High Efficiency Novel Window Air Conditioner

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ABSTRACT

This paper presents the technical development of a high efficiency window air conditioner. In order to achieve higher energy efficiency ratio (EER), the original capacity of the R410A unit was reduced by replacing the original compressor with a lower capacity but higher EER compressor, while all heat exchangers and the chassis from the original unit were retained. The other subsequent major modifications included - the AC fan motor being replaced with a brushless high efficiency electronically commutated motor (ECM) motor, the capillary tube being replaced with a needle valve to better control the refrigerant flow and refrigerant set points, and R410A being replaced with 'drop-in' environmentally friendly binary mixture of R32/R125 (85/15% molar concentration). All these modifications resulted in 21.3% enhancement in the EER of the window air conditioner.

Key words: Alternative refrigerants, window air conditioner, energy efficiency, ECM, GWP

Nomenclature

E: Energy	EER: Energy Efficiency Ratio	h: Enthalpy (Btu/lb)	$\dot{m}$ = Flow Rate (lb/h)
P: Pressure (psia)	q: Capacity (Btu/h)	T: Temperature (°F)	x: quality
Subscripts:	c: condenser	e: evaporator	f: liquid
g: vapor	SH: Superheat	SC: Sub-cooling	sat: Saturation
com: Compressor	t: Total		

1. INTRODUCTION

Window air conditioners (WAC) are inexpensive and sold in large numbers internationally as alternatives to central air-conditioning (AC) systems for space cooling, supplemental cooling and for adding AC to existing homes or extension of homes that do not have any AC to improve comfort. They are particularly attractive in older buildings that lack ducted systems and in cases where a central system upgrade is first-cost prohibitive [Bansal (2015), Bansal and Shen (2015), Nogueira (2013), Winkler et al. (2013)]. According to the US Energy Information Administration (EIA) there were nearly 46.7 million WACs operating within the United States in 2009 [EIA (2009)], accounting for approximately 1.5% of the total US residential energy use or about 0.33 quads (0.35 EJ).

WACs are available in sizes typically between 5,000 and 16,000 Btu/h (0.4-1.3 tons, or 1460–4700 W of cooling). A WAC is generally installed either in a window or in a framed wall opening depending on the design of the condenser side vents. When the project was initiated in 2011, the minimum required energy efficiency ratio (EER) levels for WACs were set at 9.7 Btu/Wh or 2.84 W/W (for capacity less than 10,000 Btu/h or 2.93 kW) to 9.8 Btu/Wh or 2.87 W/W (for 10,000 Btu/h or higher capacity) for most product classes. These levels have since been subsequently raised to 10.7 and 10.8 Btu/Wh (3.14 and 3.17 W/W) respectively in 2014 [e-CFR (2014)].

Although vapor compression refrigeration is a highly researched topic, surprisingly very little research has been devoted to the simulation and energy efficiency improvement of WACs in the open literature [Goswami et al (1993), Hajidavalloo (2007), Naphon (2010), Sawant et al (2011), Chua et al. (2013), Meissner et al. (2014)], while a few studies are devoted to investigating alternative refrigerants for WACs [Devotta et al. (2005), Jabaraj et al. (2006), Bolaji (2011), Zhou and Zhang (2010), Shen and Bansal (2014)]. One efficiency enhancement option investigated in the literature [Goswami et al. (1993), Hajidavalloo (2007)] included evaporative wet cooling media pads on the condenser of window-air-conditioner that was reported to reduce the power consumption of the WAC. However, this system had practical limitations of clogging up of media pads and maintaining sufficient moisture in the pads to sustain the evaporative cooling effect. Sawant et al (2011) sprayed dehumidified water on to the media pads placed on both sides of the air conditioner that led to energy savings of the WAC. They also studied an alternative approach of using wicks of porous cloth where one end of the wick was immersed in dehumidified water and the other end was wrapped on the air flow entering side of the condenser. Naphon (2010) found that the use of a 3-rows heat pipe with refrigerant R134a as the working fluid to cool the air entering the condenser of a window air conditioner led to an EER improvement.

There are several traditional design approaches that are generally applied to improve the energy efficiency of central HVAC systems, such as variable-speed compressors and advanced control systems. The benefits of these approaches are not captured well for WACs since the performance rating testing of WACs is limited to a single test point. It may, however, be noted that the novelty of the ‘single point’ test is that it captures the energy consumption of the unit during peak load, which is a major consideration when conserving energy for the US. Modern

WACs take advantage of slinger rings mounted on their condenser fans to spray condensate collected from the evaporator during operation onto the condenser (the 'sling' effect) to provide some evaporative cooling of the condenser. However, other design improvements for efficiency gains in the heat exchangers, compressor, and fan motors have not been widely adopted. In addition, because of the low first cost of window air conditioners, typically in the $100 to $500 range, any added costs for design improvements must be limited so that they are not cost-prohibitive for this market.

