the following relationship [18]:

$$N = 4 \cdot N_{max} \cdot X \cdot (1 - X) \tag{3}$$

Where X is the run time fraction, which is the heat on time divided by total cycle time.

By logging the behavior of the on/off heating cycles, as shown in Figure 4, across a range of load conditions (changing outdoor temperatures), the value of N_{max} can be determined empirically for each house as the slope of the linear regression formed by the cycling rates N vs the product of the run fraction and its complement $X \cdot (1 - X)$, for each cycle, divided by 4. The plots for each test house are shown in Figure 5. The coefficients of determination R^2 are 0.96, 0.96 and 0.98 respectively for House 1, 2 and 3, indicating a strong correlation.

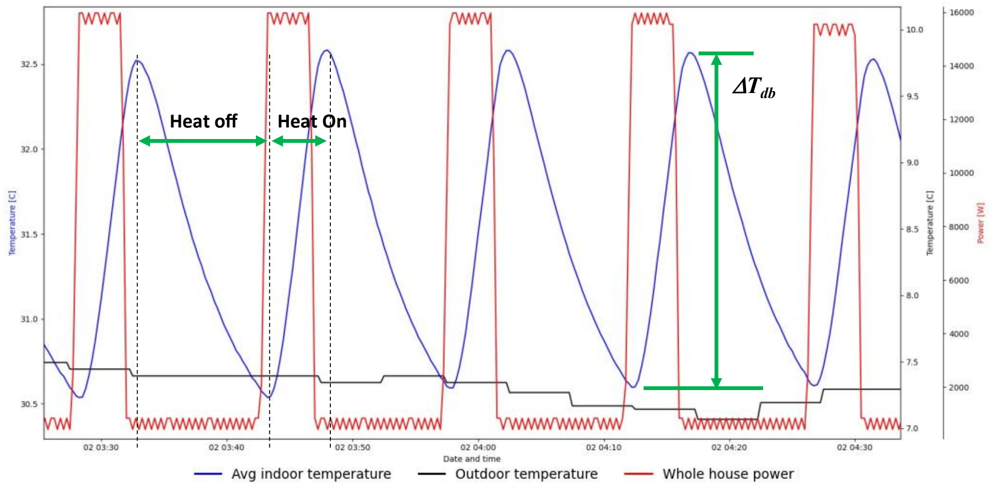

Fig. 4. Measurement of Cycle Times and Thermostat Deadband to Calculate Thermal Capacitance.

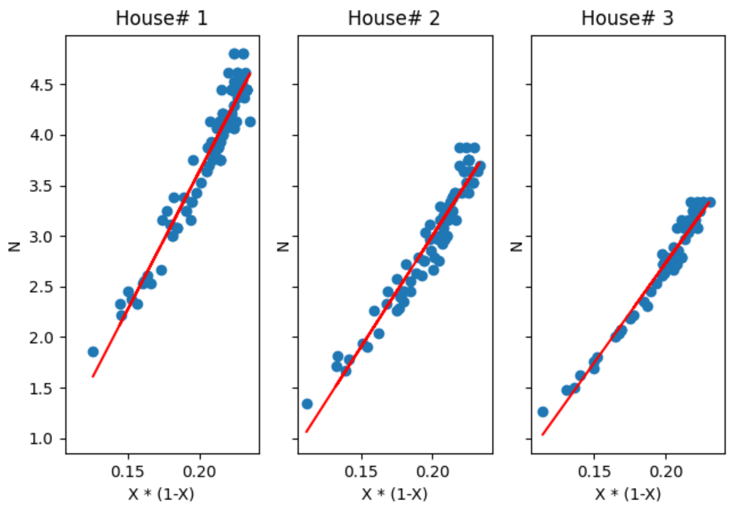

Fig. 5. Plots of Cycling Rate.

The value of C_s for the three houses were estimated to be 282, 296, and 298, slightly lower than the target of 317, and N_{max} was estimated between 4.9 and 6.8 for the three houses. In order to increase the shallow mass capacitance and increase the cycle time, 16 sheets of drywall were added to each house, divided into three rooms, and standing vertically and separated to increase surface exposure (Figure 6). The 16 sheets were estimated based on a comparison of interior surface area of a house contained within a residential home testing laboratory facility, which had been previously used to conduct validation testing and compare mass capacitance response using EXP07:19 [19].

The project schedule did not allow repeating the measurements with the final modifications in place, but data adequate to estimate the building heating and cooling loads and the equipment cycling behavior are being captured throughout the test period, so a more complete picture of the houses' dynamic response will be understood by the end of the test period. These parameters in the SPE-07 lab tests during Phase II will then be matched to the measured houses' performance.

