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## Experimental Evaluation of High Performance Integrated Heat Pump

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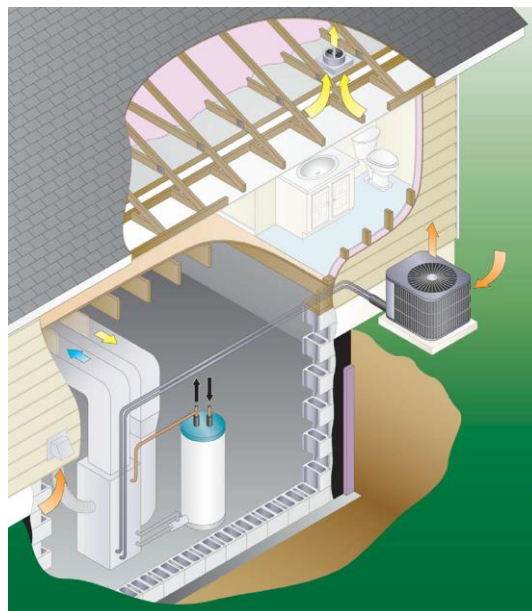
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### ABSTRACT

Integrated heat pump (IHP) technology provides significant potential for energy savings and comfort improvement for residential buildings. In this study, we evaluate the performance of a high performance IHP that provides space heating, cooling, and water heating services. Experiments were conducted according to the ASHRAE Standard 206-2013 where 24 test conditions were identified in order to evaluate the IHP performance indices based on the airside performance. Empirical curve fits of the unit's compressor maps are used in conjunction with saturated condensing and evaporating refrigerant conditions to deduce the refrigerant mass flowrate, which, in turn was used to evaluate the refrigerant side performance as a check on the airside performance. Heat pump (compressor, fans, and controls) and water pump power were measured separately per requirements of Standard 206. The system was charged per the system manufacturer's specifications. System test results are presented for each operating mode. The overall IHP performance metrics are determined from the test results per the Standard 206 calculation procedures.

### 1. INTRODUCTION and BACKGROUND

An IHP simply defined is a heat pump system that combines (or integrates) multiple functions within one system. At a minimum, space conditioning (space heating (SH), and cooling (SC)) and domestic water heating (WH), services would be combined. Additional building energy services, including ventilation (V) and dedicated dehumidification (DH) could be incorporated depending on how the IHP system is designed. Prior investigations of the IHP concept have estimated that it can achieve up to 50% or more annual energy savings vs. current minimum efficiency heating, ventilation, air-conditioning and water heating (HVAC/WH) equipment, particularly for newer or renovated buildings with high efficiency (low load) and low air leakage thermal envelope systems. (Murphy et al, 2007; Rice, et al 2014a, 2014b) Both air-source (AS-IHP) and ground-source (GS-IHP) embodiments of IHP systems are possible. This paper focuses on a laboratory evaluation of an AS-IHP prototype using the test procedures outlined in ASHRAE Standard 206-2013. (ASHRAE 2013) Figure 1 illustrates an installation approach for one possible AS-IHP design.



**Fig. 1. Conceptual installation of a residential air-source integrated heat pump.**

The increasing thermal efficiency and leak-tightness of new homes and deeply renovated homes results in reduced SH and SC demands while the WH load, which depends largely on the number of occupants in the dwelling and their life styles, remains essentially unchanged. Consequently, the WH load becomes a larger portion of the total annual load on the home's HVAC/WH system which makes integration of all three services into one system a somewhat easier task. Multifunction systems like the AS-IHP also offer the potential to make fuller use of higher-efficiency but higher cost heat pump system components like variable-speed (VS) or multiple capacity compressor systems as well as VS blowers, fans, pumps, etc. The multi-capacity capability allows the larger SH and SC peak loads to be met with higher speed operation, while the smaller off-design space conditioning and WH loads are met at lower speeds. Significant energy savings are possible from the higher efficiency operation of the components, the load matching operation of the equipment (providing heat exchanger unloading benefits), and waste heat recovery in combined SC and WH operation. Waste heat recovery for WH from desuperheating during SC is free, except for a small pump energy use, and even helps to reduce condensing temperature. WH during SH is performed at a relative high efficiency and again helps to reduce condensing temperature. Dedicated WH is also quite efficient, but does generally result in elevated condensing temperatures. All of these operations may be delivered by a single compressor or by multiple compressors depending on the specific system design.

