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Abstract

With air conditioning equipment used by 100 million homes each year, accurate energy efficiency metrics are paramount for promulgation of domestic and international energy policy, determination of Standards impact, and forecasting for energy security. Testing Standards and measurement policies stipulate allowable measurement uncertainty for certification of products consumed by the public, however, the impact of those stipulations on the accuracy of the certified cooling energy efficiency ratio for split-system air conditioners is not well characterized. This research propagates published measurement equipment uncertainties and experimental data strictly through those prescribed equations in the Testing Standards that govern EER. Illustration of the results in a novel visualization technique revealed that uncertainties in the measurement of barometric pressure and power had a larger impact on the uncertainty of EER than those of temperature, humidity, or nozzle diameter, and were those measurements that, whereupon improved, would be most effective in reducing the uncertainty of EER to improve measurement policy for residential cooling equipment.

Keywords: measurement uncertainty; uncertainty propagation; air conditioner; impact analysis; energy efficiency

I. Introduction

With air conditioners used in 100 million U.S. homes [1] and contributing 3.34 quads of energy per year to national energy demand [2], the accuracy of the cooling energy efficiency ratio (EER) is vital to producing a confident national energy consumption forecast model. Measurement policies regulate EER uncertainty through measurement device tolerances, however, such policies, being only reaffirmed since having been first published in the 1980s, may give readers pause as to whether the uncertainty can be reduced given technological improvements in metrology. To create better and more certain national energy forecasts, the impacts of measurement uncertainty on EER must be determined and potential improvements to measurement policies explored.

Shen characterized the uncertainty of EER by propagating the uncertainty of eight variables through the governing equations for EER [3]. While this approach obtained a 12% uncertainty in EER, the work did not select variables and uncertainty values that were representative of those prescribed in measurement policies. Bullard et al. determined EER uncertainty by simulating probability distributions of variables' measurement uncertainty based on measurement policies [4]. Although this approach applied values from measurement policies to EER-determining equations, the academic fluid dynamic and heat transfer equations that were used were not consistent with the equations prescribed in test procedure policy. Diaz implemented the measurement policies of ISO 5151-1994 and ANSI/ASHRAE Standard 16-1999 in equations from test procedure policy and found a 3.4% uncertainty in total cooling capacity [5]. However, his work on through-the-wall or -window air conditioning products does not directly translate to split-system products.

By employing the variables, uncertainty values, and equations used in measurement and test procedure policy, this research intends to investigate the extent to which the uncertainty of U.S. Standards-prescribed measurement devices impacts the uncertainty of EER via the Department of Energy (DOE) test procedure. This paper first introduces the equations from the test procedure policy and then performs an uncertainty analysis on such equations with measured data. The results follow, along with a discussion.

II. Determining EER

EER is a ratio of the energy out of a system to the energy in, and many approaches exist that measure energy in and energy out. ANSI/ASHRAE Standard 37-2009 has standardized the variations to five approaches: four optional approaches and one mandatory approach, which is the one used in this analysis—the Indoor Air Enthalpy method [6].

Title 10 Part 430 Appendix M to Subpart B of the Code of Federal Regulations and ANSI/ASHRAE Standard 37-2009 Section 7.3.3.1 specify the Indoor Air Enthalpy method. The EER is the ratio of the test unit's total indoor cooling capacity, $\dot{q}_{tci}$, to the measured total power input at 82°F dry-bulb outside air temperature, $\dot{E}_{(82)}$:

$$EER = \frac{\dot{q}_{tci}}{\dot{E}_{(82)}}, \tag{1}$$

where the cooling capacity is defined by:

$$\dot{q}_{tci} = 60\dot{Q}_{mi}\frac{h_{a1} - h_{a2}}{v_{nsp}}, \tag{2}$$

and $\dot{Q}_{mi}$ is the measured indoor airflow rate, h_{a1} is the enthalpy of the air entering the indoor side, h_{a2} is the enthalpy of the air leaving the indoor side, v_{nsp} is the specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure at the nozzle(s) exit.

