NISTIR 6803

A Comparison of Rating Water-Source Heat Pumps Using ARI Standard 320 and ISO Standard 13256-1

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Abstract

This investigation compares performance ratings obtained when testing water-source heat pumps using the Air-conditioning and Refrigeration Institute (ARI) Standard 320 and the International Standards Organization (ISO) Standard 13256-1. Multiple tests were run using two heat pumps of different capacities from different manufacturers. These tests included a ducted 1.75 kW (0.5 ton) unit and a non-ducted 3.52 kW (1.0 ton) unit. Air external static pressure and water flow were varied at the ISO conditions to determine the correction in capacity and total power mandated by the ISO standard. The effects of this variability were measured and compared to test results using the ARI Standard 320 as the baseline test. ISO cooling capacity for the first and second units were 0.1 % higher and 1.1 % lower than the ARI capacity, respectively. ISO cooling energy efficiency ratio (EER) for the first and second units were 4.5 % higher and 3.9 % lower than the ARI, respectively. ISO heating capacity for the first and second units were 4.8 % lower and 2.9 % lower than the ARI capacity, respectively. ISO heating coefficient of performance (COP) for the first and second units were 6.2 % higher and 1.0 % lower than the ARI, respectively.

Keywords: Air conditioner, ARI Standard 320, Capacity, COP, EER, Heat Pump, ISO Standard 13256-1, Water-Source Heat Pump

Acknowledgement

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Use of Non-SI Units in a NIST Publication: The policy of the National Institute of Standards and Technology is to use the International System of Units (metric units) in all **of** its publications. However, in North America in the heating, ventalation and air-conditioning industry, certain non-SI units are so widely used instead of **SI** units that it is more practical and less confusing to include some measurement values in customary units only.

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Nomenclature

COP	heating coefficient of performance (W/W)
DP	pressure difference (Pa)
DT	temperature difference measured by a thermopile (°C)
EER	cooling energy efficiency ratio (Btu/Wh)
h.p.	heat pump
q	air or water flow rate (L/s)
Δp	static pressure drop (Pa)
wrt	with respect to
Φ_{pa}	fan or pump power adjustment (W)
η	dimensional constant of $0.3 \cdot 10^3$ [(L/s)(Pa)(1/W)] as prescribed by the ISO Standard 13256-1

1: Introduction

Globalization of the economies creates new marketing opportunities and increases the importance of international standards. The use of international standards becomes particularly important for the manufacturing sector which products can be shipped internationally. The adoption of an international standard offers substantial economic benefits, but the transition from a national to international standard poses a question whether the ratings obtained by using these standards are equivalent.

This study was concerned with rating obtained for water-source heat pumps test using two standards: the standards developed by the Air-conditioning and Refrigeration Institute (ARI), ARI Standard 320 (1998), and the standard developed by the International Organization for Standardization (ISO), ISO Standard 13256-1 (1998). The ISO standard is increasing in use. On January 1, 2000, the ARI adopted the ISO standard as the basis for its certification programs. The standard developed by the American Society of Heating, Refrigerating, and Air-conditioning Engineers, ASHRAE Standard 90.1 (1999), references both the ARI standard and the ISO until October 29, 2001, with the ISO standard designated as the exclusive standard starting at this date. The goal of this study was to evaluate the differences in rated energy efficiency ratio (EER) for cooling operation and coefficient of performance (COP) for heating operation obtained when using these two test methods.

The test and rating results obtained when using the ARI standard and ISO standard are expected to be somewhat different because of three inherent differences between these standards:

(1) The first difference is the slightly different dry-bulb and dew-point temperatures. These different operating conditions are related to different temperature scales (Fahrenheit vs. Celsius) and do not represent a significant difference in the test operating temperatures..

(2) The second difference between the ARI standard and the ISO standard is the external air static pressure applied during the test. Under the ARI standard, the unit must be tested while operating against the external air static pressure that is specified by the standard for a given system's capacity. Under the ISO standard, the unit must be tested against static pressure specified by the manufacturer. After completion of the test, a credit is given for the indoor fan power to the total energy input, and the system capacity is credited for the heat added by the indoor fan.

(3) The third difference is the treatment of the energy input to the water pump. Under the ARI standard, this energy input is not included in the calculation of the total energy input, and the standard specifies the water flow rate that results in a 5.6 °C (10.0 °F) temperature change across the heat exchanger. Under the ISO standard, the test must be performed at the mass flow rate specified by the manufacturer, and the energy input to the water pump is measured and included in the total energy input.

The following sections present the experimental apparatus, systems tested, and laboratory test results obtained by the ARI and ISO standards. Tests of one ducted and one non-ducted water-source heat pump provided comparison data for these two test procedures. In addition to "standard" testing carried out using the two standards, expanded testing was performed under ISO testing conditions with varied air external static pressure and water flow rates. These tests provided information regarding the effect on the rating of these two parameters that are specified by the manufacturer of the heat pump. The appendices include the uncertainty analysis and the comparison of NIST test results those obtained on the same model units by their manufacturers.

2: Experimental Setup

2.1: Test Setup

The main components of the experimental apparatus are shown below in Figure 2.1 ; these include the tested heat pump, the nozzle chamber, and the pull-thru fan. The water-source heat pump was supplied with distilled water conditioned to the appropriate temperature and flow rate. Inlet air was conditioned by the environmental chamber to the appropriate dry-bulb and dew-point temperatures required by either the ISO or ARI standard.

Mixers were included in the ductwork before the thermopile and thermocouple grids to ensure well mixed air. The 15-node thermocouple grid before and after the test unit was used to verify that the air was well mixed. Air temperature difference across the test unit was measured by a 10-junction thermopile. Dew-point temperature was measured before and after the test unit. For the unit equipped with ductwork connections, air pressure drop was measured across the system. These measurements were collected according to ASHRAE Standard 37-1988.

The nozzle chamber was constructed according to ANSYAMCA 210-85 (1985). The nozzle chamber measured the volume flow of air thru each unit. Airflow rate was controlled by a variable frequency drive on the pull-thru fan. All airflow rates were converted to standard conditions as described in the standard.

The test heat pump was supplied with conditioned distilled water at the appropriate flow rate and temperature. Water temperature difference was measured by a 10-junction thermopile located in a well inserted in the inlet and exit water lines. Water temperatures were measured by individual thermocouples inserted into the thermopile wells. Water coil temperature change was measured by a 10-junction thermopile. Water coil pressure drop was measured by a wet-wet differential pressure transducer. The water coil pressure drop was used by the ISO standard to correct for pumping power consumption.

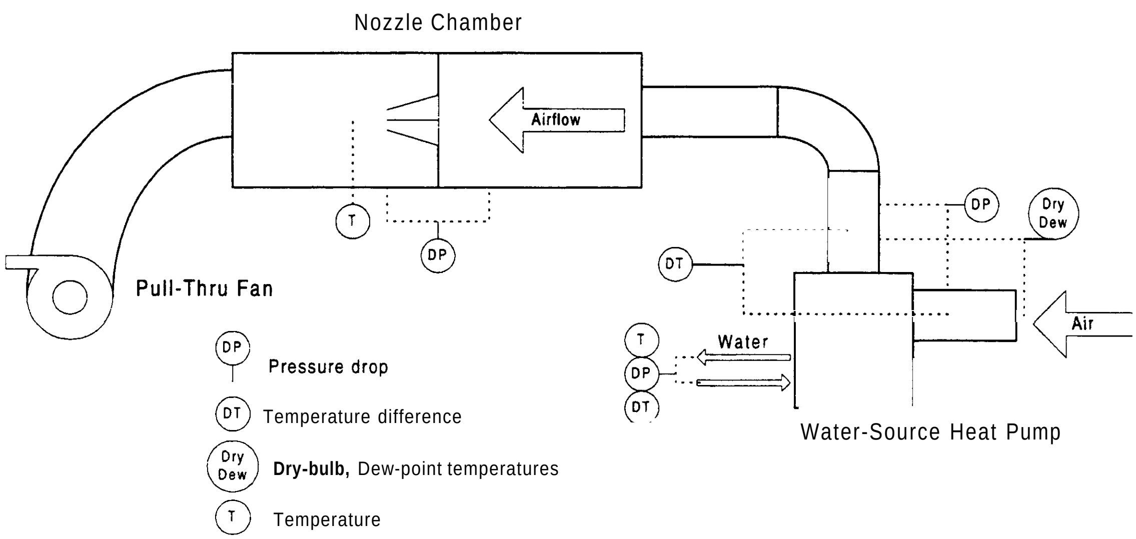

Figure 2.1: Water-source heat pump test apparatus

2.2: Instrumentation and Data Acquisition

Data were gathered using a personal computer and a multiplexed data acquisition unit. Over 50 data points were monitored throughout the testing. Table 2.1 lists measured quantities and their 95 % confidence limits. Appendix **A** gives a detailed uncertainty analysis for capacity and EER or COP.

Table 2.1: Measurement uncertainties

Ouantitv	Range	Uncertaintv*
Temperature	-18 "C to 93 "C (0 "F to 200 °F)	k0.3 "C (±0.5 °F)
Temperature change	0 "C to 28 °C (0 "F to 50 °F)	k0.3 °C (±0.5 °F)
Dew-point temperature	0 "C to 50 °C (32 "F to 122 °F)	M.2 °C (fo.4 °F)
Barometric pressure	0 mm Hg to 1270 mm Hg (0 in Hg to 50 in Hg)	k0.34 mm Hg (±0.0135 in Hg)
Air coil pressure difference	0 Pa to 1245 Pa (0 in H20 to 5.0 in H_2O)	±1.0 Pa (±0.004 in H_2O)
Water coil pressure difference	0 kPa to 69 kPa (0 psid to 10 psid)	k0.17 kPa (M.025 psid)
Air nozzle pressure difference	0 Pa to 623 Pa (0 in H20 to 2.5 in H_2O)	fo.87 Pa (±0.0035 in H_2O)
Total power	0 watts to 2000 watts	±5.0 watts

3: Experimental Procedure and Test Conditions

For both heating and cooling tests, the refrigeration chamber was maintained within 0.3 °C (0.5 °F) of a constant dry-bulb temperature and dew-point temperature.