## 2. APPROACH

The prototype IHP was installed in a two-room (indoor and outdoor) environmental test chamber. A series of tests were conducted and system performance parameters were evaluated according to the procedures outlined in ASHRAE Standard 206-2013. (ASHRAE 2013). Standard 206 was developed to provide a uniform method of testing for rating the performance of multi-purpose heat pumps like the prototype AS-IHP described in this paper. The standard allows for testing of heat pump systems that also provide additional functions, such as ventilation and/or dehumidification. It provides a single comprehensive procedure for all existing and anticipated multi-purpose heat pumps, covering nine different, basic systems: single, dual, or variable capacity air-source, liquid-source, or direct expansion ground source systems. Up to 7 different operating modes can be accommodated by the Standard: A) space conditioning only; B) space conditioning + WH; C) dedicated WH; D) DH + space conditioning; E) dedicated DH; F) DH + space conditioning + WH; and G) DH + WH. The standard specifies the test equipment for performing such tests, the physical arrangement of the interconnecting refrigerant and water lines, the data required and the calculations to be used. For the subject test system of this paper only the Mode A, B, and C tests were performed since it has no dedicated DH mode.

## 2.1 System Description and Test Set Up

The prototype test system consisted of three major components: 1) an outdoor unit containing the compressor, outdoor fan and coil, 2) an indoor air handling unit, and 3) a WH module containing the water circulating pump and a refrigerant/water heat exchanger. Figure 2 shows a schematic of the system (left side) and a schematic of the water heating module and water tank connection plumbing (right side). Photos of the test system are shown in Figures 3 and 4. The system can have up to four air handling units plus the WH module but only one was included in the test system. For combined SH and WH (Mode B) operation, the system is designed to give SH the priority. Both SH and WH priority options are investigated in this paper. The system also includes demand defrost control which is reflected in the Standard 206 system efficiency calculations.