The total power input is the sum of the power consumption of three devices—the outdoor fan, $\dot{E}_{(82)\_ofan}$, the indoor fan, $\dot{E}_{(82)\_ifan}$, and the compressor, $\dot{E}_{(82)\_compr}$—represented as:

$$\dot{E}_{(82)} = \dot{E}_{(82)\_ofan} + \dot{E}_{(82)\_ifan} + \dot{E}_{(82)\_compr}\ . \tag{3}$$

Other electrical power from controls is negligible.

The measured indoor airflow rate is determined by measuring pressure drop across the nozzle(s), ΔP_V, and utilizing the compressible Bernoulli equation of flow through an orifice, as directed by ANSI/ASHRAE Standard 37-2009 Section 7.7.2.1,

$$\dot{Q}_{mi} = CA_n\sqrt{2 \cdot \Delta P_V \cdot v_n'}, \tag{4}$$

where C is the nozzle discharge coefficient, A_n is the effective nozzle throat area, and v_n' is the specific volume of air at the nozzle in units per air-water vapor mixture.

The nozzle discharge coefficient is given by the approximation for the nozzle coefficient, based on a length-to-nozzle-diameter ratio of 0.6 and pressure taps measuring static and velocity pressure at the nozzle exit, as directed by ANSI/ASHRAE Standard 51-2007 Section 7.3.2.6 [7]:

$$C = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}. \tag{5}$$

where the Reynolds number, Re, is:

$$Re = D_{AVG}V\ \frac{\rho}{\mu}, \tag{6}$$

$$\left(Re = D_{AVG}V\frac{\rho}{60\mu}\right), \tag{6 I-P}$$

and ρ is the density and μ is the dynamic viscosity of the moist air at the nozzle. D_{AVG} is the mean of eight diameter measurements of the nozzle throat, per ANSI/ASHRAE Standard 37-2009 Section 5.3.3, that is used also in the equation for the nozzle throat area, or, if multiple nozzles, the effective nozzle throat area:

$$A_n = \pi\left(\frac{D_{AVG}}{2}\right)^2. \tag{7}$$

For the uncertainty of the nozzle diameter measurement, D_{STD} is the uncertainty of eight measurements, with each measurement having the diameter tolerance, D_{TOL}. D_{STD} and D_{TOL} are related as such:

$$D_{STD} = \frac{D_{TOL}}{\sqrt{8}}. \tag{8}$$

In the Reynolds number, the throat airflow velocity, V, is determined from the definition of volumetric flow rate:

$$V = \frac{\dot{Q}_{mi}}{A_n}, \tag{9}$$

where Eqn. 4, the measured indoor airflow rate, with Eqn. 9, can be reduced to:

$$V = C\sqrt{2 \cdot \Delta P_V \cdot v_n'}. \tag{10}$$

The specific volume of air at the nozzle in units per air-water vapor mixture, v_n', is determined from ANSI/ASHRAE Standard 37 Section 7.7.2.1:

$$v_n' = \frac{v_{nsp}}{1 + W_2}, \tag{11}$$

where W_2 is the measured humidity ratio of the air leaving the indoor side and v_{nsp}, the specific volume of dry air at standard pressure (in SI units), is calculated by referencing ANSI/ASHRAE Standard 41.6-2006 Equation 23 [8],

$$v_{nsp} = R_a\frac{t_{a2}}{P_{bar}}(1 + 1.6078W_2), \tag{12}$$

where R_a is the air-specific gas constant, t_{a2} is the measured dry-bulb temperature of the air leaving the indoor side (in Kelvin), and P_{bar} is the measured barometric pressure (in Pascal). Eqn. 11 and 12 can be combined into one equation for simplicity:

$$\upsilon_n' = \frac{R_a t_{a2}(1+1.6078W_2)}{P_{bar}(1+W_2)}. \quad (13)$$

With all airflow and pressure variables defined for Eqn. 4, the indoor airflow rate, there remains the enthalpy parameters of Eqn. 2. ANSI/ASHRAE Standard 41.6-2006 Equation 27 provides an approximation for calculating moist air enthalpy, h, from fundamental state parameters via the Ideal Gas Law (SI units). We substitute the measured dry-bulb temperature of the air entering and leaving the indoor side (in Celsius), t_{a1} and t_{a2}, respectively, and the measured humidity ratio of the air entering and leaving the indoor side, W_1 and W_2, respectively, to obtain:

$$h_{a1} = 1.005t_{a1} + W_1(2500.9 + 1.805t_{a1}), \quad (14)$$

$$h_{a2} = 1.005t_{a2} + W_2(2500.9 + 1.805t_{a2}). \quad (15)$$

The humidity ratios are found by iteratively solving for them after measuring the dew point of the air and using the relation between partial pressure and humidity ratio. Because the partial pressure of saturated water vapor in air is equal to moist air's saturation pressure, for perfect gases, the following relation from the 2009 ANSI/ASHRAE Handbook—Fundamentals Equation 38 [9], can be utilized:

$$p_{ws}(t_{dp}) = \frac{P_{bar}\cdot W}{0.621945+W}, \quad (16)$$

where p_{ws} is the saturation pressure, at a given dew point temperature, t_{dp}, and is related to dry-bulb temperature where the following relation and unit-less constants, C_i, for each *i*, in Table 1, are also from Fundamentals [9]:

$$\ln p_{ws}(t_{dp}) = \frac{C_1}{T} + C_2 + C_3T + C_4T^2 + C_5T^3 + C_6T^4 + C_7\ln T. \quad (17)$$

Table 1. Constants for the calculation of saturation pressure [9]

C_1	$-1.0214165 \cdot 10^4$
C_2	-4.8932428
C_3	$-5.3765794 \cdot 10^{-3}$
C_4	$1.9202377 \cdot 10^{-7}$
C_5	$3.5575832 \cdot 10^{-10}$
C_6	$-9.0344688 \cdot 10^{-14}$
C_7	4.1635019

III. Determining the Uncertainty of EER

The measurements of each variable in the equations described previously are bucketed into four categories: airflow, power, temperature and humidity, and pressure. Current measurement policy guided device selection for the *baseline* group. To improve on the *baseline* group state-of-the-art, market-available, "secondary standard" measurement devices formed the *proposed* group. The measurement device and its uncertainty for the *proposed* group are discussed in Section V. Measurement details for both groups are shown in Table 2.

Nominal measured values and the measurement uncertainty were used in the equations to determine the uncertainty of EER through accepted uncertainty propagation techniques [10]. Nominal measured values for this analysis were procured from proprietary test data of a unitary air-cooled split-system air conditioner with no indoor fan. The equipment was rated at 13 SEER and approximately 34,000 BTU/hr. The indoor coil was paired with a Carrier condensing unit. The unit underwent the "A-test" of the DOE test procedure at an independent testing laboratory. A summary of the nominal measured values and the measurement uncertainties are shown in Table 3.

The nozzle diameter and the number of nozzles in the test were not provided in the procured data; therefore, an effective nozzle diameter was determined. Because the effective diameter is used by two relations, the definition of flow rate (Eqn. 9) and compressible flow through a nozzle (Eqn. 10), it was determined by simultaneously solving Eqns. 5, 6 I-P, 7, 9, and 10. Airflow data through the indoor airflow measuring apparatus and the differential pressure data across the nozzle were provided in the procured data and used to constrain the system of equations.

Measured nominal power data was provided for the outdoor fan and the compressor; data was not provided for the indoor fan. Since each power was measured separately, each measurement had the same relative uncertainty.

The dry-bulb and wet-bulb temperatures of the air entering and leaving the indoor coil were provided. Wet-bulb temperatures were converted to dew point [11] and then to humidity ratio [9], as only the humidity ratio is used in the equations of the Indoor Air Enthalpy method. Duct heat losses and pre-coil additional heat were not considered, as measured data for them were not provided.

The barometric pressure and pressure drop across the nozzle(s) were provided.