Distilled water was brought into the system at a temperature specified by the appropriate standard (Tables 3.1, 3.2, 3.3 and 3.4). In the cooling mode, the water flow rate was adjusted to give a 5.6 °C (10.0 °F) temperature increase for the ARI standard and as specified by the manufacturer for the ISO standard.

Inlet air dry-bulb and dew-point temperatures were maintained for one hour within the specified range with the systems at steady-state before tests began. The temperature across the exit thermocouple grid was monitored to ensure well mixed air. Air coil static pressure drop was measured and recorded for the ducted unit tested.

In the heating mode, all fan settings and water flow rates were maintained the same from the respective cooling tests. For the ISO standard, the fan power correction was added to the heating capacity and to the total power. All other procedures followed those used during the cooling tests.

Table 3.1: ARI cooling conditions

Location	Setpoint	Tolerance
Indoor Dry-bulb Temperature	26.7 °C (80.0 °F)	M.3 °C (±0.5 °F)
Indoor Dew-point Temperature	**15.8** °C (60.4 °F)	a . 3 °C (±0.5 °F)
Inlet Water Temperature	29.4 °C (85.0 °F)	a . 3 °C (±0.5 °F)
Outlet Water Temperature	*35.0* °C (95.0 °F)	±0.3 °C (±0.5 °F)

Location	Setpoint	Tolerance
Indoor Dry-bulb Temperature	27.0 °C (80.6 *0F)*	M.3 °C (±0.5 *0F)*
Indoor Dew-point Temperature	**14.7** °C (58.5 °F)	**M.3** °C (±0.5 °F)
Inlet Water Temperature	30.0 °C **(86.0** °F)	M.3 °C (±0.5 °F)
Water Flow	Water flow specified by the manufacturer	

Location	Setpoint	Tolerance
Indoor Dry-bulb Temperature	21.1 "C (70.0 °F)	±0.3 "C (±0.5 °F)
Inlet Water Temperature	21.1 "C (70.0 °F)	±0.3 °C (±0.5 °F)
Water Flow	Same water flow as the cooling test	

Table 3.4: ISO heating conditions

Location	Setpoint	Tolerance
Indoor Dry-bulb Temperature	20.0 "C (68.0 °F)	M.3 "C (±0.5 °F)
Maximum Dew Point	11.7 "C (53.1 °F)	M.3 "C (±0.5 °F)
Inlet Water Temperature	20.0 "C (68.0°F)	H.3 °C (±0.5 °F)
Water Flow	Same as in the cooling test above	

4: Units Tested, Tests Performed and Data Reduction

Two water-source heat pumps were selected for this study. The first unit was a ducted design with a nominal cooling capacity of 1.75 kW (0.5 ton). The second unit was a non-ducted console type design with a nominal cooling capacity of 3.52 kW (1.0 ton). For the non-ducted unit, the air static pressure at the exit of the unit was maintained at zero for all tests. Neither unit included a pump for circulating water through the water coil. Both units were tested according to ARI Standard 320 and ISO Standard 13256-1. Table 4.1 below summarizes the tests performed on each unit for the cooling and heating modes. In addition to the "normal" ISO test with airflow and water flow specified by the manufacturer, tests with increased and decreased air static pressure and water coil pressure drop were performed to examine their effects upon EER and COP. These tests are described in Table 4.1 as Modified ISO tests.

Table 4.1: Test matrix summary for cooling and heating modes

	External Air Static Pressure*	Water Coil Pressure Drop
ARI 320	Normal	Normal
ISO 13256-1	Normal	Normal
Modified ISO	High	Normal
	LOW	Normal
	Normal	High
	Normal	LOW

* The non-ducted unit **was** maintained at zero exit static pressure for all tests.

Air-side capacity was calculated using the measured air **flow** rate, specific heat, and changes in air dry-bulb temperature and moisture content. Barometric pressure was also used to calculate air properties for the given conditions. The nozzle pressure drop was converted to a volumetric flow rate. The nozzle temperature and humidity ratio were used to calculate the air density and convert volumetric flow rate into a mass flow rate. For the ISO standard, a correction to the air-side capacity and total power were calculated based on the external static air pressure drop of the air coil and the pressure drop across the water coil. This correction was calculated by Equation 4.1 below with the Ap being the static pressure drop of the fluid considered, air or water. The fan power correction, in watts, was added to the total power consumption and subtracted from the total capacity for the cooling tests. The fan power correction was added to the total power consumption and capacity for the heating tests. Pumping power was added to the total power for all heating and cooling tests.

$$\Phi_{pa} = \frac{q \times \Delta p}{\eta} \qquad (4.1)$$

where Φ_{pa} is the pump or fan power adjustment (watts)

q is the nominal fluid flow rate (L/s)

Ap is the measured pressure drop (Pa)

η is $0.3 \cdot 10^3$ as specified by ISO Standard 13256-1.

Total power was measured by a wattmeter during the test period, which was never shorter than 30 min. The total power measurement was combined with the water coil capacity as a secondary calculation of the air-side capacity. For the ARI standard, the reported capacity is based on the air-side measurements. For the ISO standard, the reported capacity is the average of the air-side and secondary method capacities. The agreement between the two methods was within 5.0 % for all tests. Note that the corrected values of capacity, EER, and COP are the heat pump ratings obtained from the ISO test procedure.

5: Experimental Results

5.1: Unit 1 – 1.75 kW (0.5 Ton) Nominal Cooling Capacity, Ducted System

Cooling tests

Table 5.1 summarizes the cooling test results for Unit 1. For the ISO test, the table presents detailed information; the uncorrected capacity, power, and EER are presented first. The following entries are system operating parameters, ISO corrections for capacity and power, and the corrected capacities and EERs. These corrected values are the reported capacities and EERs when tests are performed using the ISO method.

Under the ARI cooling conditions, air-side capacity and EER were 2352 **W** (8024 Btu/h) and 13.21, respectively. For the ISO 13256-1 cooling conditions, air-side capacity and EER were 2353 **W (8028** Btu/h) and 13.80. The ISO results include the fan power, capacity corrections, and the pump power correction. Correcting the capacity and power for the fan and pump, according to Equation 4.1, changed the EER from 12.89 (the uncorrected value in Table 5.1) to 13.80 (an increase of 7.06 %).

Air static pressure and water flow rate to the unit were varied to determine their effects upon capacity and EER within the ISO 13256-1 conditions. Air static pressure has the greatest effect upon capacity and EER due to the capacity correction of Equation 4.1 and fan power correction required by the ISO standard. For the low and high air static pressure tests air-side, capacity changed by 0.3 % and –3.6 %, respectively, as air volume flow changed by +30 % and **–30** %. EER change due to the changes in air volume flow rate were -0.7 % and –2.2 %. Changes in water flow rate through the water coil produced even smaller effects upon the ISO cooling test results. As water flow was varied by –10 % and +10 %, ISO air-side capacity changed by –1.6 % and +0.1 %, respectively. EER changed by –1.4 % and +0.7 %.

ARI capacity was 1.6 % higher than the ISO uncorrected capacity. EER was 1.6% higher than the ISO uncorrected EER. These differences were due to the differences in test conditions (dry-bulb and dew point). Capacity increased by 1.6 % due to correcting for fan capacity according to equation 4.1. The pump power correction produced a minimal effect upon EER as it was less than 1.5 % of the total power for all tests. EER increased by 7.0 % due to the corrections for fan heat, fan power, and pump power.

Table 5.1: ARI and ISO cooling test results for Unit 1

Cooling	External Air Static Pressure			Water Coil Pressure Drop		
	Low	Normal	High	Low	Normal	High
Using ISO 13256-1: Uncorrected Capacity, W (Btu/h)	2326 (7935)	2316 (7902)	2227 (7598)	2276 (7766)	2316 (7902)	2318 (7908)
Uncorrected Total Power, W	620	613	610	617	613	610
Uncorrected EER, Btu/Wh	12.80	12.89	12.46	12.59	12.89	12.97
Water Flow, L/s (gpm)	0.128 (2.0)	0.127 (2.0)	0.127 (2.0)	0.119 (1.9)	0.127 (2.0)	0.135 (2.1)
Water Temp Change, °C (°F)	5.44 (9.8)	5.44 **(9.8)**	5.38 (9.7)	5.83 (10.5)	5.44 (9.8)	5.17 (9.3)
Water Pressure Drop, Pa (psid)	13807 (2.00)	14134 (2.05)	13578 (1.97)	11038 (1.60)	14134 (2.05)	16368 (2.37)
Air Flow, L/s (cfm)	155 (328)	142 (301)	127 (270)	141 (299)	142 (301)	143 (302)
Air Temp Change, °C (°F)	10.22 (18.4)	10.61 (19.1)	11.0 (19.8)	10.67 (19.2)	10.61 (19.1)	10.72 (19.3)
Air Static, Pa (in H_2O)	69 (0.28)	78.2 (0.31)	99.4 (0.40)	82 (0.33)	78.2 (0.31)	79 (0.32)
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	36 (122)	37 (126)	42 (144)	39 (132)	37 (126)	37 (128)
ISO Power Adjustment: For Fan Power, W	36	37	42	39	37	37
For Pump Power, W	6	6	6	4	6	7
Corrected Capacity, W (Btu/h)	2361 (8057)	**2353 (8028)**	2269 (7742)	2315 (7898)	**2353 (8028)**	2355 (8036)
Corrected EER, Btu/Wh	13.65	**13.80**	13.50	13.56	**13.80**	13.86
Using ARI 320 Capacity, W (Btu/h)		**2352 (8024)**			**2352 (8024)**	
Total Power, W		608			608	
EER, Btu/Wh		**13.21**			**13.21**	

Heating tests

Table 5.2 summarizes the heating test results for Unit 1. Under the ARI heating conditions air-side capacity and COP were 3270 W (11157 Btu/h) and 4.81, respectively. For the ISO 13256-1 cooling conditions air-side capacity and COP were 3114 W (10624 Btu/h) and 5.11.