The airtightness of each house was measured with a blower door, with results ranging from 3.1 to 3.9 air changes per hour at 50 Pa. The leakage impact on heating and cooling load is integrated into the overall UA value of each house and the houses are tight enough that wind effects should be small.

Fig. 6. Drywall Used to Increase Thermal Capacitance

3.4. Heat pump selection

The design heating load was estimated at 5.5 kW at the heating design temperature of -15 °C. This was done by analyzing billing data of identical homes, and with standard load calculations. The design cooling load was calculated to be 5.2 kW at a design temperature of 35°C. Heat pump models from manufacturers who are engaged in the project were selected, so that the manufacturers will be able provide testing support during Phase II to ensure that the proprietary test modes required for AHRI 210/240 testing can be accessed properly. Five of the selected systems are variable-speed, and one (a ducted system) has a 2-stage compressor. The variable-speed systems are all considered “cold-climate” systems according to the NEEP cold-climate air-source heat pump listing specification [20], whereas the 2-stage system (Unit B in Figure 7) has a steeper drop of heating capacity at colder ambient conditions, typical of more conventional heat pumps. It was important to choose systems that are not significantly oversized, so that they will typically run at full speed operation near design conditions and modulate well across a range of heating and cooling load conditions. Figure 7 shows the systems’ heating and cooling capacities, compared with the estimated heating and cooling design loads. Because the UA measurements were done on the houses in advance, actual heating loads could be estimated with reasonable confidence, whereas cooling loads are driven by solar gain, which could not be measured in advance of test period.

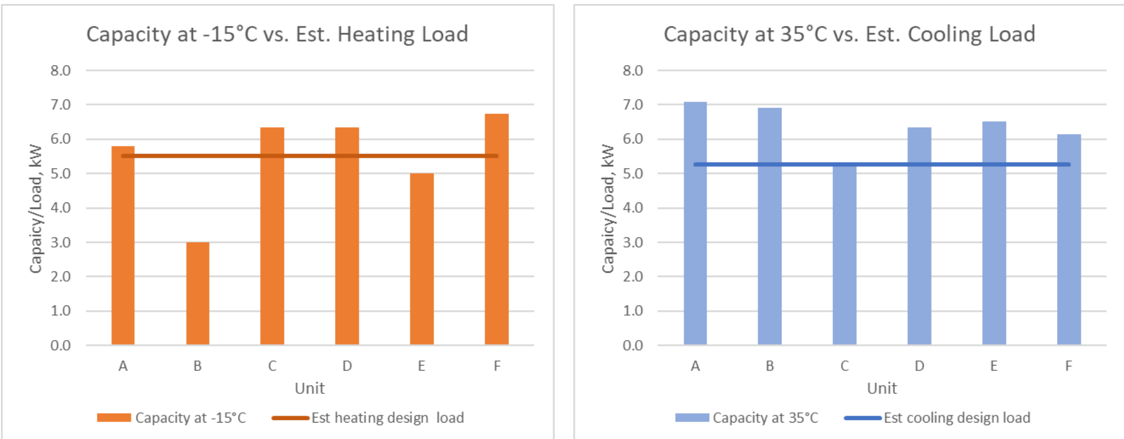

Fig. 7. Installed Heat Pump Heating and Cooling Capacities and Design Loads.

The heat pumps were installed carefully following manufacturers' specifications, with line lengths selected to require no adjustment to refrigerant charge amount. Line lengths were calculated to account for the interior volume of the mass flow sensors and bypass piping. Most systems' outdoor units came pre-charged, so that no refrigerant was added during installation. Two of the systems were charged at the site. Both had charge weighed in, and the charge amount was confirmed by comparing the liquid line subcooling measurement to the manufacturer specification.

Figure 8 shows a typical ductless outdoor unit installation, and one steel window panel, which was used as a location for mechanical penetrations.

Fig. 8. Typical Ductless Outdoor Unit Installation, Showing Steel Window Replacement.

3.5. Example test results

The systems were operated in cooling mode from 19 August and switched over to heating on 25 October, 2022, and are expected to run through early March 2023. The outdoor temperatures have ranged from 39.4 °C (103 °F) to -25 °C (-13 °F), providing a range of operating conditions that exceeds the normal design conditions. The units within each test house were alternated at intervals ranging from 3 days to a week. Figure 9 shows an example of two of the units operating from noon until 8:00 AM the following morning, during which a peak 39 °C ambient temperature occurred. The power to unit D (House 1) modulated over a wider range during the hottest hours, while unit F (House 3) appears to modulate more smoothly, and uses about 70% of the energy of unit D. Unit D appears to manage the supply air temperature within a narrow range when the cooling load is large. Analysis is still in progress, so calculated values (e.g., capacity) are not available at the time of writing.