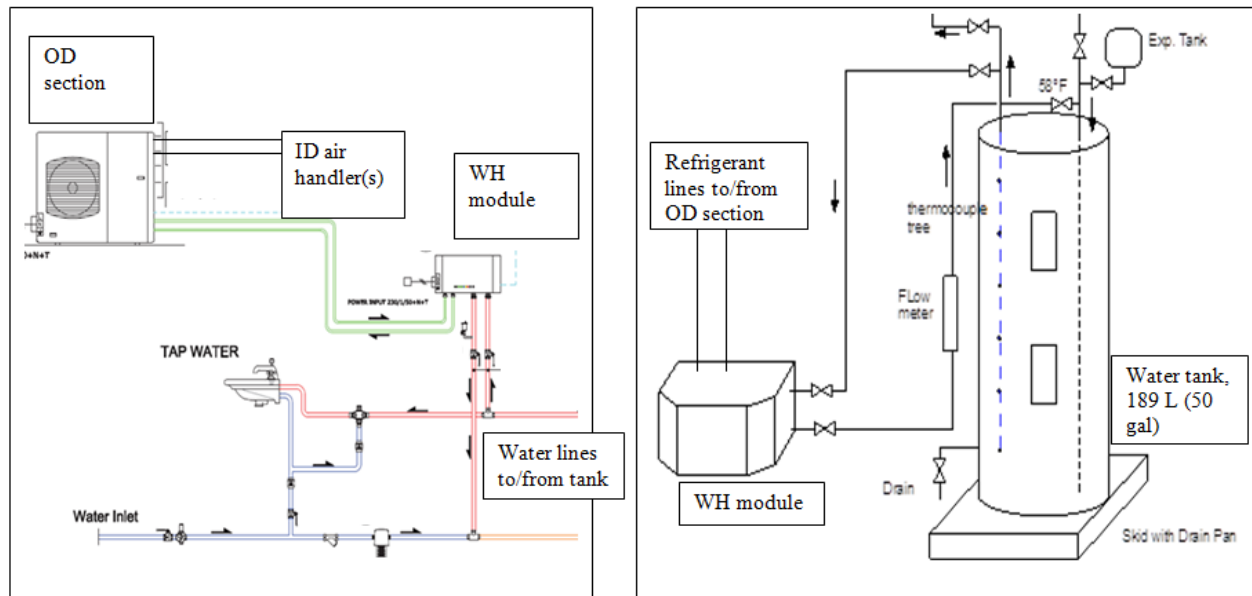


Fig. 2. Test IHP system and water connection schematics (left and right sides, respectively).



Fig. 3. Test system.

Table 1 provides a list of the principle instrumentation and measurement accuracies used for experimental performance determination of the system. Airside measurements of heating and cooling capacities along with water-side measurements of water heating capacities and electric power measurements were the primary values used to determine the system efficiency metrics as defined in Standard 206 (ASHRAE 2013). A high accuracy Coriolis flow meter was used to measure water flow between the WH module and water tank. Supply air flow rates were measured using a multi-point, self-averaging Pitot traverse station with an integral air straightener/equalizer honeycomb cell (Figure 4). Refrigerant side measurements were also used to provide a quality check on the airside measurements. Compressor map curve fits were used in conjunction with saturated condensing and evaporating refrigerant conditions to determine the refrigerant mass flowrate, in turn used to evaluate the refrigerant side performance. Airside vs. refrigerant side capacity for several Mode A tests are given in Table 2. Refrigerant side capacities generally ranged from +/- 8% of the airside measurements except for the low temperature heating condition where it was ~12% lower than the airside measurement.

**Table 1:** Test instrumentation and accuracy

Measurement	Instrument	Accuracy
Water flow between tank and heat pump	Coriolis flowmeter	+/- 0.12% for 5.7-57 L/Min (1.5-15 gpm) range
Indoor unit air flow	Pitot tube array	+/- 0.5%
System power	Watt/watt-hr meter	+/- 0.5%
Water pump power	Watt/watt-hr meter	+/- 0.5%
Air temperature in/out indoor unit	Averaging type-T thermocouple (TC) grid	+/- 0.5 °C or 0.4% for 0-350 °C, whichever greater
Water temperature in/out WH module	Immersion type-T TCs	+/- 0.5 °C or 0.4% for 0-350 °C, whichever greater
Tank water temperature	Six immersion type-T TCs in a vertical tree inserted into the HW outlet pipe	+/- 0.5 °C or 0.4% for 0-350 °C, whichever greater
Refrigerant temperatures in/out all major components	Surface mount type-T thermocouples	+/- 0.5 °C or 0.4% for 0-350 °C, whichever greater
Relative humidity in/out indoor unit	RH sensor	+/- 2% RH
Refrigerant pressures; discharge, suction, liquid	Electronic pressure gauges	+/- 0.05% of full scale
Indoor coil condensate	Electronic scale	+/- 0.5 gm linearity



**Fig 4. Indoor air flow measurement station.**

**Table 2: Airside vs. refrigerant side capacity comparisons**

Test condition	Airside capacity (kW)	Refrigerant side capacity (kW)	% difference vs. airside
Cooling (35°C OD; high compressor speed)	9.29	9.94	+7.7%
Cooling (30.6°C OD; intermediate compressor speed)	6.69	6.92	+3.5%
Cooling (27.8°C OD; high compressor speed)	9.94	10.46	+5.2%
Heating (8.3°C OD; high compressor speed, heating)	11.84	10.97	-7.7%
Heating (8.3°C OD; high compressor speed, cooling)	11.25	10.54	-6.3%
Heating (1.7°C OD; high compressor speed)	8.40	8.31	-1.1%
Heating (1.7°C OD; intermediate compressor speed)	4.82	4.60	-4.6%
Heating (-8.3°C OD; high compressor speed, heating)	7.23	6.32	-11.6%