Air static pressure and water flow rate to the unit were varied to determine their effects upon capacity and COP within the ISO 13256-1 conditions. Air static pressure had the greatest effect upon capacity and efficiency due to the capacity correction and fan power correction. For the low and high air static pressure tests, air-side capacity changed by 1.8 % and –1.6 %, respectively, as air volume flow changed by +30 % and –30 %. COP change due to the changes in air volume flow rate were 1.7 % and –2.3 %. Changes in water flow rate through the water coil produced even smaller effects upon the ISO heating test results. **As** water flow was varied by –10 % and +10 %, **ISO** air-side capacity changed by –0.8 % and +0.7 %, respectively. COP changed by +0.2 % and +0.0 %.

ARI capacity was 1.2 % higher than the ISO uncorrected capacity. COP was 1.0 % lower than the ISO uncorrected COP. These differences were due to the differences in test conditions (dry-bulb and dew point). Capacity decreased by 0.3 % due to correcting the tests for fan capacity according to equation 4.1. The pump power correction produced a minimal effect upon COP as it was less than 1.5 % of the total power for all tests. COP increased by 5.0 % due to the corrections for fan heat, fan power, and pump power.

Table 5.2: ARI and ISO heating test results for Unit 1

Heating	Externa Air Static Pressure			Water Coil Pressure Drop		
	LOW	Normal	High	LOW	Normal	High
Using ISO 13256-1: Uncorrected Capacity, W (Btu/h)	3213 (10964)	(10787)	(10618)	3153 (10757)	3161 (10787) (16587)	3198 (10914) (16944)
Uncorrected Total Power, W	*646*			648		
Uncorrected COP	4.97	4.85	4.74	4.87	4.85	4.89
Water Flow, L/s (gpm)	0.125 (1.982)	0.124 (1.97)	0.125 (1.987)	0.106 (1.684)	0.124 (1.97)	0.136 (2.149)
Water Coil Temp Change, °C (°F)	4.72 (8.5)	4.72 (8.5)	4.67 (8.4)	5.44 (9.8)	4.72 (8.5)	4.39 (7.9)
Water Pressure Drop, Pa (psid)	14107 (2.046)	13600 (1.973)	13983 (2.028)	9990 (1.449)	13600 (1.973)	16237 (2.355)
Air Flow, L/s (cfm)	156 (331)	147 (311)	136 (287)	149 (315)	147 (311)	148 (315)
Air Coil Temp Change, "C (°F)	18.28 (32.9)	19.0 (34.2)	20.0 (36.0)	18.89 (34.0)	19.0 (34.2)	19.11 (34.4)
Air Static, Pa (in H_2O)	82 (0.329)	(0.393)	(0.427)	91 (0.366)	98 (0.393)	90 (0.362)
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	43 (146)			45 (154)	48 (163)	45 (152)
ISO Power Adjustment: For Fan Power, W	43	48	48	**45**	48	45
For Pump Power, W	5.9	*5.6*	***5.8***	3.5	5.6	7.3
Corrected Capacity, W (Btu/h)	3170 (10818)	**3114 (10624)**	3064 (10454)	3107 (10603)	**3114 (10624)**	3154 (10761)
Corrected COP	5.20	~~**5.11**~~	~~4.99~~	5.13	**5.12**	5.12
Using ARI 320 Capacity, W (Btu/h)		**3270 (11157) (16657)**			**3270 (11157)**	
Total Power, W					667	
EER, Btu/Wh		**4.81**			**4.81**	

5.2: Unit 2 – 3.52 kW (1.0 Ton) Nominal Cooling Capacity, Non-ducted System

Results for capacity, EER, and COP are reported below. Unit 2 was a console heat pump designed for wall mounting with no ductwork; therefore, air static pressure was maintained at zero for all tests to simulate free discharge to the indoor space.

Cooling tests

Table 5.3 summarizes the cooling test results for Unit 2. Under the ARI cooling conditions, air-side capacity and EER were 3085 W (10528 Btu/h) and 14.18, respectively. For the ISO 13256-1 cooling conditions, air-side capacity and EER were 3051 W (10412 Btu/h) and 13.63.

Water flow was varied to determine the effects upon capacity and EER. Changes in water flow rate through the water coil produced a small effect upon the ISO cooling test results. As water flow was varied by –20 % and +20 %, ISO averaged capacity changed by –1.3 % and **-0.3** %, respectively. EER changed by –2.8 % and +0.2 %. Unit 2 was designed for free air discharge to the conditioned space and, therefore, tests with varying external air static pressure were not performed.

ARI capacity was 1.1 % higher than the ISO uncorrected capacity. EER was 2.4 % higher than the ISO uncorrected EER. These differences were due to the differences in test conditions (dry-bulb and dew-point temperatures). The pump power correction produced a small effect upon EER as it was less than 2.8 % of the total power for all tests. EER decreased with respect to the ISO raw results by 1.6 %.

Table 5.3: **ARI** and **ISO** cooling test results for Unit **2**

Cooling	Water Coil Pressure Drop		
	Low	Normal	High
Using **ISO** 13256-1: Uncorrected Capacity, W (Btu/h)	3010 (10272)	3051 (10412)	3043 (10382)
Uncorrected Total Power, W	769	752	739
Uncorrected EER	13.36	13.85	14.05
Water Flow, L/s (gpm)	0.136 (2.15)	0.170 (2.694)	0.204 (3.23)
Water Coil Temp Change, "C (°F)	6.67 (12.0)	5.33 (9.6)	4.44 (8.0)
Water Pressure **Drop,** Pa (psid)	14403 (2.09)	21774 (3.16)	30585 (4.44)
Air Flow, L/s (cfm)	158 (335)	157 (332)	157 (333)
Air Coil Temp Change, "C (°F)	12.39 (22.3)	12.44 (22.4)	12.39 (22.3)
Air Static, Pa (in H_2O)	3.74 (0.015)	2.74 (0.011)	2.74 (0.011)
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	0	0	0
ISO Power Adjustment: For Fan Power, W	0	0	0
For Pump Power, W	7	12	21
Corrected Capacity, **W** (Btu/h)	3010 (10272)	**3051** **(10412)**	3043 (10382)
Corrected EER	13.25	**13.63**	13.66
Using **ARI** 320 Capacity, **W** (Btu/h)		**3085** **(10528)**	
Total Power, W		**742**	
EER		**14.18**	

Heating tests

Tables 5.4 summarizes the heating test results for Unit 2. Table 5.4 does not include tests at a low water flow rate. Two tests were performed at a lowered water flow rate, but they were excluded due to unacceptable variations (pulses) in water flow rate through the water coil.

Under the **ARI** heating conditions and normal airflow (Table 5.4), air-side capacity and COP were 4668 W (15927 Btu/h) and 4.94. For the **ISO** 13256-1 normal airflow heating conditions air-side capacity and COP were 4534 W (15469 Btu/h) and 4.89. For the normal airflow tests, the change in pumping power from normal to high water flow rate produced a minimal effect upon capacity and COP. When water flow rate was increased by 16.9 %, capacity increased by 0.8 % and COP decreased by 1.6 %. For the high water flow rate case, the pump power correction was 3.5 % of the total power. **ARI** capacity was 1.2 % higher than the ISO uncorrected capacity. COP was 2.6 % lower than the ISO uncorrected COP. These differences were due to the differences in test conditions (dry bulb and dew point) between the ISO and **ARI** standards. The pump power correction produced a minimal effect upon COP as it was less than 2.8 % of the total power for all tests.

In addition to the tests of Table 5.4, several tests were performed during the heating with varied water flow rate at a lowered airflow rate due to increased external static pressure. These low airflow tests were performed to determine whether consistent changes in capacity and COP were produced with changes in the water flow rate at the low and normal airflow rates. Under the **ARI** heating conditions and low airflow (Table 5.5), air-side capacity and COP were 4269 W (14567 Btu/h) and 4.23. For the **ISO** 13256-1 heating conditions and low airflow, air-side capacity and COP were 4210 W (14367 Btu/h) and 4.26. Lowering the water flow rate by 20.9 % had the effect of decreasing the capacity by 1.7 % and increasing the COP by 0.5%. Increasing the water flow rate by 21.7 % increased the capacity **by** 1.2 % and decreased the COP by 0.2 %. The lower water flow rate decreased the pumping power correction by 47.1 % from 17 **W** to 9 W. The higher water flow rate increased the pumping power by 70.6 % from 17 W to 29 W. The highest water flow rate, the pumping power correction was 3.0 % of the total power requirement.

Table 5.4: ARI and ISO normal airflow heating test results for Unit 2

Heating	Water Coil Pressure Drop	
	Normal	High
<u>Using ISO 13256-1:</u> Uncorrected Capacity, W (Btuh)	4534 (15469)	4571 (15598)
Uncorrected Total Power, W	911	919
Uncorrected COP		4.97
Water Flow, L/s	0.198 (3.135)	0.231 (3.666)
Water Coil Temp Change, "C (°F)	4.28 (7.7)	3.72 (6.7)
Water Pressure Drop, Pa (psid)	31523 (4.57)	42154 (6.114)
Air Flow. L/s (cfm)	188.6 (400)	188.9 (400)
Air Temp Change, "C (°F)	21.39 (38.5)	21.56 (38.8)
Air Static, Pa (in H_2O)	1.5 (0.006)	1.5 (0.006)
<u>ISO Capacity Adjustment:</u> For Fan Heat. W (Btu/h)	0	0
<u>ISO Power Adjustment:</u> For Fan Power, W	0	0
For Pump Power, W	21	32
Corrected Capacity, W (Btuh)	**4534 (15469)**	4571 (15598)
Corrected COP	**4.89**	4.81
<u>Using ARI 320</u> Capacity, W (Btu/h)	**4668 (15927)**	
Total Power, W	946	

Table 5.5: ARI and ISO low airflow heating: test results for Unit 2

Heating	Water Coil Pressure Drop		
	LOW	Normal	High
Using ISO 13256-1: Uncorrected Capacity, W (Btu/h)	4140 (14128)	4210 (14367)	4259 (14534)
Uncorrected Total Power, W	960	972	973
Uncorrected COP	4.32	4.33	4.38
Water Flow, L/s (gpm)	0.144 (2.289)	0.183 (2.895)	0.222 (3.524)
Water Coil Temp Change, °C (°F)	5.4 (9.7)	4.3 (7.8)	3.6 (6.5)
Water Pressure Drop, Pa (psid)	18568 (2.69)	27676 (4.01)	39073 (5.67)
Air Flow, L/s (cfm)	146 (310)	146 (310)	146 (310)
Air Temp Change, °C (°F)	24.56 (44.2)	25.06 (45.1)	25.28 (45.5)
Air Static, Pa (in H_2O)	23.91 (0.096)	23.41 (0.094)	23.66 (0.095)
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	0	0	0
ISO Power Adjustment: For Fan Power. W	0	0	0
Corrected Capacity, W (Btu/h)	**4140** (14128)	**4210 (14367)**	4259 (14534)
Corrected **COP**	4.28	**4.26**	4.25
Using; ARI 320 Capacity, W (Btu/h)		**4269 (14567)**	
Total Power, W		1010	
COP		**4.23**	

6: Summary

The purpose of this experimental investigation was to examine differences in water-source heat pump performance ratings obtained from tests according to ARI Standard 320 and ISO Standard 13256-1. This investigation also included tests at different volumetric flow rates of air and water to examine the effect of capacity and power corrections on the rating obtained by the ISO test procedure. Two water-source heat pumps were tested according to both standards.