This project to develop a high efficiency window air conditioner was initiated by the Building Technology Program (BTO) of the Department of Energy (USA), whose goal is to reduce the energy and carbon emissions used by the energy service equipment (equipment providing space heating and cooling, water heating, etc.) by 50% compared to 2010 baseline best common practice by 2030. The average lifetime of WACs is about 10 to 13 years. This fact coupled with the potential for rapid replacement of older, less efficient units, some of which have EERs as low as 5 Btu/Wh (1.47 W/W) means they have potential to significantly reduce the annual U.S. energy consumption [Bansal (2015)]. This project endorses the BTO's long term goal to create marketable technologies and design approaches that address energy consumption in existing and new buildings. Therefore, improving the efficiency of window units is essential to enhancing overall residential building energy performance. Such an initiative would not only bring energy savings to the end user but would also assist in protecting the environment as a result of reduced greenhouse gas emissions worldwide.

As per the literature review, this is perhaps the first study of its kind to design and develop a high efficiency WAC with novel features. This paper, therefore, discusses an experimental evaluation of a number of system design options applied to improve the energy efficiency of a WAC with base capacity of 10,000 BTU/h (or 2.93 kW) and charged with R410A (mixture of R32/R125 with 70/30% molar concentration). These included- i) ECM (electronically commutated motor) fan motor, ii) higher rated efficiency and lower rated capacity (8,000 Btu/h, or 2.3 kW) compressor and iii) a new 'drop-in' refrigerant mixture of R32 (with 85% molar concentration) and R125 (15% molar concentration). This R32/R125 blend has a 34% lower global warming potential (GWP) than R410A (~1,377 vs. 2,079). Due to the compact size configuration and small refrigerant charge in a typical WAC (less than 1 kg), the slight flammability of the 85/15 R32/R125 refrigerant mixture is less of a concern. This is an important feature in view of the environmental friendliness of future WACs. This study documents the design details of the baseline WAC, specifically the condenser fan slinger ring, and the added features in the performance improvement of the WAC. Data analysis to determine efficiency was performed primarily using energy balance on the refrigerant side. The paper presents the system configurations, experimental approach and methodology to develop the high efficiency WAC, and the results obtained. The project has been carried out for the larger benefit of the global community.

2. SYSTEM CONFIGURATION AND EXPERIMENTAL TEST FACILITY

A WAC unit using R410A refrigerant and having a nominal rated capacity of 10,000 Btu/h (2.93 kW) was selected as the baseline (or stock) platform for performance evaluation and efficiency enhancement. The baseline WAC had a single-speed rotary compressor, a fin-&-tube evaporator and condenser, a capillary tube refrigerant flow control device and a single electric motor mounted on a single axis shaft to drive both the evaporator blower and the condenser fan. In addition to these basic components, the WAC had a fin-&-tube sub-cooler submerged in a water collection pan mounted beneath the evaporator (see Fig.1). The submerged sub-cooler provides further sub-cooling (up to 10°F) to the liquid refrigerant exiting the condenser. The schematic diagram of the WAC and its P-h diagram are shown in Figures 1 and 2 respectively. As noted earlier the condenser fan blade is specially configured with a “slinger” ring designed to pick up water from the water collection pan and to spray it in the air stream flowing over the condenser coil surface. The water droplets evaporate and enhance the condenser heat transfer. This feature is called the “sling” effect [LBNL (1997)]. Figures 3, 4 and 5 respectively show the single axis fan, the “slinger” and the instrumented WAC. To promote mixing of air at the condenser outlet, a 24” section of insulated duct (shown in Figure 5) was added upstream of the condenser to allow accurate measurement of exiting air temperature and relative humidity (RH). The unit was fully instrumented and mounted between the indoor and outdoor test rooms of an environmental chamber, as shown in Figure 6.

Analysis: As noted earlier, the WAC system efficiency and the evaporator and condenser heat transfer analysis was performed on the refrigerant side. With reference to Figures 1 and 2, the condenser and evaporator heat transfer rates are respectively given by;

$$q_c = \dot{m}[h(T_2, P_2) - h(T_3, P_3)] \quad (1)$$

$$q_e = \dot{m}[h_1(T_1, P_1) - h(P_5, X_5)] \quad (2)$$

where T and P are the measured temperature and pressure at various refrigerant state points, h is the interpolated specific enthalpy of the refrigerant and ṁ is the measured refrigerant mass flow rate. It is worthwhile to note here that the expansion from state 4 to 5 is assumed isenthalpic, or;

$$h_5 = h_4(T_4, P_4) \quad (3)$$

With the quality at state 5 calculated as;

$$x_5 = \frac{h_5 - h_f(P_5)}{h_g(P_5) - h_f(P_5)} \quad (4)$$

where h_f and h_g are the interpolated specific enthalpies for the liquid and gas phases respectively.