System target compressor, blower, outdoor fan, and pump speeds were set by the PC-based data acquisition and control (DAQ) system using test control files provided by the system manufacturer for each test condition. Minimum and maximum compressor speeds were established per the manufacturer's internal controls recommendations. The air handler static pressure rise at maximum air flow was set to 0.294 kPa (1.18 in. water) per the manufacturer requirements prior to each individual test using an adjustable damper downstream of the air flow measurement station.

## 2.2 Test Plan and Results

Tests were conducted with the system in a set of environment controlled chambers (indoor and outdoor) for test conditions prescribed in Table 10.7a of ASHRAE Standard 206 for variable-capacity air-source systems. A total of 32 separate test conditions are given in the standard, however since the subject system did not include a dedicated DH operation mode only 24 tests were required for this study. Frost accumulation tests were performed but cyclic tests were not. Cyclic degradation coefficients (Cd) of both 0.25 and 0.05 were assumed for the performance calculations per Standard 206 to investigate overall performance sensitivity to that parameter. Table 3 lists the tests conducted along with outdoor (OD) air temperature conditions and compressor speed settings. Indoor (ID) conditions were held to 27.8 °C (80 °F) dry bulb and 19.4 °C (67 °F) wet bulb for space cooling tests and 21.1 °C (70 °F) dry bulb for space heating tests. For the Mode B and C water heating tests a series of water draws were imposed on the system with entering water temperature held to 14.4 °C (58 °F) and a water heating set point of 57.2 °C (135 °F).

**Table 3:** Tests run and operating conditions

Test conditions	OD temp	Compressor speeds
<b>Mode A – space conditioning only</b>		
Space heating (SH)	16.7 °C (62 °F)	Low (heat)
	8.3 °C (47 °F)	Hi (heat), Hi (cool)*, Low (heat)
	1.7 °C (35 °F), frost accumulation	Hi (heat), Intermediate (heat)
	-8.3 °C (17 °F)	Hi (heat)
Space cooling (SC)	35 °C (95 °F)	Hi (cool)*
	30.6 °C (87 °F)	Intermediate (cool)
	27.8 °C (82 °F)	Hi (cool)*, Low (cool)
	19.4 °C (67 °F)	Low (cool)
<b>Mode B – space conditioning + water heating (WH)</b>		
SH+WH	16.7 °C (62 °F)	Low (heat)
	8.3 °C (47 °F)	Hi (heat), Low (heat)
	-8.3 °C (17 °F)	Hi (heat)
SC+WH	35 °C (95 °F)	Hi (cool)*
	27.8 °C (82 °F)	Hi (cool)*, Low (cool)
	19.4 °C (67 °F)	Low (cool)
<b>Mode C – dedicated WH</b>		
WH only	35 °C (95 °F)	Per unit controls
	19.4 °C (67 °F)	
	16.7 °C (62 °F)	
	8.3 °C (47 °F)	

\*Hi speed for cooling was lower than Hi speed for heating

It must be acknowledged here that the compressor speed conditions used for the subject unit differed in two main respects from those required for strict compliance with Standard 206. First, the minimum speed used for the Mode B tests was higher than that for the Mode A tests (Standard 206 requires that they be the same). Secondly, the maximum speed used for the -8.3 °C (17 °F) Mode B SH+WH test was higher than that used for the same Mode A SH only test (Standard 206 requires that they be the same). The note on Table 2 indicates that the maximum heating speed was higher than the maximum cooling speed as well. However, this is allowable under the Standard 206 procedures. These compressor speed differences resulted from the internal control algorithms of the test unit, which established the safe operating speed based on internal safety protections at each test condition.