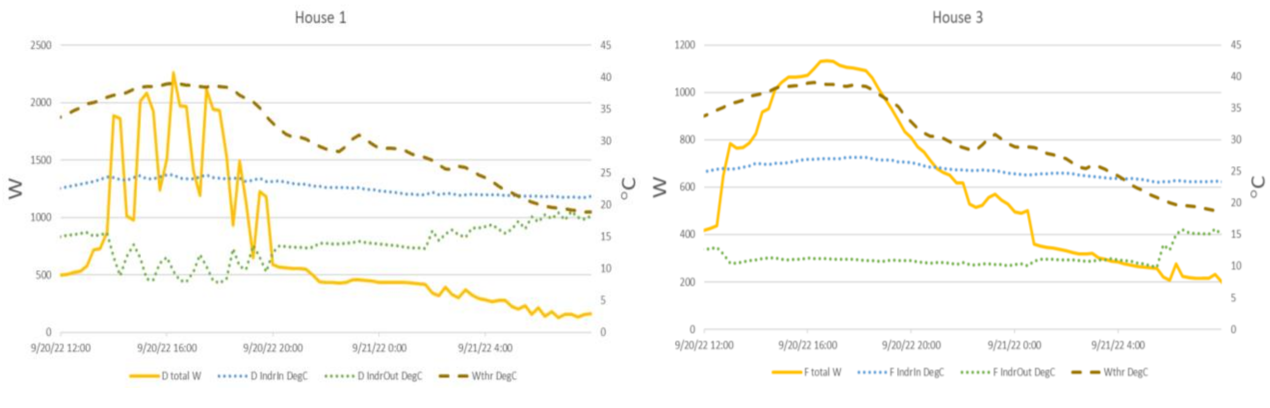

Fig. 9. Systems D and F in Cooling Mode During a 20-hour operating period with a 39 °C Peak Ambient Temperature.

3.6. Measurements and instrumentation

Each of the houses has almost 100 sensors connected to a data acquisition system (DAQ), which is cloud-connected, allowing real-time monitoring. The sensors provide a comprehensive set of measurements within the house, such as temperatures in important locations, and of the heat pumps. Electrical power is also monitored at end uses, including the heat pumps and air handlers, and for the house overall, so that internal heat gain is known. The set of monitored variables facilitates characterization of the houses' cooling and heating loads, as well as the capacity and efficiency of the heat pumps. Heat pump capacity is characterized both on the air-side (by measuring the air flow rate and the indoor coil entering and leaving temperature and humidity) and on the refrigerant side (by measuring refrigerant mass flow rate and characterizing enthalpy entering and leaving the indoor coil with temperature and pressure measurements). This provides redundant measurement of the key variable, heat pump capacity.

Coriolis mass flow sensors are used to measure refrigerant mass flow. Cutting into the liquid line was considered to be too invasive; for most systems it would involve substantial modification of the outdoor unit, and removal then replacement of the refrigerant. Instead, the mass flow sensors were installed in the line set, which carries vapor or mixed-phase refrigerant, depending on the mode of operation and the location of the expansion devices. Mass flow sensors can only measure single-phase refrigerant, so the sensors were located to provide vapor flow measurements.

Some systems do not have a pressure port that allows measurement of the high side pressure during cooling mode, which is needed to calculate liquid line enthalpy (that can be assumed to equal the enthalpy at the evaporator inlet). Because the outdoor unit could not be modified, additional pressure ports could not be added. In these systems, the pressure is inferred by measuring temperature at several locations in the condenser, finding the region of saturation, and using the saturation pressure at that temperature. Figure 10 shows two indoor units with the Coriolis mass flow sensors installed in the refrigerant lines near an indoor unit.

Fig. 10. Ducted (left) and Ductless (right) Indoor Units, Showing Refrigerant Mass-flow Sensors.

The houses each have temperature and humidity sensors measuring the air condition in several locations throughout the house. Each house also has temperature sensors on the: (1) interior surface of the drywall; (2) the outward-facing surface of the drywall; and (3) in the middle of the wall cavity. These trios of sensors are located on: (a) an interior wall; (b) a south facing wall; and (c) on a north-facing wall. A temperature sensor is also located at the location of the thermostats controlling the heat pumps. Outdoor conditions – temperature, humidity, wind and solar irradiance – are monitored with a weather station, and each outdoor unit is also outfitted with radiant-shielded temperature sensors in the air intake of the outdoor unit. Finally, condensate from the evaporator in cooling mode is measured with a tipping-bucket type sensor.