Tables 6.1 summarizes results for capacity changes. ISO cooling capacity for the ducted unit and non-ducted unit were 0.1 % higher and 1.1 % lower than the ARI capacity, respectively. In the heating mode, the ISO capacities were 4.8 % lower and 2.9 % lower than the ARI capacities. Variation of the external air static pressure had a greater effect than the variation of the water flow rate. The range of capacity change was from –3.6 % to 1.8 %.

As shown by Table 6.2, the ISO cooling EERs for the ducted unit and the non-ducted unit were 4.5 % higher and 3.9 % lower than the ARI EERs, respectively. The ISO heating COPs for the first and second units were 6.2 % higher and 1.0 % lower than the ARI COPs, respectively. The range of ISO EER and ISO COP changes due to variation of the external air static pressure and water flow rate was from –2.8 % to 1.8 %. Similar differences between ISO and ARI EERs and COPs were obtained by manufacturers of these two units. As shown in Appendix C, the differences between the ISO and ARI EERs and COPs obtained by NIST were smaller than those obtained by the manufacturers with the exception of the heating COP for the non-ducted system. In this case, the manufacturer reported no difference between the two COPs while NIST measurements showed a 1.4 % lower ISO COP than the ARI COP.

The uncertainties for NIST results were calculated applying the uncertainty propagation law and considering the uncertainties of all involved temperature, pressure, and power measurements. For the 95 % confidence level, the maximum uncertainty for EER and COP was found to be 5.2 % and 5.9 %, respectively. Hence, the differences between the ISO and ARI ratings are near or within the limits of uncertainty.

Table 6.1: ARI 320 and ISO 13256-1 capacity comparison

Test	Cooling Capacity % Difference wrt ARI		Heating Capacity % Difference wrt ARI		
	Ducted h.p.	Non-ducted h.p.	Ducted h.p.	Non-ducted h.p.	
ISO	0.1	-1.1	-4.8	-1.4*	-2.9
	Cooling Capacity % Difference wrt ISO		Heating Capacity % Difference wrt ISO		
ISO Low Airflow	-3.6	NA	-1.6	NA	NA
ISO High Airflow	0.4	NA	1.8	NA	NA
ISO Low Water Flow	-1.6	-1.4	-0.2	-1.7	NA
ISO High Water Flow	0.1	**-0.3**	1.3	1.2	0.8

*Tests performed at a lower airflow across the indoor air coil than specified by the manufacturer

Table 6.2: ARI 320 and ISO 13256-1 efficiency comparison

Test	Cooling EER % Difference wrt ARI		Heating COP % Difference wrt ARI		
	Ducted h.p.	Non-ducted h.p.	Ducted h.p.	Non-ducted h.p.	
ISO	4.5	-3.9	6.2	0.7*	-1.0
	Cooling EER % Difference wrt ISO		Heating COP % Difference wrt ISO		
ISO Low Airflow	-2.2	NA	-2.4	NA	NA
ISO High Airflow	-1.1	NA	1.8	**NA**	**NA**
ISO Low Water Flow	-1.7	-2.8	0.4	0.5	NA
ISO High	0.4	0.2	0.2	-0.2	-1.6

*Tests performed at a lower airflow across the indoor air coil than specified by the manufacturer

7: References

ANSI/ASHRAE Standard 37-1988. ***Methods of testing for rating unitary air conditioning and heat pump equipment.*** American Society of Heating, Refrigerating and Air-conditioning Engineers. 1791 Tullie Circle NE, Atlanta, GA, **USA.**

ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51-1985. ***Laboratory methods of testing fans for rating.*** American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1791 Tullie Circle **NE,** Atlanta, GA, USA.

ARI Standard 320-1998. ***Standard for water-source heat pumps.*** Air-conditioning and Refrigeration Institute. 4301 North Fairfax Drive, Arlington, VA, USA.

ASHRAE Standard 116-1993. ***Method of testing for seasonal efficiency of unitary air-conditioners and heat pumps.*** American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1791 Tullie Circle NE, Atlanta, GA, USA.

ASHRAE Standard 90.1-1999. ***Energy Standard for Buildings Except Low-Rise Residential Buildings.*** American Society of Heating, Refrigerating and Air-conditioning Engineers. 1791 Tullie Circle **NE,** Atlanta, GA, USA.

DOE-1999. U. **S.** Department of Energy rule making regarding Test Procedures and Efficiency Standards for Commercial Air Conditioners and Heat Pumps, Docket Number: EE-RM/TP-99-460, Comment 5.

ISO 13256-1-1998. ***Water-source heat pumps-Testing and rating for performance-Part I: Water-to-air and brine-to-air heat pumps.*** International Organization for Standardization. Case postale 56, CH-1211 Geneva 20, Switzerland.

Appendix A: Uncertainty Analysis

A. 1 General Remarks

The uncertainty analysis was performed to gain knowledge about the uncertainty of the measured and calculated data. This Appendix presents the major equations used for the uncertainty analysis.

A.2 Theory

The uncertainty of a quantity R calculated from n independent measurements x_i is a function of the individual uncertainty of each measurement.

$$R = f(x_1, x_2, x_3, \ldots\ldots, x_n) \tag{A.1}$$

When each measurement, x_i, has a given uncertainty, dx_i, the maximum uncertainty of R is given by:

$$E_R = \left|\frac{\partial f}{\partial x_1}dx_1\right| + \left|\frac{\partial f}{\partial x_2}dx_2\right| + \left|\frac{\partial f}{\partial x_3}dx_3\right| + \ldots\ldots + \left|\frac{\partial f}{\partial x_n}dx_n\right|. \tag{A.2}$$

However, using the maximum error to judge the uncertainty of a calculated quantity is not common. Usually the standard deviation (root sum square) is regarded to be a much better approach to a quantity's uncertainty.

$$E_R = \sqrt{\left(\frac{\partial f}{\partial x_1}dx_1\right)^2 + \left(\frac{\partial f}{\partial x_2}dx_2\right)^2 + \left(\frac{\partial f}{\partial x_3}dx_3\right)^2 + \ldots\ldots + \left(\frac{\partial f}{\partial x_n}dx_n\right)^2} \tag{A.3}$$

The absolute error calculated with equation **(A.3)** is often converted to a relative error having the units of percent.

$$e, = \frac{E_R}{R}100 \tag{A.4}$$

A.3 Temperature Measurements

Most of the temperature measurements performed for these tests were determined by thermocouples. Their voltage signals were measured with the data acquisition system and then converted into a temperature.

The equation used in the test rig's control program to convert the voltage signals into temperatures was a sixth degree polynomial **of** the form:

$$\vartheta = f(V) = \frac{9}{5}(A + B\,V + C V^2 + D\,V^3 + E\,V^4 + F\,V^5 + G\,V^6) + 32 \qquad \text{(A.5)}$$

where:

79	–	temperature (°F)
V	–	measured voltage (μV)

If one premises that the uncertainty of the equation itself can be neglected, only one derivation is needed to evaluate the uncertainty in the temperature measurements.

$$\frac{\partial\vartheta}{\partial V} = \frac{9}{5}\left(B + 2C\,V + 3D\,V^2 + 4E\,V^3 + 5F\,V^4 + 6G\,V'\right) \qquad \text{(A.6)}$$

According to the manufacturer of the datalogger voltmeter, the 95 % uncertainty of the voltage measurement (*VM*) was: $E_{VM} = dV(VM) = \pm\ 0.007\ \%\ of\ reading + 5\ \mu V$.

The measurement of a temperature (ϑ) actually is the measurement of the difference to a reference temperature. The data acquisition system provided a temperature compensation to 0 °C (*32*°F) with a given uncertainty of: $E_{TC} = dTC = \pm\ 0.2236$ °C $= \pm\ 0.4025$ °F.

Rewriting equation A.3 for the measurement of the absolute temperature gives:

$$\vartheta = f(V) \qquad \text{(A.7)}$$

$$E_T = \sqrt{\left(\frac{\partial\vartheta}{\partial V}\,dVM\right)^2 + (dTC)^2} \qquad \text{(A.8)}$$

In addition to the common thermocouple measurements, the dew-point temperature in the air duct was measured to evaluate the humidity ratio of the moist air in the duct.

The manufacturer of the dew-point hygrometer specified the 95 % uncertainty in this measurement to be: $E, \quad = dT_{dew} = \pm\, 0.05\%$ of reading.

A. 4 Temperature Difference Measurements

The evaluation of the uncertainty of a temperature difference ($\Delta\vartheta$) measurement using a thermopile is slightly more complicated than that for a normal temperature measurement. The uncertainty evaluation is presented using the air duct temperature difference as an example, because this shows the most complicated case.

Again there are two independent uncertainties being part of the measurement uncertainty. The first is the uncertainty caused **by** the voltage signal measurement, discussed in section A.3. The cause for the second uncertainty influencing the measurement of a temperature difference is the nonlinear character of the temperature/voltage function (see equation A.5). The nonlinearity requires temperature at one end of the thermopile used for the temperature difference measurement to be known.