The evaporator superheat and condenser sub-cooling are given by;

$$T_{SH} = T_1 - T_{sat}(P_1) \quad (5)$$

$$T_{SC} = T_{sat}(P_3) - T_3 \quad (6)$$

where T_{sat} is the interpolated saturation temperature at pressure P.

The work input to the refrigerant by the compressor is-

$$w_{com} = \dot{m}[h_2(T_2, P_2) - h(T_1, P_1)] \quad (7)$$

The overall efficiency metric, Energy Efficiency Ratio (EER), of the unit is computed by;

$$EER = \left\{\frac{Refrigeration\ capacity, q_e}{Total\ power\ input,\ \ E_t}\right\} \quad (8)$$

where q_e is the system capacity in Btu/h and E_t is the total electrical power usage (in watts) including both the measured compressor power and the fan/blower power.

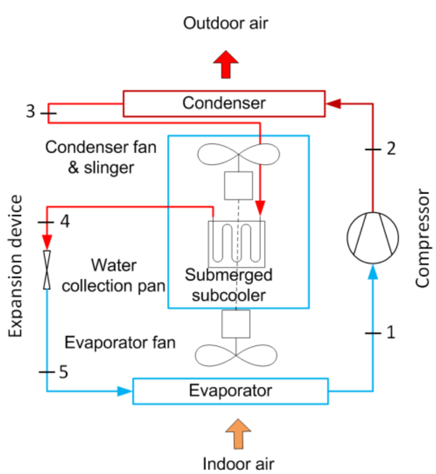

Figure 1: Schematic of Window Air Conditioner

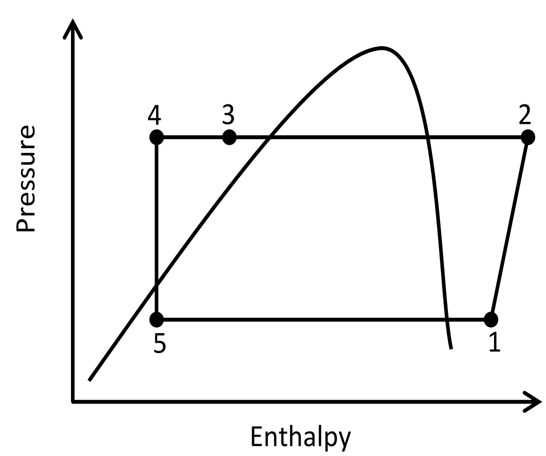

Figure 2: P-h diagram of WAC

Figure 3: Single axis blower/fan

Figure 4: Slinger ring on condenser fan

Figure 5: Instrumented WAC

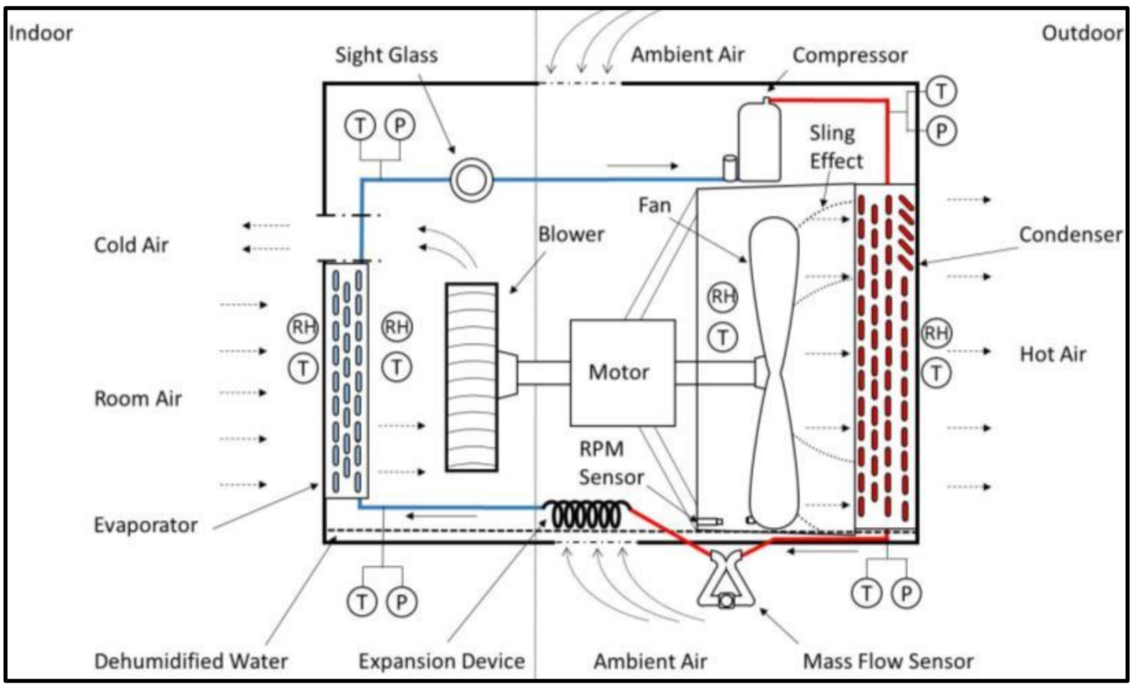

Figure 6: Schematics of the window air conditioner with instrumentation details

3. INSTRUMENTATION AND DATA ACQUISITION

In order to quantify the performance of the unit, a measurement/DAS system was developed using a Campbell Scientific CR3000 data logger. The setup consists of the logger, as well as an AM16/32B multiplexer and PS100 power supply mounted in a Campbell Scientific enclosure. The system included integrated refrigerant lookup tables as well as a graphical user interface (GUI) on the control PC to provide real-time feedback during testing. In addition, a MathCAD based analysis tool was developed to perform a zero order uncertainty analysis in both the base measurements and the corresponding calculated quantities. The program tracks the propagation of uncertainties in the base measurements through to the final calculated quantities. (Capacity, COP, EER etc.). This unit was fully instrumented to log 51 measurements including refrigerant mass flow rate using a Coriolis mass flow sensor, 3 in-line refrigerant temperatures (at suction, discharge and liquid line), 4 refrigerant side pressures, 11 refrigerant side and tube wall temperatures, 24 air-side temperatures, 4 air side %RH (relative humidity) measurements, 3 power measurements including fan, compressor, and total power; and barometric pressure. Figure 7 shows the GUI developed for this project, displaying pertinent measured and calculated quantities updated every 10s.