Table 1 gives a list of typical measurements in each house.

Table 1: List of measurements in each house

	Variable	Type	Description
1	W_dhp	Current transf.	Ducted heat pump power
2	W_dhp_i	Current transf.	Ducted indoor unit power
3	W_dlhp	Current transf.	Ductless heat pump power
4	W_dlhp_i	Current transf.	Ductless indoor unit power
5	W_house	Current transf.	Total house power
6	W_furnace	Current transf.	Furnace power
7	T_amb	RTD	Outdoor temp.
8	RH_amb	Capacitance	Outdoor humidity
9	WS_amb	Vane	Outdoor windspeed
10	WD_amb	Vane	Outdoor wind direction
11	p_amb	Piezoelectric	Atmospheric pressure
12	Q_solar	Si pyranometer	Solar irradiance
13	T_wall_ii	Thermocouple	Interior wall indoor surface temp.
14	T_wall_ib	Thermocouple	Interior wall back of drywall temp.
15	T_wall_ic	Thermocouple	Interior wall cavity temp.
16	T_wall_ni	Thermocouple	North wall indoor surface temp.
17	T_wall_nb	Thermocouple	North wall back of drywall temp.
18	T_wall_nc	Thermocouple	North wall cavity temp.
19	T_wall_si	Thermocouple	South wall indoor surface temp.
20	T_wall_sb	Thermocouple	South wall back of drywall temp.
21	T_wall_sc	Thermocouple	South wall cavity temp.
22	T_air_br1	RTD	BR1 space temp.
23	RH_air_br1	Capacitance	BR1 space humidity
24	T_air_br2	RTD	BR2 space temp.
25	RH_air_br2	Capacitance	BR2 space humidity
26	T_air_br3	RTD	BR3 space temp.
27	RH_air_br3	Capacitance	BR3 space humidity
28	T_stat	Thermocouple	Thermostat temp.
29	T_air_fur	Thermocouple	Furnace inlet air temp.
30	RH_air_fur	Capacitance	Furnace inlet air relative humidity
31	T_sa_dl1	RTD	Ductless supply air temp. 1
32	T_sa_dl2	RTD	Ductless supply air temp. 2
33	T_sa_dl3	RTD	Ductless supply air temp. 3
34	T_sa_dl4	RTD	Ductless supply air temp. 4
35	RH_sa_dl1	Capacitance	Ductless supply air humidity 1
36	RH_sa_dl2	Capacitance	Ductless supply air humidity 2
37	RH_sa_dl3	Capacitance	Ductless supply air humidity 3
38	RH_sa_dl4	Capacitance	Ductless supply air humidity 4
39	T_ra1	RTD	Ductless return air temp. 1
40	T_ra2	RTD	Ductless return air temp. 2
41	RH_ra_dl1	Capacitance	Ductless return air humidity 1
42	RH_ra_dl2	Capacitance	Ductless return air humidity 2
43	T_ref_mf	Thermocouple	Ductless MF sensor inlet temp.
44	T_ref_ei	Thermocouple	Ductless evaporator inlet temp.
45	T_ref_eo	Thermocouple	Ductless evaporator outlet temp.
46	T_ref_suc	Thermocouple	Ductless suction temp.
47	T_ref_tsuc	Thermocouple	Ductless suc. after rev. vlv temp.
48	T_ref_dis	Thermocouple	Ductless discharge temp.
49	T_ref_rvlv	Thermocouple	Ductless after rev. valve temp.
50	T_ref_c1	Thermocouple	Ductless condensing temp. 1
51	T_ref_c2	Thermocouple	Ductless condensing temp. 2
52	T_ref_c3	Thermocouple	Ductless condensing temp. 3
53	T_ref_c3	Thermocouple	Ductless condensing temp. 3
54	T_ref_ll	Thermocouple	Ductless liquid line temp.
55	m_ref1	Coriolis	Ductless Refrigerant mass flow
56	T_air_c1	RTD	Ductless cndnsr air in temp. 1
57	RH_air_c1	Capacitance	Ductless cndnsr air in rel humidity 1
58	T_air_c2	RTD	Ductless cndnsr air in temp. 2
59	RH_air_c2	Capacitance	Ductless cndnsr air in rel humidity 2
60	p_ref_suc	Piezoelectric	Ductless ref. suction pressure
61	p_ref_ll	Piezoelectric	Ductless ref. liquid line pressure
62	m_cond	Rain gauge	Evaporator condensate mass flow
63	V_ra	Orifice plate	Ducted indoor unit airflow
64	T_sa_d1	RTD	Ducted supply air temp. 1
65	T_sa_d2	RTD	Ducted supply air temp. 2
66	T_sa_d3	RTD	Ducted supply air temp. 3
67	T_sa_d4	RTD	Ducted supply air temp. 4
68	RH_sa_d1	Capacitance	Ducted supply air humidity 1
69	RH_sa_d2	Capacitance	Ducted supply air humidity 2
70	RH_sa_d3	Capacitance	Ducted supply air humidity 3
71	RH_sa_d4	Capacitance	Ducted supply air humidity 4
72	T_ra1	RTD	Ducted return air temp. 1
73	T_ra2	RTD	Ducted return air temp. 2
74	RH_ra_d1	Capacitance	Ducted return air humidity 1
75	RH_ra_d2	Capacitance	Ducted return air humidity 2
76	T_ref_mf	Thermocouple	Ducted MF sensor inlet temp.
77	T_ref_ei	Thermocouple	Ducted evaporator inlet temp.
78	T_ref_eo	Thermocouple	Ducted evaporator outlet temp.
79	T_ref_suc	Thermocouple	Ducted suction temp.
80	T_ref_tsuc	Thermocouple	Ducted suc. after rev. valve temp.
81	T_ref_dis	Thermocouple	Ducted discharge temp.
82	T_ref_rvlv	Thermocouple	Ducted after rev. valve temp.
83	T_ref_c1	Thermocouple	Ducted condensing temp. 1
84	T_ref_c2	Thermocouple	Ducted condensing temp. 2
85	T_ref_c3	Thermocouple	Ducted condensing temp. 3
86	T_ref_c3	Thermocouple	Ducted condensing temp. 3
87	T_ref_ll	Thermocouple	Ducted liquid line temp.
88	m_ref1	Coriolis	Ducted Refrigerant mass flow
89	T_air_c1	RTD	Ducted cndnsr air in temp. 1
90	RH_air_c1	Capacitance	Ducted cndnsr air in rel humidity 1
91	T_air_c2	RTD	Ducted cndnsr air in temp. 2
92	RH_air_c2	Capacitance	Ducted cndnsr air in rel humidity 2
93	p_ref_suc	Piezoelectric	Ducted ref. suction pressure
94	p_ref_ll	Piezoelectric	Ducted ref. liquid line pressure