The temperature difference across the indoor coil was calculated using both the voltage signals of the temperature difference measurement (**AV**) and the average voltage signal ($V_{av.}$) of the entering temperature measurement of the air duct. The equation used to do so was:

$$\Delta\vartheta = f\left(V_{av.} + \Delta V\right) - f\left(V_{av.}\right) \tag{A.9}$$

The entering temperature was measured using 15 thermocouples equally distributed over the air duct's cross section. The average of the 15 temperature signals was considered to be the entering temperature. For the uncertainty in this average entering temperature the average voltage measurement uncertainty $E_{VM,av.}$ of the 15 measurements was calculated.

$$E_{VM,av.} = dV_{av}\left(VM\right) = \sum_{x=1}^{15} \frac{dV_{av.}\left(VM_x\right)}{15} \tag{A.10}$$

All 15 thermocouples were connected to the same temperature compensation. This means the overall uncertainty of the air's average entering temperature voltage signal $V_{av.}$ was:

$$dV_{av.} = \sqrt{E_{VM,av.}^{2} + E_{TC}^{2}} = \sqrt{\left(dV_{av.}(VM)\right)^2 + \left(dV_{av.}(TC)\right)^2} \tag{A.11}$$

To evaluate equation **A.**11 the uncertainty in the temperature compensation must be rewritten to have the unit of μV. Using equation **A S** one finds that an uncertainty of $E_{TC} = dTC = \pm 0.2236°C = \pm\ 0.4025\ {}^{O}F$ in the temperature compensation to 0 °C (32 °**F**) is equivalent to a voltage signal uncertainty of $dV_{av.}(TC) = \pm\ \mathbf{8.6264}\ \mu V$. **As** already mentioned, the uncertainty of the voltage signal measurement was given from manufacturer data.

The nonlinearity of the voltage/temperature function (AS) causes an uncertainty, *dslope,* in the temperature difference that depends on the uncertainty in the entering temperature voltage signal $V_{av.}$.

$$E_{slope} = dslope = \Big|\left(\vartheta\left(V_{av.} + dV\right) - \vartheta\left(V_{av.}\right)\right) - \left(\vartheta\left(V_{av.} + dV_{av.} + \Delta V\right) - \vartheta\left(V_{av.} + dV_{av.}\right)\right)\Big| \tag{A.12}$$

where:

$V_{av.}$	=	entering temperature voltage signal (μV)
$dV_{av.}$	=	uncertainty of the entering temperature voltage signal (μV)
AV	=	temperature difference voltage signal (μV)

Remembering that an additional uncertainty in the temperature difference is caused by the voltage measurement of the temperature difference voltage signal (AV), the uncertainty of the air duct temperature difference is given to be:

$$E_{\Delta\vartheta} = d\Delta\vartheta = \left[\left(\frac{\partial\vartheta}{\partial V}\, d\Delta V\right)^2 + dslope^2\right]^{112} \quad \textbf{(A.13)}$$

A. 4 Uncertainty of the Air Side Capacity

The air side capacity of the heat pump was evaluated using the equation:

$$\dot{Q}_C = \dot{Q}_S + QL \quad \textbf{(A.}14)$$

where:

$\dot{Q}_S$	=	sensible capacity, kW (*Btu/h*)
Q_L	=	latent capacity, kW (*Btu/h*)

The sensible capacity is the heat needed to cool or heat the moist air passing the heat pump's indoor coil. The latent capacity is the heat rejected by water vapor condensing on the air coil. Condensation does not occur in the heating mode.

The two different capacities were calculated separately and then added **(A.**14). Therefore the uncertainty of the air-side capacity can be written as:

$$E_{\dot{Q}_C} = \left[\left(\frac{\partial \dot{Q}_C}{\partial \dot{Q}_S}\, d\dot{Q}_S\right)^2 + \left(\frac{\partial \dot{Q}_C}{\partial \dot{Q}_L}\, d\dot{Q}_L\right)^2\right]^{1/2} = \left(d\dot{Q}_S^{\,2} + d\dot{Q}_L^{\,2}\right)^{1/2} \quad \text{(A.15)}$$

The equations for both the sensible and latent capacities and their uncertainties are presented on the following pages.

A. 4.1 Uncertainty of the Sensible Capacity

According to **ASHRAE** Standard 116-1993 the sensible capacity Q_S is given by:

$$\dot{Q}_s = 3600\ C_D\ A_n\ (0.24 + 0.444\ W_{av.})(\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1-\beta^2)}\right]^{1/2} \quad (A.16)$$

where:

C_D	=	nozzle discharge coefficient (0.986)
A_n	=	nozzle throat area, m^2 (ft^2)
$W_{av.}$	=	$(W_e + W_1)/2$ average humidity ratio, kg H_2O/ kg dry air (*lb H_2O/lb dry* ***air)***
$\vartheta_1 - \vartheta_e$	=	indoor coil air temperature rise, °C (^oF)
g_C	=	gravity constant (32.174 $ft \cdot lb_m / lb_f \cdot s^2$)
Δp_n	=	static pressure drop across nozzle, kPa *(psia)*
ρ_{nact}	=	density of the moist air, kg/m^3 (lb/ft^3)
144	=	unit conversion factor from in^2 to ft^2
β	=	area relation factor (0 for nozzle chamber)

The partial derivatives required for the uncertainty analysis of Q_S are:

$$\frac{\partial \dot{Q}_S}{\partial A_n} = 3600\ C_D\ (0.24 + 0.444\ W_{av.})(\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1-\beta^2)}\right]^{1/2} \quad (A.17)$$

$$\frac{\partial \dot{Q}_S}{\partial W_e} = 1800\ C_D\ A_n\ 0.444\ (\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1-\beta^2)}\right]^{1/2} \quad (A.18)$$

$$\frac{\partial Q_S}{\partial W_1} = 1800\ C_D\ A_n\ 0.444(\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1-\beta^2)}\right]^{1/2} \quad (A.19)$$

$$\frac{\partial \dot{Q}_S}{\partial(\vartheta_1 - \vartheta_e)} = 3600\ C_D\ A_n\ (0.24 + 0.444\ W_{av.})\left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1-\beta^2)}\right]^{1/2} \quad (A.20)$$

$$\frac{\partial \dot{Q}_S}{\partial \Delta p_n} = 1800\ C_D\ A_n\ (0.24 + 0.444\ W_{av.})(\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \rho_{nact}}{144(1-\beta^2)\Delta p_n}\right]^{1/2} \quad (A.21)$$

$$\frac{\partial \dot{Q}_S}{\partial \rho_{nact}} = 1800\ C_D\ A_n\ (0.24 + 0.444\ W_{av.})(\vartheta_1 - \vartheta_e)\left[\frac{2\ g_C\ \Delta p_n}{144(1-\beta^2)\rho_{nact}}\right]^{1/2} \quad (A.22)$$

$$\frac{\partial \dot{Q}_S}{\partial \beta} = 3600\ C_D\ A_n (0.24 + 0.444\ W_{av.})(\vartheta_l - \vartheta_e)\beta \left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144\ (1-\beta^2)^3}\right]^{112} \qquad \textbf{(A.23)}$$

Using the above partial derivatives for rewriting equation **A.3** gives:

$$E_{Q_S} = \left[\left(\frac{\partial \dot{Q}_S}{\partial A_n} dA_n\right)^2 + \left(\frac{\partial \dot{Q}_S}{\partial W_e} dW_e\right)^2 + \left(\frac{\partial \dot{Q}_S}{\partial W_l} dW_l\right)^2 + \left(\frac{\partial \dot{Q}_S}{\partial \Delta p_n} d\Delta p_n\right)^2 + \right.$$

$$\left. + \left(\frac{\partial \dot{Q}_S}{\partial (\vartheta_l - \vartheta_e)} d(\vartheta_l - \vartheta_e)\right)^2 + \left(\frac{\partial \dot{Q}_S}{\partial \rho_{nact}} d\rho_{nact}\right)^2 + \left(\frac{\partial Q_S}{\partial \beta} d\beta\right)\right]^{1/2} \qquad \textbf{(A.24)}$$

Equation **A.24** can be evaluated to give the uncertainty of $\dot{Q}_S$ if each of the individual uncertainties is known. However, **A,** β , W_e , W_l and ρ_{nact} are calculated quantities, so their uncertainties were not known, but had to be calculated using equation **A.3.**

The flow in the air duct was measured using an ASME nozzle. The nozzle throat area 4, which is part of equation **A.16,** was calculated from the throat diameter. Thus its uncertainty can be evaluated very easily.

$$A_n = \frac{\pi\ d_n^{\ 2}}{4} \qquad \textbf{(A.25)}$$

$$E_{A_n} = \frac{\partial A_n}{\partial d_n} dd_n = \frac{\pi\, d_n}{2} dd_n \qquad \textbf{(A.26)}$$

The uncertainty of the throat diameter was given to be: $E_{dn} = dd_n = \pm 0.254$ mm $= \pm$ **0.01** in.
The required uncertainty in the inlet diameter was also
$E_{d_{en}}$ = dd,, $= \pm 0.254$ mm $= \pm$ **0.01** in.

The humidity ratios W_e and W_l are **a** function **of** the water vapor pressure p_w and the atmospheric pressure p.

$$W = 0.62198 \cdot \frac{p_w}{p - p_w} \qquad \textbf{(A.27)}$$

The factor **0.62198** comes from the ratio of the mole weights of the two components, water and air.

The required partial derivatives of equation **A.27** are:

$$\frac{\partial W}{\partial p} = 0.62198 \frac{P_w}{(p - p_w)^2} \qquad \textbf{(A.28)}$$

$$\frac{\partial W}{\partial p_w} = 0.62198 \frac{P}{(p - p_w)^2} \qquad \textbf{(A.29)}$$

They lead to the uncertainty in W:

$$E, = dW = \left[\left(\frac{\partial W}{\partial p} dp \right)^2 + \left(\frac{\partial W}{\partial p_w} dp_w \right)^2 \right]^{1/2} \qquad \textbf{(A.30)}$$

The water saturation pressure is a calculated quantity itself, which means its uncertainty had to be calculated.