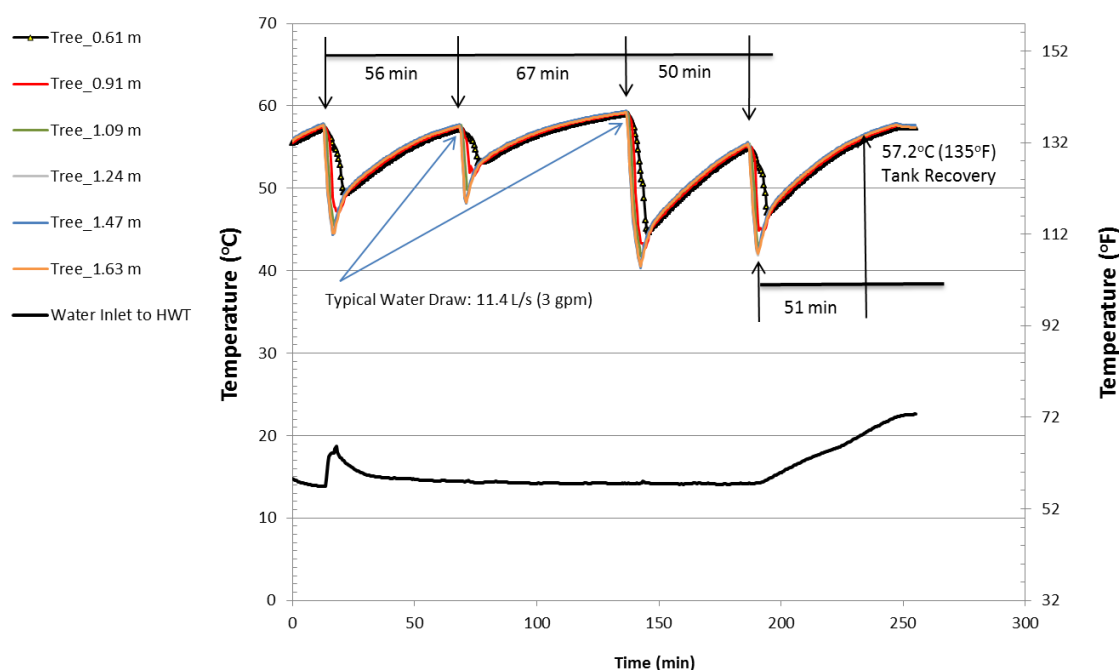
One of the primary energy saving advantages for IHP systems is their ability to recover rejected SC energy to serve the WH load and to provide WH at heat pump efficiencies year-round. Table 4 illustrates the system efficiency results for the Mode B SC+WH and the Mode C tests, ranging as high as ~3.8 (for the Mode C 35 °C test). Combined SH+WH operation mode efficiencies are given in Table 4 as well. These were about the same as for

SC+WH but in this case approximately 25% of the total output capacity is redirected to WH compared to SH only operation.

**Table 4:** SC+WH (Mode B) and dedicated WH (Mode C) system COP results

OD temperature	Mode B SC+WH mode	Mode B SH+WH mode	Mode C Ded. WH mode
35.0 °C (95 °F)	3.24 (max speed)	--	3.79
27.8 °C (82 °F)	3.65 (min speed)	--	--
19.4 °C (67 °F)	3.70 (min speed)	--	2.83
16.7 °C (62 °F)	--	3.80 (min speed)	2.37
8.3 °C (47 °F)	--	3.33 (max speed)	2.25
8.3 °C (47 °F)	--	3.23 (min speed)	--
-8.3 °C (17 °F)	--	2.65 (max speed)	--

Figure 5 shows an example of the water tank temperature profile for one Mode C test condition (19.4 °C (67 °F)) as measured by the thermocouple tree in the tank. Test data collection began with the second water draw (at ~75-min point) and continued until final tank recovery to WH set point. Due to the continuous circulation of water between the tank and the heat pump system's WH module during the test there is little stratification in the tank water temperatures except for a relatively short time after each water draw.