Table 2. Measurement details

	Measurement	Quantity	Measurement Locations	Uncertainty Baseline	Uncertainty Proposed	References	Referenced in Equation(s)
Airflow	Nozzle diameter	8	Nozzle	±0.20%	±0.025%	ANSI/ASHRAE 37-2009 §5.3.3 [6]	(6), (7), (8)
Power	Motor power	2	Outdoor fan Compressor	±2.0%	±0.15%	ANSI/ASHRAE 37-2009 §5.4.2 [6]	(3)
Temp/Humidity	Dry-bulb	2	Indoor pre-coil Indoor post-coil	±0.1°C (±0.2°F)	±0.002°C (±0.0036°F)	ANSI/ASHRAE 41.1-2006 Table 1 [8] ANSI/ASHRAE 37-2009 §5.1.2 [6]	(14), (15)
	Dew point	2		0.2°C (±0.4°F)	(±0.1°C) (±0.18°F)	10 CFR 430 (Appendix M) §2.5.6 [14]	(14), (15)
Pressur	Barometric pressure	1	Test facility	±2.5%	±0.05%	ANSI/ASHRAE 37-2009 §5.2.2 [6]	(12), (13)
	Differential pressure	1	Across the nozzle	±1.0%	±0.05%	ANSI/ASHRAE 37-2009 §5.3.1 [6] ANSI/ASHRAE 116-95 §6.6.6 [15]	(4), (10)

Table 3. Nominal values and measurement uncertainties for the independent variables

	Independent Variable	Nominal Value	Absolute Uncertainty	Relative Uncertainty	Unit
Airflow	D_{AVG}	0.158 (0.52)	1.122E-4 (0.000368)	0.07%	m (ft)
Power	$\dot{E}_{(82)\_ofan}$	314.4	6	2.00%	W
	$\dot{E}_{(82)\_compr}$	2696.5	54	2.00%	
Temp/Humidity	t_{a1}	26.4 (79.6)	0.1 (0.2)	-	°C (°F)
	t_{a2}	13.7 (56.6)	0.1 (0.2)	-	
	$t_{wb\,a1}$	19.6 (67.3)	0.1 (0.2)	-	
	$t_{wb\,a2}$	12.9 (55.3)	0.1 (0.2)	-	
	$t_{dp_{a1}}$	16.1 (61.0)	0.2 (0.4)	-	
	$t_{dp_{a2}}$	12.4 (54.3)	0.2 (0.4)	-	
Pressure	ΔP_V	254 (1.02)	2.49 (0.01)	1.00%	Pa (inH_2O)
	P_{bar}	97.3	2.433	2.50%	kPa
Constants	C	0.99	-	-	-
	R_a	0.52	-	-	$\frac{Pa \cdot m^3}{mole \cdot K}$

Other parameters used in this analysis are constants. The nozzle discharge coefficient was one of the outputs of the equations simultaneously solved during the calculation of the effective nozzle diameter. The air-specific gas constant is 28.7055 Pa-m^3/mole-K [12].

IV. **Results and Discussion**

The capacity and EER were calculated from measured data. The measured data resulted in a capacity of 32,167 BTU/h and an EER of 10.68, while this analysis resulted in a capacity of 31,763 BTU/h and an EER of 10.55. The error in both capacity and EER is less than 1.3%, with the difference possibly due to varying barometric pressure over the duration of the test, cooling losses through the plenum skin that were unaccounted for, or other parametric variations during testing. Table 4 shows the nominal values and the maximum and RSS (root-sum-square) uncertainties of the intermediate and final variables.

To document the numerical results of the uncertainty analysis and connectedness of the variables, a novel visualization is introduced that depicts the flow of measurement uncertainty, as shown in Fig. 1. For each equation, the dependent variable is connected to its independent variables with lines. Their thicknesses qualitatively correspond to the independent variable's contributing uncertainty. Cross-hatched boxes represent the independent variables.

While Eqn. 1 shows that EER is impacted by capacity and power, Fig. 2 shows that more uncertainty is propagated to EER by the capacity variable than the power variable.