The equation that was used to calculate the saturation pressure from the dew-point temperature, T_{dew}, (°**R**), is given below. The equation was assumed to cause no additional uncertainties.

$$p_w = EXP \left[\frac{C_8}{T_{dew}} + C_9 + C_{10} \; T_{dew} + C_{11} \; T_{dew}{}^2 + C_{12} \; T_{dew}{}^3 + C_{13} \; \ln T_{dew} \right] \qquad \text{(A.31)}$$

The partial derivative of equation **A.31** with respect to T_{dew} is:

$$\frac{\partial p_w}{\partial T_{dew}} = \left[\frac{-C_8}{T_{dew}{}^2} + C_{10} + 2C_{11} \; T_{dew} + 3C_{12} \; T_{dew}{}^2 + \frac{C_{13}}{T_{dew}} \right] p_w \qquad \textbf{(A.32)}$$

The uncertainty in p_w is now given by:

$$E_{p_w} = dp_w = \frac{\partial p_w}{\partial T_{dew}} dT_{dew} \qquad \textbf{(A.33)}$$

As already mentioned in section **A.3,** the uncertainty of the dew-point temperature measurement was given to be: $E_{T_{dew}} = dT_{dew} = \pm 0.05\,\%$ of reading **.**

Finally, the uncertainty in the moist air's density ρ_{nact} had to be evaluated. The density was calculated using the ideal gas equation and the humidity ratio.

$$\rho_{nact} = \frac{p_n \; 144 \; (1 + W)}{R_a \; T_n \; (1 + 1.6078 \; W)} \qquad \textbf{(A.34)}$$

The factor 1.6078 is the ratio of the molar weights of air and water.

The partial derivatives of equation A.34 are:

$$\frac{\partial \rho_{\text{nact}}}{\partial p_{\text{n}}} = \frac{144\,(1+W)}{R\,T_{\text{n}}\,(1+1.6078\,W)} \tag{A.35}$$

$$\frac{\partial \rho_{\text{nact}}}{\partial T_{\text{n}}} = \frac{-p_{\text{n}}\;144\,(1+W)}{R\,T_{\text{n}}^{2}\,(1+1.6078\,W)} \tag{A.36}$$

$$\frac{\partial \rho_{\text{nact}}}{\partial W} = \frac{-0.6078\,p_{\text{n}}\;144}{R\,T_{\text{n}}\,(1+1.6078\,W)^{2}} \tag{A.37}$$

Rewriting equation A.3 with the above partial derivatives gives:

$$E_{\rho_{\text{nact}}} = \left[\left(\frac{\partial \rho_{\text{nact}}}{\partial p_{\text{n}}}\,dp_{\text{n}}\right)^{2} + \left(\frac{\partial \rho_{\text{nact}}}{\partial T_{\text{n}}}\,dT_{\text{n}}\right)^{2} + \left(\frac{\partial \rho_{\text{nact}}}{\partial W}\,dW\right)^{2}\right] \tag{A.38}$$

The pressure p_{n} in the nozzle throat was calculated as the difference of atmospheric pressure and nozzle pressure drop. The uncertainty of the nozzle pressure can be derived as follows:

$$p_{\text{n}} = p_{\text{atm}} - \Delta p \tag{A.39}$$

$$E_{p_{\text{n}}} = dp_{\text{n}} = \left[(dp_{\text{atm}})^{2} + (d\Delta p)^{2}\right]^{1/2} \tag{A.40}$$

The uncertainties of the pressure measurements were given from manufacturer data:
$E_{p_{\text{atm}}} = dp_{\text{atm}} = M.3429$ mm Hg $= \pm\ 0.0135$ in Hg and
$E_{Dp_{\text{n}}} = dDp_{\text{n}} = \pm 2.489$ mm $H_2O = \pm\ 0.098$ in H_2O.

A. 4.2 Uncertainty of the Latent Capacity

The latent cooling capacity (ASHRAE Standard 116-1983)is given **by:**

$$\dot{Q}_{\text{L}} = 63600\;60\;C_{\text{D}}\;A_{\text{n}}\left(W_{\text{e}} - W_{\text{l}}\right)\left[\frac{2\,g_{\text{C}}\,\Delta p_{\text{n}}\,\rho_{\text{nact}}}{144\left(1-\beta^{2}\right)}\right]^{1/2} \tag{A.41}$$

where:

C_{D} = nozzle discharge coefficient (0.986)

A_n	=	nozzle throat area (ft^2)
W_e	=	entering humidity ratio (lb H_2O/lb dry air)
W_l	=	leaving humidity ratio (lb H_2O/lb dry air)
g_C	=	gravity constant (32.174 $ft \cdot lb_m / lb_f \cdot s^2$)
Δp_n	=	static pressure drop across nozzle (psia)
ρ_{nact}	=	density of the moist air (lb / ft^3)
144	=	unit conversion factor from in^2 to ft^2
β	=	area relation factor (0 for a nozzle chamber)

The partial derivatives of this equation are:

$$\frac{\partial \dot{Q}_L}{\partial A_n} = 63600\ 60\ C_D\ (W_e - W_l) \left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144\ (1 - \beta^2)} \right]^{1/2} \tag{A.42}$$

$$\frac{\partial \dot{Q}_L}{\partial W_e} = 63600\ 60\ C_D\ A_n \left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1 - \beta^2)} \right]^{1/2} \tag{A.43}$$

$$\frac{\partial \dot{Q}_L}{\partial W_l} = -63600\ 60\ C_D\ A_n \left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1 - \beta^2)} \right]^{1/2} \tag{A.44}$$

$$\frac{\partial \dot{Q}_L}{\partial \Delta p_n} = 31800\ 60\ C_D\ A_n\ (W_e - W_l) \left[\frac{2\ g_C\ \rho_{nact}}{144(1 - \beta^2)\Delta p_n} \right]^{1/2} \tag{A.45}$$

$$\frac{\partial \dot{Q}_L}{\partial \rho_{nact}} = 31800\ 60\ C_D\ A_n\ (W_e - W_l) \left[\frac{2\ g_C\ \Delta p_n}{144(1 - \beta^2)\rho_{nact}} \right]^{1/2} \tag{A.46}$$

$$\frac{\partial \dot{Q}_L}{\partial \beta} = 63600\ 60\ C_D\ A_n\ (W_e - W_l)\beta \left[\frac{2\ g_C\ \Delta p_n\ \rho_{nact}}{144(1 - \beta^2)^3} \right]^{1/2} \tag{A.47}$$

If the above derivatives are used to rewrite equation A.3, one obtains the uncertainty of the latent capacity:

$$E_{Q_L} = \left[\left(\frac{\partial \dot{Q}_L}{\partial A_n} dA_n \right)^2 + \left(\frac{\partial \dot{Q}_L}{\partial W_e} dW_e \right)^2 + \left(\frac{\partial \dot{Q}_L}{\partial W_l} dW_l \right)^2 + \right.$$

$$+\left(\frac{\partial \dot{Q}_{\mathrm{L}}}{\partial \Delta p_{\mathrm{n}}} d\Delta p_{\mathrm{n}}\right)^2 + \left(\frac{\partial \dot{Q}_{\mathrm{L}}}{\partial \rho_{\mathrm{nact}}} d\rho_{\mathrm{nact}}\right)^2 + \left(\frac{\partial \dot{Q}_{\mathrm{L}}}{\partial \beta} d\beta\right)^2\Bigg]^{1/2} \qquad \textbf{(A.48)}$$

In this equation, all the needed uncertainties are known. Either because the quantities are directly measured or their uncertainties have already been calculated in Appendix A.4.1.

The final step was calculating the uncertainty **of** the air-side capacity by using the now known uncertainties of sensible and latent capacity in equation A. 15.

A. 5 Uncertainty of the COP

To calculate the COP's uncertainty it is necessary to know the uncertainties of the air-side capacity,& and the mechanical power, P.

$$COP = \frac{\dot{Q}}{P} \qquad \textbf{(A.49)}$$

The uncertainty of the COP is determined by:

$$E_{COP} = \left[\left(\frac{\partial COP}{\partial \dot{Q}} d\dot{Q}\right)^2 + \left(\frac{\partial COP}{\partial P} dP\right)^2\right]^{1/2} = \left[\left(\frac{d\dot{Q}}{P}\right)^2 + \left(-\frac{\dot{Q}}{P^2} dP\right)^2\right]^{1/2} \qquad \text{(A.50)}$$

All of these components are directly measured or know from the above calculations.

A. 6 Uncertainty Analysis Results for Selected Tests

Table A.1 gives an example of the error associated with COP and air-side capacity for several tests.

Table **A.**1: Measurement uncertainty for typical tests

Filename			Value	Percent Uncertainty at a 95 % Confidence Limit on the Mean
F010411B	ARI Standard Cooling:	EER	13.29 ± 0.58	3.91
		Capacity	2364 ± 52 W (8068 ± 178 Btu/h)	3.83
F010412A	ARI Standard Heating	COP	4.97 ± 0.19	3.29
		Capacity	3317 ± 122 W (11317 ± 418 Btu/h)	3.20
F010412B	ISO Standard Heating	COP	5.02 ± 0.19	3.32
		Capacity	3278 ± 122 W (11184 ± 417 Btuh)	3.23
F010413A	ISO Standard Cooling	EER	13.01 ± 0.60	4.00
		Capacity	2338 ± 105 W (7977 ± 360 Btuh)	3.92
W010607A	ISO Standard Cooling	EER	13.13 ± 0.51	5.21
		Capacity	2920 ± 111 W (9964 ± 379 Btuh)	5.17
W010611A	ARI Standard Heating	COP	4.17 ± 0.18	5.82
		Capacity	4208 ± 180 W (14357 ± 613 Btu/h)	5.80
WO10612A	ISO Standard Heating	COP	4.26 ± 0.18	5.86
		Capacity	4139 ± 178 W (14121 ± 607 Btu/h)	5.84
WO10620A	ARI Standard Cooling	EER	14.18 ± 0.47	4.44
		Capacity	3085 ± 100 W (10528 ± 340 Btu/h)	4.39

Appendix B: Heat Pump Test Data

This appendix provides detailed tests data for tests performed at NIST. The designation of the heat pumps tested (Unit 1 and Unit *2)* is consistent with that used in Section 5.