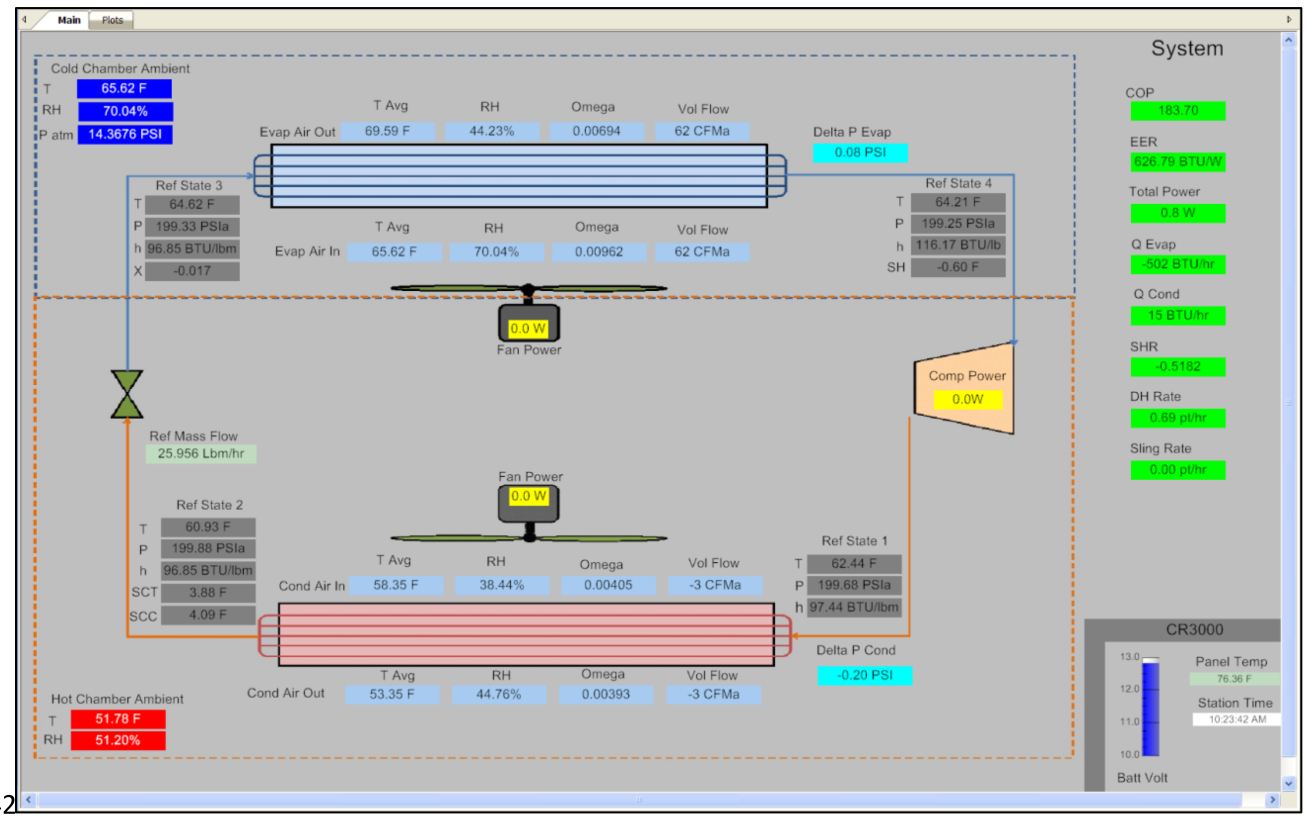

Figure 7: Screenshot of DAS system GUI

Experimental uncertainty: To quantify the level of uncertainty in both the base measurements and the calculated quantities, a zero order analysis was performed. The study establishes overall uncertainties for various sensors at the standard test conditions. The analysis then tracks how these base uncertainties propagate through to the final calculated quantities (EER, COP, etc.). Some of the assumptions used in the analysis are given below-

(1) Uncertainties due to transient conditions are not considered since the tests focus on steady state analysis.

(2) Uncertainties introduced from linear interpolation of the refrigerant and saturated water vapor tables are considered negligible. Analysis of the data shows the linear approximation to be quite good over the small temperature/pressure increments used for the interpolation subroutines.

(3) The accuracy of any calculated value will be a function of both the ability to accurately measure the parameters used to calculate it as well as how accurately the equation models the actual physics of the process. The values presented in this study do not account for the latter.

(4) The results of the analysis presented here are for a single instantaneous measurement. Since many of the final results are based upon the average of N multiple measurements, the uncertainty in the average will decrease approximately by the factor of $N^{-1/2}$, assuming that the sensor error is random in nature. Because of this, even in cases where the uncertainty for a single measurement is high (as is the case for the calculated condenser airflow), the average of 30 minutes of data is still a fair estimate.

The results of the design point uncertainty analysis are reported in Table A1 of the Appendix.

4. SALIENT EXPERIMENTAL DETAILS OF THE WINDOW UNIT

In order to establish the baseline performance, the instrumented unit was first mounted in a plywood partition between the indoor and outdoor test chambers. The temperature and relative humidity of the indoor chamber were adjusted based upon the measurements at the evaporator air inlet. The outdoor chamber was adjusted based upon a single temperature/RH probe mounted a few feet from the intake louvers on the top of the unit. The outdoor and the indoor conditions inside the chamber were maintained at [95ºF (35ºC), 40.1%RH] and [80ºF (26.7ºC), 51.5%RH] respectively.

Determination of Optimum Charge: Optimal refrigerant charge of R410A for the base unit was re-determined because of the increased volume of the system after the sensors were installed (see Figures 5 and 6). The system was initially filled with the factory charge and the charge was then added in small increments to achieve the maximum EER at a superheat of around 4.4 K (=8^{0}F) at the standard rating conditions. As can be seen from Figure 8 that initially, the level of superheat (left axis) and sub-cooling (right axis) would respectively decrease and increase substantially with the incremental charge but superheat eventually becomes asymptotic with subsequent additions at refrigerant charge above the 790 gm level. The corresponding normalized EER and the system capacity are plotted in Figure 9, where the charge level of 790 gm, prior to the asymptotic behaviour, was considered optimum because the EER did not increase appreciably for greater charge levels.