Table 4. Nominal values and maximum and RSS uncertainties for the dependant variables

Dependent Variable	**Nominal Value**	**Maximum**		**RSS**		**Unit**
		Absolute Uncertainty	**Relative Uncertainty**	**Absolute Uncertainty**	**Relative Uncertainty**	
$\dot{q}_{tci}$	9.31 (31762.5)	0.94 (3217.8)	10.1%	0.33 (1132.1)	3.6%	kW (BTU/h)
υ_{nsp}	0.8584 (13.7491)	0.0247 (0.3953)	2.9%	0.0217 (0.3472)	2.5%	m^3/kg (ft^3/lbm_{da})
υ_n'	0.8504 (13.6218)	0.0246 (0.3935)	2.9%	0.0215 (0.3439)	2.5%	m^3/kg (ft^3/lbm_{total})
$h_{a1} - h_{a2}$	19.615 (8.4328)	1.014 (0.4360)	5.2%	0.414 (0.1782)	2.1%	kJ/kg (BTU/lbm_{da})
$\dot{E}_{(82)}$	3010.9	60	2.0%	54	1.8%	W
W_2	0.00934	1.376E-04	1.47%	1.376E-04	1.47%	lbs/lbs
W_1	0.01192	1.712E-04	1.44%	1.712E-04	1.44%	lbs/lbs
$\dot{Q}_{mi}$	0.407 (863.1)	0.009 (18.0)	2.1%	0.006 (11.8)	1.4%	m^3/s (cfm)
h_{a2}	55.294 (23.7721)	0.462 (0.1987)	0.8%	0.366 (0.1574)	0.7%	kJ/kg (BTU/lbm_{da})
h_{a1}	74.909 (32.2049)	0.552 (0.2373)	0.7%	0.194 (0.0835)	0.3%	kJ/kg (BTU/lbm_{da})
A_n	0.49538 (0.212977)	0.00070 (0.000301)	0.1%	0.00070 (0.000301)	0.1%	m^2 ($ft^{2)}$
EER	**10.55**	**1.28**	**12.1%**	**0.42**	**4.0%**	**BTU/W-h**