Table B.1: Unit 1, data #1

Description	Filename	Water-In		Indoor-In		Indoor Dew		Air-DT		Water-DT	
Cooling Tests		F	C	F	C	F	C	F	C	F	C
ISO Standard Repeat	F010502A	85.8	29.9	80.7	27.1	58.4	14.7	19.28	10.71	9.87	5.48
ISO -10% Water DP	F010425A	86.0	30.0	80.4	26.9	58.5	14.7	19.02	10.57	9.85	5.47
ISO +10% Water DP	F010426A	85.9	30.0	80.8	27.1	58.4	14.7	19.29	10.72	9.27	5.15
ISO Standard Repeat	F010502A	85.8	29.9	80.7	27.1	58.4	14.7	19.28	10.71	9.87	5.48
ISO +10% Air SCFM	F010502B	85.9	29.9	80.4	26.9	58.4	14.7	18.30	10.17	9.94	5.52
ISO -10% Air SCFM	F010426C	86.0	30.0	80.8	27.1	58.6	14.8	19.87	11.04	9.60	5.33
ARI Standard Repeat	F010501A	85.1	29.5	79.8	26.6	60.1	15.6	17.91	9.95	9.96	5.53
Heating Tests											
ISO Standard Repeat	F010509A	67.9	19.9	67.8	19.9	45.8	7.7	34.48	19.15	8.57	4.76
ISO +10% Water DP	F010427A	67.9	19.9	68.1	20.1	42.6	5.9	34.41	19.12	7.86	4.37
ISO -10% Water DP	F010427B	68.1	20.0	68.0	20.0	42.5	5.8	33.97	18.87	9.84	5.47
ISO Standard Repeat	F010509A	67.9	19.9	67.8	19.9	45.8	7.7	34.48	19.15	8.57	4.76
ISO +10% Air SCFM	F010427C	68.0	20.0	67.6	19.8	42.2	5.7	32.94	18.30	8.53	4.74
ISO -10% Air SCFM	F010430A	68.0	20.0	67.9	20.0	44.5	6.9	36.04	20.02	8.40	4.67
ARI Standard Repeat	F010507A	69.9	21.1	70.1	21.2	46.6	8.1	34.76	19.31	8.62	4.79

Filename	In-H20	Pa	Psid	Pa	cfm	L/s
F010502A	0.32	79.7	1.924	13265.5	299.30	141.25
F010425A	0.32	79.7	1.925	13272.4	298.79	141.01
F010426A	0.329	81.9	1.601	11038.5	299.33	141.26
F010502A	0.32	79.7	1.924	13265.5	299.30	141.25
F010502B	0.267	66.5	1.956	13486.1	328.00	154.79
F010426C	0.402	100.1	2.053	14154.9	269.47	127.17
F010501A	0.312	77.7	1.897	13079.3	297.55	140.42
F010509A	0.401	99.8	1.933	13327.5	306.67	144.73
F010427A	0.362	90.1	2.355	16237.1	314.60	148.47
F010427B	0.366	91.2	1.449	9990.5	314.77	148.55
F010509A	0.401	99.9	1.933	13327.5	306.67	144.73
F010427C	0.329	82.0	2.046	14106.6	331.24	156.32
F010430A	0.427	106.4	2.028	13982.5	287.19	135.53
F010507A	0.402	100.1	1.929	13300.0	308.63	145.65

	Water Mass Flow		Water Flow		Sensible Capacity		Latent Capacity		SHR	Total Capacity		Water-side Cap	
	lb/h	kg/h	GPM	Us	Btu/h	W	Btu/h	W		Btdh	W	Btu/h	
F010502A	1003.1	455.0	2.00	0.126	6383	1871	1677	492	0.792	8060	2362	7794	2284
F010425A	1004.1	455.4	2.01	0.127	6292	1844	1648	483	0.792	7940	2327	7776	2279
F010426A	1071.5	486.0	2.14	0.135	6458	1893	1528	448	0.809	7986	2340	7831	2295
F010502A	1003.1	455.0	2.00	0.126	6383	1871	1677	492	0.792	8060	2362	7794	2284
F010502B	1001.6	454.3	2.00	0.126	6633	1944	1389	407	0.827	8022	2351	7827	2294
F010426C	1026.0	465.4	2.05	0.129	5940	1741	1539	451	0.794	7480	2192	7758	2274
F010501A	1001.7	454.4	2.00	0.126	5898	1729	2082	610	0.739	7981	2339	7885	2311
F010509A	979.8	444.4	1.96	0.123	10884	3190	0	0	1	10884	3190	10620	3112
F010427A	1075.6	487.9	2.15	0.136	11149	3267	0	0	1	11149	3267	10679	3130
F010427B	843.1	382.4	1.68	0.106	11008	3226	0	0	1	11008	3226	10506	3079
F010509A	979.8	444.4	1.96	0.123	10884	3190	0	0	1	10884	3190	10620	3112
F010427C	992.1	450.0	1.98	3.125	11266	3302	0	0	1	11266	3302	10662	3125
F010430A	994.7	451.2	1.99	0.125	10640	3118	0	0	1	10640	3118	10596	3105
F010507A	980.5	444.7	1.96	0.124	10997	3223	0	0	1	10997	3223	10728	3144

Table B.4: Unit 1, data #4

	Average Capacity Air + Water		Total Power	Uncorrected EER1	Uncorrected COP	Fan Power and Heat Adjustrnent		Fan Heat
Filename	Btu/h	W	W	Btu/Wh	W/W	W	Btu/h	% Tot Cap
F010502A	7926.99	2323	613	12.931	3.790	37.529	128.064	1.616
F010425A	7857.95	2303	614.4	12.790	3.748	37.465	1127.846	1.627
F010426A	7766.17	2318	616.8	12.591	3.690	38.588	131.679	1.696
F010502A	7926.99	2323	613	12.931	3.790	37.529	128.064	1.616
F010502B	7924.61	2322	620.2	12.778	3.745	34.316	117.100	1.478
F010426C	7618.85	2233	608.9	12.512	3.667	42.447	144.846	1.901
F010501A	7932.74	2325	608.2	13.043	3.823	36.377	124.132	1.565
F010509A	10751.75	3151	650.8	16.521	4.842	48.187	164.432	1.529
F010427A	10913.67	3198	653.7	16.695	4.893	44.625	152.278	1.395
F010427B	10757.15	3153	647.9	16.603	4.866	45.142	154.044	1.432
F010509A	10751.75	3151	650.8	16.521	4.042	48.187	164.432	1.529
F010427C	10964.14	3213	646.1	16.970	4.973	42.702	145.717	1.329
F010430A	10617.73	3112	656.2	16.181	4.742	48.052	163.971	1.544
F010507A	10862.53	3183	667.2	16.281	4.771	48.615	1165.896	1.527

Filename	Fan Only Corrected Tot Power (W)	Fan Corrected Capacity (Btu/h)	W	Pump Power (W) Adjustment	Pump&Fan Corrected Tot Power (W)	W/W Corrected COP	Btu/Wh Corrected EER4
F010502A	575.5	8055	2361	5.590	581.061	4.062	13.863
F010425A	576.9	7986	2340	5.599	582.534	4.017	13.709
F010426A	572.5	8036	2355	4.371	582.583	3.973	13.557
F010502A	575.5	8055	2361	5.590	581.061	4.062	13.863
F010502B	585.9	8042	2357	5.675	591.559	3.984	13.594
F010426C	566.5	7764	2275	6.102	572.555	3.974	13.560
F010501A	571.8	8057	2361	5.504	577.327	4.090	13.955
F010509A	602.6	10587	3103	5.485	608.098	5.102	17.411
F010427A	609.1	10761	3154	7.338	616.413	5.116	17.458
F010427B	602.8	10603	3107	3.538	606.296	5.125	17.488
F010509A	602.6	10587	3103	5.485	608.098	5.102	17.411
F010427C	603.4	10818	3171	5.880	609.278	5.203	17.756
F010430A	608.1	10454	3064	5.843	613.991	4.989	17.026
F010507A	618.6	10697	3135	5.479	624.064	5.023	17.140

Table B.6: Unit 2. data #1

Description	Filename	Water-In		Indoor-In		Dewpoint		Air-DT		Water-DT	
Cooling Tests		F	C	F	C	F	C	F	C	F	C
ISO Standard Repeat	W010614A	85.9	30.0	80.9	27.2	58.7	14.8	22.38	12.43	9.60	5.33
ISO +20% GPM	W010614B	86.0	30.0	80.5	26.9	58.5	14.7	22.28	12.38	7.99	4.44
ISO -20% GPM	W010614C	86.0	30.0	80.6	27.0	58.1	14.5	22.33	12.40	12.01	6.67
ARI Standard 350scfm 1 ODT	W010620A	84.9	29.4	80.4	26.9	60.3	115.7	20.94	11.63	10.45	5.81
Heating Tests											
ISO Standard 300cfm	W010612A	68.0	20.0	68.1	20.0	40.1	4.5	45.06	25.03	7.80	4.33
ISO 300cfm +20% GPM	W010612B	68.0	20.0	68.3	20.1	40.6	4.8	45.53	25.30	6.49	3.61
ISO 300cfm -20% GPM	W010612C	68.0	20.0	68.1	20.1	40.5	4.7	44.23	24.57	9.69	5.38
ARI Standard 300cfm	W010611A	70.0	21.1	70.2	21.2	40.5	4.7	45.76	25.42	8.21	4.56

Table B.7: Unit 2, data #2

Filename	in H20 Air-DP	Pa Air-DP	Psid Water-DP	Pa Water-DP	Air-cfm	Air-Us
W010614A	0.011	2.740	3.158	21773.6	331.81	156.59
W010614B	0.011	2.740	4.436	30585.1	333.21	157.25
W010614C	0.015	3.736	2.089	14403.1	335.41	158.29
W010620A	0.01	2.491	2.845	19615.5	345.14	162.88
W010612A	0.094	23.414	4.014	27675.5	309.74	146.18
W010612B	0.095	23.663	5.667	39072.5	309.95	146.27
W010612C	0.096	23.913	2.693	18567.5	309.74	146.18
W010611A	0.093	23.165	3.597	24800.4	311.54	147.02