**Fig. 5. IHP water tank temperature profile for Mode C 19.4 °C (67 °F) test condition. Resistance heat in WH was not required by IHP to fully recover water temperature to 57.2°C (135°F).**

The prescribed system efficiency calculation method defines metrics for both non-integrated (baseline) and IHP, or combined appliance (ca), operations. For the baseline heat pump metrics, the traditional seasonal energy efficiency ratio (SEER) and heating seasonal performance factor (HSPF) are used. SEER and HSPF are based on the Mode A performance results of the system (ASHRAE 2013). The Energy Factor (EF) metric is used for the base water heater, in this case an electric storage unit with EF=0.95 (for systems manufactured after April 2015). For the IHP operation, the Standard's calculation procedure uses all of the test results (Modes A, B, and C for the subject system) to calculate "combined appliance" versions of SEER, HSPF, and EF, called SEERca, HSPFca and EFwca, respectively. Table 5 provides the system performance metrics determined from the "as tested" system performance results (based on Region IV climate data) for two levels of Cd. Recall earlier it was noted that several of the



compressor speeds used in the testing were higher than that specified for strict compliance with Standard 206, therefore the calculated ca metrics are subject to some uncertainty. This is due to the fact that the calculation procedure was developed assuming that maximum and minimum compressor speeds for modes A and B are the same so a common interpolation procedure could be used for both. In an attempt to correct for that uncertainty, the Mode A test data was used to estimate capacities and power input rates for Mode B assuming the same compressor speeds as for Mode A. Table 6 provides SEERca, HSPFca, and EFWca based on those estimated data also for two Cd levels.

**Table 5:** System performance metrics per Standard 206 calculation procedure (as tested performance)

Baseline separate systems			IHP		
	Cd = 0.25	Cd = 0.05		Cd = 0.25	Cd = 0.05
SEER	4.03 W/W (13.76 Btu/Wh)	4.07 W/W (13.91 Btu/Wh)	SEERca	4.13 W/W (14.08 Btu/Wh)	4.28 W/W (14.62 Btu/Wh)
HSPF	2.72 W/W (9.27 Btu/Wh)	2.73 W/W (9.32 Btu/Wh)	HSPFca	2.73 W/W (9.32 Btu/Wh)	2.75 W/W (9.40 Btu/Wh)
EF	0.95	0.95	EFwca	2.64	2.74

**Table 6:** System performance metrics per Standard 206 calculation procedure (estimated test results for strict Standard 206 compliance)

Baseline separate systems			IHP		
	Cd = 0.25	Cd = 0.05		Cd = 0.25	Cd = 0.05
SEER	4.03 W/W (13.76 Btu/Wh)	4.07 W/W (13.91 Btu/Wh)	SEERca	4.37 W/W (14.91 Btu/Wh)	4.42 W/W (15.08 Btu/Wh)
HSPF	2.72 W/W (9.27 Btu/Wh)	2.73 W/W (9.32 Btu/Wh)	HSPFca	2.84 W/W (9.70 Btu/Wh)	2.85 W/W (9.74 Btu/Wh)
EF	0.95	0.95	EFwca	3.11	3.14

Compared to the “as tested” performance in Table 5 the Table 6 SEERca increased by about 6%, the HSPFca increased by ~3%, and the EFWca increased by approximately 13-17%. The increase in SEERca and HSPFca is due to the increased efficiency during space heating for mode B at minimum speed. The increase in EFWca is due to the increased efficiency at B minimum speed, and increased use of mode C operation in place of resistance water heating.

The IHP efficiency metrics in Tables 5 and 6 are calculated assuming that WH has priority over space conditioning in the heating season. Assuming that space conditioning has priority results in a slight increase in HSPFca accompanied by a somewhat larger decrease in EFWca, while SEERca does not change. HSPFca increases by ~0.1% and EFWca, decreases by ~5% (from 3.11 to 2.94). It can also be observed that a Cd of 0.05 results in relatively small increases in SEER, HSPF, SEERca, HSPFca, and EFWca (1.1%, 0.5%, 1.1%, 0.4%, and 1.0%, respectively).