4. Future Comparison of Lab and Field Performance

Properly comparing field performance and the two types of rating approaches requires that each system be run through the entire SPE-07 and AHRI testing process. From that process unofficial efficiency performance metrics can be calculated for each tested system, which may or may not correspond closely to the catalogued ratings. Further, because the load-based approach in SPE-07 inherently includes a virtual model of the building load and mass capacitance (scaled to the size of the tested system), this virtual model will be adjusted to match the actual test houses, to make the comparisons more meaningful. The initial characterization testing of the three houses showed that they were quite close to the SPE-07 virtual building model, providing an initial validation that the SPE-07 virtual building model is reasonable.

Finally, to compare the field- and lab-tested performance fairly, the field performance of each tested system will be characterized by a performance curve, primarily capacity and input power over a range of outdoor temperature conditions. That performance curve will then be fitted to the prototypical climates that have been defined for each test method, to create field equivalent performance metric results for each rating procedure, that can be compared with the unofficial efficiency performance metrics from the laboratory testing. The performance curve can also be fitted back to the actual weather from the field test sites, to determine how closely the simplified curve approximates the actual seasonal performance during the test period.

Besides simply making numerical comparisons of the unofficial rating results, the analysis will examine *where* in the testing processes there may be the largest discrepancies, and whether there are other factors that create systematic bias or other errors in one or both ratings methodologies.

5. Conclusions

This project is unique because it will provide field data that can be directly compared to multiple lab test methods. Having six heat pumps tested in nearly identical homes, under identical use patterns but without occupant behavior, enables researchers to evaluate how effective current and future test procedures are at generating representative ratings. Much of the lab testing to date [8-11-12] has shown that conducting tests under a heat pump's built-in algorithms sometimes reveals performance issues for variable speed and single speed heat pumps alike. As heat pumps increasingly become microprocessor controlled, it is imperative that we learn what conditions are most sensitive to the effects of built-in control algorithms. This research project will be able to inform improvements that can be made to both load-based and fixed-speed test methods, to provide ratings that better inform contractors and end users which product will perform better for their climate conditions. In addition, more representative test procedures can lead to better input parameters for simulations that are used for electrical utility forecasting and grid resilience, energy code compliance, engineering analysis and system designs.

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