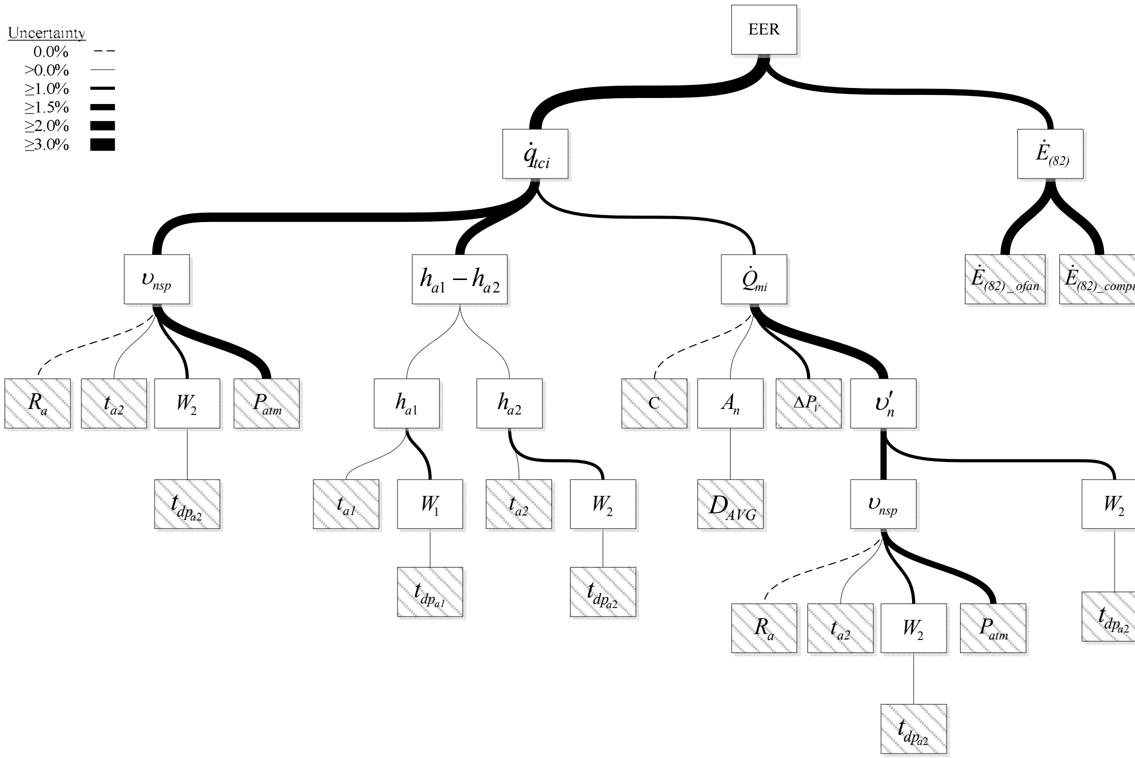

Figure 1. The flow of measurement uncertainty, depicting the contributing uncertainty of various independent variables

Capacity is governed by specific volume, enthalpy change, and air flow rate. Uncertainty is larger in the specific volume and enthalpy change variables, however, it is relatively similar among these variables. Examining the uncertainty contributions of each variable, the uncertainty of the specific volume is more greatly affected by the uncertainty of the pressure measurement than the other three variables that govern specific volume. This is because the other variables—the gas constant, exiting air dry-bulb temperature, and exiting air humidity ratio—have an uncertainty that is low; indeed, their combined uncertainties would not round the tenth percent digit of the uncertainty of the specific volume. Indeed, the 2.5% uncertainty of the specific volume, υ'_n in Table 4, is not affected by the relative uncertainty contributed by the gas constant (0.0%), exiting air dry-bulb temperature (0.35%), and exiting air humidity ratio (0.67%) and is, in fact, virtually the same as the 2.5% uncertainty of the differential pressure measurement, ΔP_V, in Table 3.

Low relative uncertainty may have a significant impact on the dependent variable when summed or subtracted, as is the case with the enthalpy difference. Although the uncertainty of the enthalpy of the air entering and exiting the indoor coil are represented in Fig. 2 with thin lines and are shown in Table 4 with RSS relative uncertainties of 0.3% and 0.7%, their combined impact on its dependent variable, enthalpy difference, is large. This is also reflected in work by Cherem-Pereira and Mendes [13]. Equation 2 shows that the entering and exiting enthalpies are subtracted. Because the differences of the nominal values create a small nominal value, and add/subtract operations on uncertainties sum their absolute uncertainties, the ratio of a larger absolute uncertainty to a smaller nominal value creates an uncertainty that can be unexpectedly high.

Not all instances of combining variables lead to higher uncertainty. For example, while airflow rate is heavily influenced by differential pressure and more so by specific volume, airflow rate only propagates a 1.4% uncertainty to EER due to the square root function. In another case, while exiting air humidity ratio has a 1.5% uncertainty and the specific volume of dry air at standard pressure has a 2.5% uncertainty, the uncertainty of the specific volume of air at the nozzle in units per air-water vapor mixture still remains at 2.5%. This is caused by the inclusion of one to the denominator of Eqn. 11, and any uncertainty that would have been propagated by the exiting air humidity ratio would be marginalized. A similar situation occurs with the total power ($\dot{E}_{(82)}$). It is governed by two parameters that each holds 2.0% relative uncertainty. When the power of the outdoor fan and compressor is summed, the nominal values and absolute uncertainties are summed. The ratio of absolute uncertainty to nominal measured power remains the same through each addition, causing the maximum uncertainty to remain at 2.0%. However, for RSS uncertainty, the power for these two components is applied in an RSS operation during the summation of their absolute uncertainties; therefore, the RSS relative uncertainty is actually lower, 1.6%.