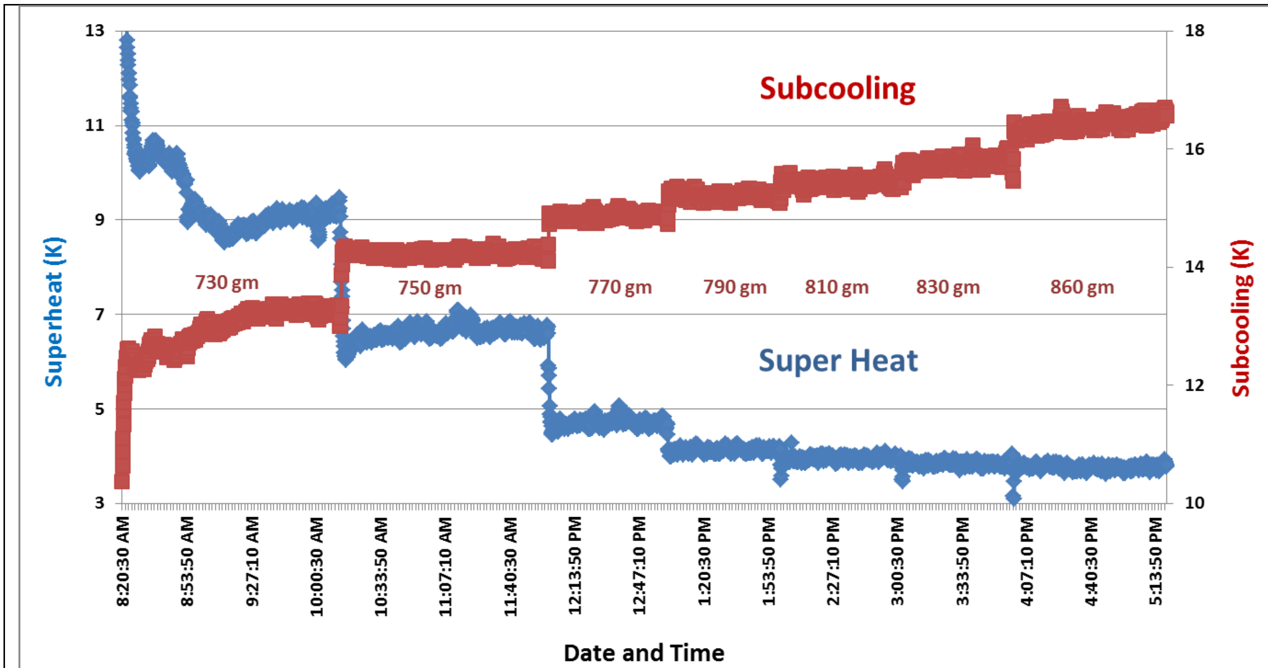

Figure 8: Variation of superheat and sub-cooling with refrigerant charge

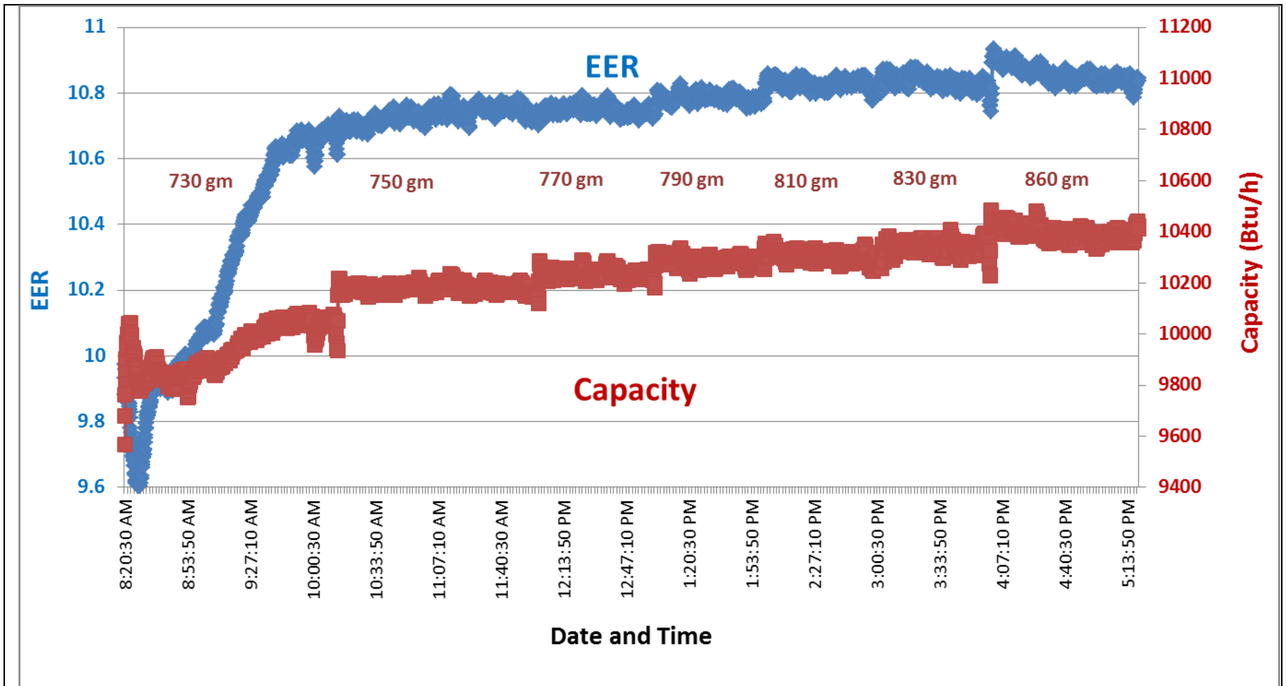

Figure 9: Variation EER and capacity (1 Btu/h = 0.293 W) with refrigerant charge

With the unit installed in the chamber and charged, a test procedure was developed to generate the data as follows. With the unit running, bring both chambers to near the desired set points.