Filename	Water Mass Flow		Water Volume Flow		Sensible Capacity		Latent Capacity		SHR	Total Capacity		Water-Side Capacit	
	lb/h	kg/h	GPM	L/s	Btuh	W	Btuh	W		Btu/h	W	Btu/h	W
W010614A	1348.7	611.7	2.69	0.170	8275	2425	2191	642	0.791	10466	3067	10357	3035
W010614B	1618.1	734.0	3.23	0.204	8280	2427	2093	613	0.798	10373	3040	10391	3045
W010614C	1076.2	488.1	2.15	0.136	8345	2446	1922	563	0.813,	10266	3009	10277	3012
W010620A	1267.0	574.7	2.53	0.160	8046	2358	2481	727	0.764	10528	3085	10686	3132
W010612A	1449.1	657.3	2.90	0.183	14121	4139	0	0	1	14121	4139	14612	4282
W010612B	1764.4	800.3	3.52	0.222	14272	4183	0	0	1	14272	4183	14796	4336
W010612C	1145.7	519.7	2.29	0.144	13883	4069	0	0	1	13883	4069	14373	4212
W010611A	1380.5	626.2	2.76	0.174	14357	4208	0	0	1	14357	4208	14777	4331

Table **B.9:** Unit 2, data #4

Filename	Averaged Capacity		Total Power	Uncorrected EER1	Uncorrected COP	Pump Power Adjustment	Corrected Total Power	Corrected EER2	Corrected COP
	Btu/h	W	W	Btu/Wh	W/W	W	W	Btu/Wh	W/W
W010614A	10412	3051	752	13.85	4.06	12.3	764	13.63	3.99
W010614B	10382	3043	739	14.05	4.12	20.8	760	13.66	4.00
W010614C	10272	3010	769	13.36	3.92	6.5	775	13.25	3.88
W010620A	10607	3109	742	14.18	4.16	10.4	753	14.09	4.13
W010612A	14367	4210	972	14.78	4.33	16.8	989	14.53	4.26
W010612B	14534	4259	973	14.94	4.38	29.0	1002	14.51	4.25
W010612C	14128	4140	960	14.72	4.32	8.9	968	14.59	4.28
W010611A	14567	4269	1010	14.22	4.17	14.4	1024	14.22	4.17

Appendix C: Summary of Manufacturer and NIST Test Data

The two units tested at NIST were selected for this study from the pool of fifteen water-source heat pumps tested by their respective manufactures according to the ARI and ISO test procedures. Test results obtained by these manufacturers were submitted to DOE by ARI in support of ARI's comments on the DOE's proposed rule making regarding test procedures and efficiency standards for commercial air conditioners and heat pumps (DOE-1999). The tables in this appendix present manufacturers' test results and comparison of manufacturers' and NIST relative ratings obtained by the two test methods. On average, the disparity between the ARI and ISO ratings obtained by NIST is smaller than that obtained by the two manufacturers. The designation of the heat pumps tested (Unit 1 and Unit 2) is consistent with that used in the main body of this report.

Cooling	Values
Using ISO 13256-1: Uncorrected Capacity, W (Btuh)	2149 (7333)
Uncorrected Total Power, W	NA
Uncorrected EER	11.75
Water Flow, L/s (gpm)	0.121 (1.92)
Water Coil Temp Change, °C (°F)	5.56 (10.0)
Water Pressure Drop, Pa (psid)	11358 (1.65)
Air Flow, L/s (cfm)	142 (300)
Air Static, Pa (in H_2O)	97 (0.39)
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	46 (156)
ISO Power Adjustment: For Fan Power, W	46
For Pump Power, W	5
Corrected Capacity, W (Btu/h)	2195 (7489)
Corrected EER	12.84
Using ARI 320 Capacity, W (Btu/h)	2181 (7443)
Total Power, W	NA
EER	12.04

	Manufacturer				NIST			
	Capacity W	Capacity % Difference wrt ARI 320	EER	EER % Difference wrt ARI 320	Capacity W (Btu/h)	Capacity % Difference from ARI 320	EER	EER % Difference wrt ARI 320
ARI 320	2181 (7443)	NA	12.04	NA	2352 (8024)		13.21	NA
ISO Raw	2149 (7333)	-1.5	11.75	-2.4	2316 (7902)	-1.5	12.89	-1.5
ISO Corrected	2195 (7489)	0.6	12.84	6.6	2353 (8028)	0.04	13.79	5.3

	NIST Capacity % Difference wrt Manufacturer	NIST EER % Difference wrt Manufacturer
ARI 320	7.8	9.7
ISO Raw	7.8	9.8
ISO Corrected	7.2	7.5

Table C.3: Manufacturer heating test results for Unit I

Heating	Values
<u>Using ISO 13256-1:</u> Uncorrected Capacity, W (Btu/h)	2890 (9860)
Uncorrected Total Power, W	NA
Uncorrected COP	4.6
Water Flow, L/s (gpm)	0.119 (1.89)
Water Coil Temp Change, °C (°F)	5.56 (10.0)
Water Pressure Drop, Pa (psid)	11376 (1.65)
Air Flow, L/s (cfm)	139 (294)
Air Static, Pa (in H_2O)	97 (0.39)
<u>ISO Capacity Adjustment:</u> For Fan Heat, W (Btu/h)	45 (153)
<u>ISO Power Adjustment:</u> For Fan Power, W	45
For Pump Power, W	5
Corrected Capacity, W (Btu/h)	2845 (9706)
Corrected COP	4.84
<u>Using ARI 320</u> Capacity, W (Btu/h)	2942 (10037)
Total Power, W	NA
COP	4.52

Table C.4: Manufacturer and NIST heating test results for Unit 1

	Manufacturer				NIST			
	Capacity W (Btu/h)	Capacity % Difference from ARI 320	COP	COP % Difference from ARI 320	Capacity W (Btu/h)	Capacity % Difference from ARI 320	COP	COP % Difference from ARI 320
ARI 320	2942 (10037)	NA	4.52	NA	3237 (11045)	NA	4.85	NA
ISO Raw	2890 (9860)	-1.8	4.6	1.8	3200 (10917)	-1.15	4.90	1.01
ISO Corrected	2845 (9706)	-3.3	4.84	7.1	3153 (10760)	-1.44	5.14	6.05

	NIST Capacity % Difference wrt Manufacturer	NIST COP % Difference wrt Manufacturer
ARI 320	10.04	7.31
ISO Raw	10.72	6.52
ISO Corrected	10.86	6.28

Table C.5: Manufacturer cooling: test results for Unit 2

Cooling	Values
Using ISO 13256-1: Uncorrected Capacity, W (Btu/h)	3177 (10839)
Uncorrected Total Power. W	NA
Uncorrected EER	12.81
Water Flow, L/s (gpm)	0.196 (3.1)
Water Coil Temp Change, °C (°F)	4.99 (8.99)
Water Pressure Drop, Pa (psid)	22717 (3.29)
Air Flow, L/s (cfm)	160 (340)
Air Static. Pa (in H_2O)	0
ISO Capacity Adjustment: For Fan Heat, W (Btu/h)	0
ISO Power Adjustment: For Fan Power, W	0
For Pump Power, W	15
Corrected Capacity, W (Btu/h)	3177 (10839)
Corrected EER	12.58
Using ARI 320 Capacity, W (Btu/h)	3341 (11399)
Total Power, W	NA
EER	13.67

:ooling test results for Unit **2**

	Manufacturer				NIST			
	Capacity W	Capacity % Difference from ARI 320	EER	EER % Difference from ARI 320	Capacity W (Btu/h)	Capacity % Difference from ARI 320	EER	EER % Difference from ARI 320
ARI 320	3341 (11399)	NA	13.67	NA	3085 (10528)	NA	14.18	NA
ISO Raw	3177 (10839)	-4.9	12.81	-6.3	3051 (10412)	-1.1	13.85	-2.3
ISO Corrected	3177 (10839)	-4.9	12.58	-8.0	3051 (10412)	-1.1	13.63	-3.9
	NIST Capacity % Difference wrt Manufacturer				NIST EER % Difference wrt Manufacturer			
ARI 320	-7.7				3.7			
ISO Raw	-4.0				8.1			
ISO Corrected	-4.0				8.3			

Table C.7: Manufacturer heating: test results for Unit 2

Heating	Values
<u>Using ISO 13256-1:</u> Uncorrected Capacity, W **(Btu/h)**	4572 (15599)
Uncorrected Total Power, W	**NA**
Uncorrected COP	4.67
Water Flow, L/s (gpm)	0.195 (3.1)
Water Coil Temp Change, °C (°F)	4.44 (8.0)
Water Pressure Drop, Pa (psid)	28158 (4.08)
Air Flow, L/s (cfm)	186 (395)
Air Static, Pa (in H_2O)	0
<u>ISO Capacity Adjustment:</u> For Fan Heat, W **(Btu/h)**	0
<u>ISO Power Adjustment:</u> For Fan **Power, W**	0
For Pump Power, W	18
Corrected Capacity, W (Btu/h)	4572 (15599)
Corrected COP	4.59
<u>**Using** ARI 320</u> Capacity, W (Btu/h)	4641 **(15837)**
Total Power, W	NA
COP	4.59

Table C.8: Manufacturer and NIST heating test results for Unit 2

	Manufacturer				NIST			
	Capacity W (Btu/h)	Capacity % Difference from ARI 320	COP	COP % Difference from ARI 320	Capacity W (Btu/h)	Capacity % Difference from ARI 320	COP	COP % Difference from ARI 320
ARI 320	4641 (15837)	NA	4.59	NA	4668 (15927)	NA	4.94	NA
ISO Raw	4572 (15599)	-1.5	4.67	I.7	4534 (15469)	-2.9	4.98	0.8
ISO Corrected	4572 (15599)	-1.5	4.59	0.0	4534 (15469)	-2.9	4.87	-1.4

	NIST Capacity % Difference wrt Manufacturer	NIST COP % Difference wrt Manufacturer
ARI 320	0.6	7.6
ISO Raw	-0.8	6.6
ISO Corrected	-0.8	6.1