### 2.3 Comments on IHP Energy Savings Potential

Annual energy savings potential for the tested IHP system vs. the baseline separate systems was estimated using the Region IV energy usage as computed by the Standard 206 calculations. Results are summarized in Table 7 for the 0.25 Cd case. SC and SH energy savings estimates are 2.3% and 1.4%, respectively for the “as tested” efficiencies and rise to 7.7% and 4.3%, respectively, the scaled Mode B case. For WH, energy savings based on the “as tested” efficiencies were ~64% and for the scaled Mode B data case they were ~69%. Annual energy savings were estimated to be ~21% “as tested” and ~25% for scaled Mode B results as compared to a similar baseline VS air source heat pump with electric WH. Changing from WH priority to SH priority reduces the WH and total savings estimates by about 0.5%.

**Table 7:** Estimated energy savings for IHP system vs. baseline from Standard 206 calculations (Cd = 0.25)

Operating mode	IHP savings based on “as tested” data		IHP savings based on Mode B test data scaled to same compressor speeds as Mode A	
	WH priority	SH priority	WH priority	SH priority
SC	2.3%	2.3%	7.7%	7.7%
SH	1.4%	1.4%	4.3%	4.4%
WH	64.0%	63.7%	69.3%	67.5%
Total	21.9%	21.4%	25.6%	25.1%

### 3. CONCLUSIONS

Based on the results from test of the subject prototype IHP system and seasonal/annual efficiencies estimated using Standard 206 procedures, the following concluding observations are drawn.

- Combining the WH function with space conditioning functions in an IHP system can yield annual WH use energy savings of >60% (as high as ~70% if the estimated IHP performance based on the scaled Mode B results are valid).
- Assigning priority to WH operation over SH (per the default Standard 206 calculation assumption) results in an increase in the EFWca of ~5% at the cost of a modest drop in HSPFca of ~0.1%. However, the impact on estimated annual energy use for WH is only about 0.5% for Region IV.
- The impact of a reduction in the cyclic degradation coefficient, Cd, from the default value of 0.25 to 0.05 is minor for this variable-speed unit. The seasonal efficiency metrics for both the baseline and IHP cases increased from 0.4 to 1.1%.

Finally, readers are cautioned that the IHP system evaluated in this project used compressor speed settings for the Mode B (combined space conditioning and WH) operation that were not the same as those for Mode A (space conditioning only). The standard 206 calculation procedure for determination of seasonal efficiency metrics assumes that Mode A and B compressor speeds are identical. The disparity in the present tests will cause some uncertainty in the computed results. However, there was enough test data taken in Mode A to estimate Mode B performance for the same speeds as used in Mode A. The overall trends in the estimated system performance are consistent for both groups of data.

### NOMENCLATURE

AS-IHP	air-source integrated heat pump	(-)
ca	combined appliance (per ASHRAE Standard 206)	(-)
DAQ	data acquisition and control	(-)
DH	dehumidification operation mode	(-)
EF	Energy Factor for standard electric water heater	(W/W)
EFwca	WH Energy Factor for IHP	(W/W)
GS-IHP	air-source integrated heat pump	(-)
HSPF	heating season performance factor	(W/W or Btu/Wh)
HSPFca	HSPF for IHP	(W/W or Btu/Wh)
IHP	integrated heat pump	(-)
RH	relative humidity	(%)
SEER	seasonal energy efficiency ratio	(W/W or Btu/Wh)
SEERca	SEER for IHP	(W/W or Btu/Wh)
SC	space cooling operation mode	(-)
SH	space heating operation mode	(-)
TC	thermocouple	(-)
VS	variable speed	(-)
WH	water heating operation mode	(-)

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