The presentation of EER to consumers is concerning. Current products in certified product performance databases provide EERs for central air conditioners to two digits after the decimal. This suggests that EERs are accurate to that second decimal place, meaning that the stated accuracy deviates no more than ±0.004 EER. In this analysis, EER is reported with an uncertainty of ±0.42, or in the worst case, ±1.28. It seems that certified values may overestimate the accuracy of the EER by two to three orders of magnitude. Although certifiers may not likely use metrology instruments that date to the 1980s, the lack of transparency in test reports and equipment databases, and the unchanged measurement tolerances in published Standards, could give rise to doubt and skepticism in the institutions that provide information to the consumer.

V. **Conclusions**

The accuracy of the EER for central air conditions is important to study as the metric is commonly used by energy policy makers and relied on by consumers. However, obsolete measurement devices specified by current measurement policies may be inaccurate yet have been used for decades. To pave the way for improvements to measurement policy that directly impacts the accurate determination of energy efficiency ratio (EER), the uncertainty with those measurement devices needs investigation. The assessment of current measurement policies on the uncertainty of EER is summarized in these conclusions:

- In an uncertainty analysis, the impact of the contributing uncertainty of various independent variables can be qualitative visualized by using a novel visualization technique that depicts the flow of measurement uncertainty.

- Uncertainty analysis based on current measurement and test procedure policy reveals a range of the uncertainty in the EER of 4.0% (root-sum-square uncertainty) up to 12.1% (maximum uncertainty).

- Based on equipment tested under current measurement and test procedure policy, certified EER values may be overestimated.

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References

[1] Energy Information Administration. (2012). Air conditioning in nearly 100 million U.S. homes, released August 19, 2011; retrieved November 16, 2012 from http://www.eia.gov/consumption/residential/reports/2009/air-conditioning.cfm.

[2] Energy Information Administration. (2012). Annual Energy Outlook 2012. "Summed total energy consumption by space cooling end use for 2010 residential", pg. 140.

[3] Shen, B. (2006). Improvement and Validation of Unitary Air Conditioner and Heat Pump Simulation Models at Off-Design Conditions. Ph.D. thesis.

[4] Mullen, C.E., Bullard, C.W. (1994). Room Air Conditioner System Modeling, Air Conditioning and Refrigeration Center at the University of Illinois at Urbana-Champaign, Technical Report 60.

[5] Diaz, N.F. (2009). Methodology for uncertainty calculation of net total cooling effect estimation for rating room air conditioners and packaged terminal air conditioners. International Journal of Refrigeration, Volume 32, pg. 1472–1477.

[6] ASHRAE (2009). ANSI/ASHRAE Standard 37-2009, Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment, ASHRAE, 1791 Tullie Circle N.E., Atlanta, GA, 30329.

[7] ASHRAE (2007). ANSI/ASHRAE Standard 51-2007, Laboratory Methods of Testing Fans for Aerodynamic Performance Rating, American National Standards Institute, 1899 L Street NW, 11th Floor, Washington, DC, 20036.

[8] ASHRAE (2006). ANSI/ASHRAE Standard 41.1-1986 (RA 2006), Standard Method for Temperature

Measurement, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA, 30329.

[9] ASHRAE (2009). ASHRAE Handbook - Fundamentals. ASHRAE, 1791 Tullie Circle N.E., Atlanta, GA, 30329.

[10] ASHRAE (2010). ASHRAE Guideline 2-2010, Engineering Analysis of Experimental Data, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA, 30329.

[11] Snyder, R.L., Shaw, R.H. (1984). Converting Humidity Expressions with Computers and Calculators. Cooperative Extension Leaflet 21372. University of California – Davis.

[12] ASHRAE (2006). ANSI/ASHRAE Standard 41.6-1994 (RA 2006), Standard Method for Measurement of Moist Air Properties, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA, 30329.

[13] Cherem-Pereira, G., Mendes, N. (2012). Empirical modeling of room air conditioners for building energy analysis. Energy and Buildings, Volume 47, April 2012, pgs. 19–26.

[14] Title 10, Code of Federal Regulations (CFR), Part 430, Appendix M of Subpart B, U.S. National Archives and Records Administration, 8601 Adelphi Road, College Park, MD, 20740-6001.

[15] ASHRAE (1995). ANSI/ASHRAE Standard 116-95, Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA, 30329.