(1) Fine tune the chamber set points to achieve the desired inlet conditions as measured at the evaporator inlet sensors for the indoor chamber and the single temp/RH probe near the intake louvers on the outdoor chamber.

(2) When steady temperature/RH readings are achieved, allow at least ½ hour for the window unit to achieve equilibrium. This allows time for the condensate level in the pan to reach a steady value. The capacity and total electrical power measurements should be steady within ±2% of the mean value before testing begins. Allowing ½ hour after any changes was found to achieve this condition.

(3) Once steady state is achieved, data is collected for an additional ½ hour period.

Performance comparison with the rated baseline case: The WAC was tested in the environmental chamber as per the standard rating conditions noted above [DOE (2011), e-CFR (2014)]. In general, the experimental results were found to correlate well with the published numbers, with differences of -3% and +2.6% for the experimental values of EER and capacity respectively, as shown in Table 1.

Table 1: Comparison of experimental values to rated performance

Outdoor= 95ºF, 40.1% RH	Indoor= 80ºF, 51.5% RH			
	EER (Btu/Wh)	**Capacity (Btu/h)**	**Evaporator Airflow (CFM)**	**Condenser Airflow (CFM)**
Rated	10.8	10,000	300	500
Experimental	10.48	10,261	296	497
Difference	-3.0%	+2.6%	-1.4%	-0.6%

Air flow rate in cubic feet per minute (1 CFM = 0.472 l/s); 1 BTU/h = 0.293 W;

Performance enhancement with "slinger" effect: In order to assess the effect of "slinger" on the performance of the window air-conditioner (WAC), a hole was drilled at the bottom of the WAC in the condenser pan that allowed the condensed water to be collected in a bucket and eliminate any entrained water from blowing over the condenser. The system was run at the standard test conditions and was allowed to reach steady state. The system EER was found to decrease by approximately 8% compared to the baseline data. No noticeable change was observed in the fan power. The condensate production rate, collected in the bucket, was measured to be about 1 liter/h.

5. FEATURES TO ENHANCE EFFICIENCY OF A WINDOW AIR CONDITIONER

In order to develop a higher efficiency WAC, several primary design strategies were considered in order to improve the WAC efficiency, including the following-

(1) Use of micro-channel heat exchangers (MCHXs)
(2) Reducing thermal bridging and internal air leakage through the divider wall
(3) Use of high EER compressor with lower rated capacity
(4) Use of an ECM (or brushless DC) fan motor in place of AC induction fan motor
(5) Reduce unit capacity from 10,000 Btu/h to 8000 Btu/h by downsizing its compressor
(6) Increase the molar concentration of R32 (from 70% to 85%) in R410A

MCHX: MCHXs are known to offer a higher volumetric heat capacity and a reduced refrigerant charge that may result in higher efficiency, particularly, in large capacity systems. This was the case when Yun et al. (2006) replaced the fin-and-tube condenser of a 7 kW cooling capacity residential air conditioner with a MC condenser. The COP of the system increased by up to 10%.

However, when Kim and Bullard (2002) replaced the fin-and-tube condenser of a room air conditioner (of 4.1 kW cooling capacity as opposed to 2.93 kW of the current system), both the capacity and EER of the new WAC were lower than the baseline system. Since the current designs of micro-channel heat exchangers (MCHXs) use lower fin density (e.g. 16 fins per inch on evaporator) while the chassis of the WAC has fixed dimensions and the capacity of the WAC is quite low (=2.93 kW), it is difficult to match the face area of the round tube plate fin coils with that of the micro-channel. Although MCHXs offer better performance in most HVAC&R applications, they are not as deep as the copper tube heat exchangers and therefore, offer inferior performance (i.e. higher pressure drops, lower refrigeration capacity and less EER) than the round tube plate fin coils heat exchanger in a WAC. Therefore, this feature was found to be impractical to improve the efficiency of the current lower capacity WAC.

Reducing thermal bridging and internal air leakage: Although some of the low efficiency units may suffer from thermal bridging and internal air leakage through the divider wall, the unit under investigation was well engineered and had hardly any room for improving either the thermal bridging or the internal air leakage through the divider wall.

Testing the WAC with other high efficiency novel features: In order to assess the impact of the remaining efficiency improvement measures listed above, these were embedded into the WAC one-by-one to assess their individual performance. These were then integrated together into the unit to assess their overall impact on the performance of the WAC.

ECM Fan Motor- The first modification was the replacement of the stock (i.e. baseline) AC induction motor with a permanent magnet ECM. The factory AC motor was a permanent split capacitor type with a rated output of 95W. During the baseline tests, the motor was found to consume 177W, 158W and 137W at the high (1343 RPM), medium (1256 RPM), and low (1150 RPM) fan speeds respectively. The replacement motor was a sensor-less ECM with the rated 100W mechanical output. The system was charged to the same level as the baseline case and the test chambers were run at the standard testing conditions noted previously. The ECM fan motor was then run at the same RPM as the high and low settings of the baseline fan. Data was collected at 5s intervals for 30 minutes at each speed and 30 minutes was allocated for the system to stabilize after the RPM was changed. Figure 10 shows a significant increase of more than 8% in system EER (= 11.7 compared with 10.8 of baseline) through this change alone. The use of higher efficiency ECM brushless fan motor offers essentially twofold benefits. Not only is less energy used to produce the same airflow, less heat is being dissipated in the chassis resulting in lower temperature of the condenser inlet air.

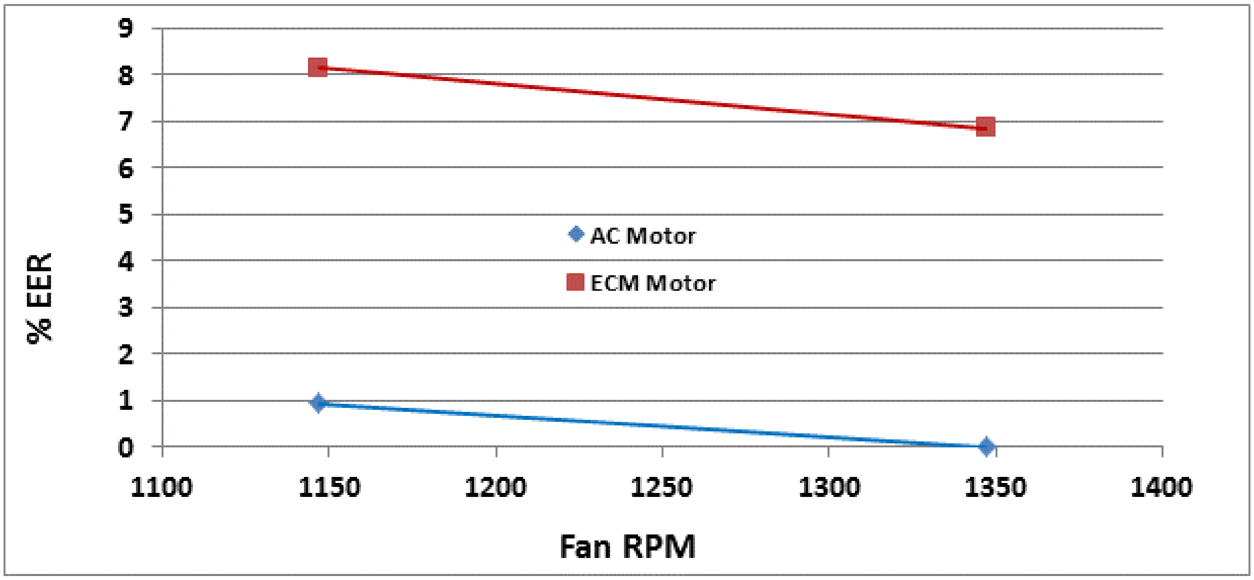

Figure 10: Variation of EER with Fan Speed for Stock System and System with ECM Fan

System Capacity Reduction- In an attempt to further increase system efficiency, the stock 10,000 Btu/h compressor was replaced with one rated at 8,000 Btu/h. The stock heat exchangers were retained and because they are essentially oversized for the derated unit, the system can be run at a lower pressure ratio than the original baseline WAC. To optimally control this, the capillary tube on the baseline unit was replaced with a fine adjust needle valve. A sight glass was also added at the compressor suction of the system to avoid liquid slugging of the compressor by inadvertently opening the valve too far. Because the system volume is changed by the integration of the smaller compressor, the charge optimization study was re-performed, while maintaining the superheat closer to 4.4 K (8°F). Figure 11 shows the variation of modified system EER and its capacity with charge, where a maximum EER improvement of 6.2% (at a capacity of 8626 Btu/h) was achieved at a charge level of 680 gm over the baseline case.

System Capacity Reduction and ECM Fan Motor- With the incremental improvement from the smaller compressor quantified, the same reduced capacity system was tested with the ECM fan motor. The unit was charged with 680 gm and the airflow was varied between 1050 and 1400

RPM, again with time allocated for the unit to stabilize between changes. The needle valve was adjusted at each airflow rate to maintain suction superheat at about 4.4 K (=8.0°F). Figure 12 shows the variation of system EER and fan power input with fan RPM. The test data is fit with a 2nd order polynomial which predicts a maximum EER improvement of 18.2% over the baseline case (i.e. EER enhancement from 10.8 to 12.77) at a fan speed of about 1210 RPM. The corresponding unit capacity and fan power input were respectively 8,310 Btu/h and 79 W.

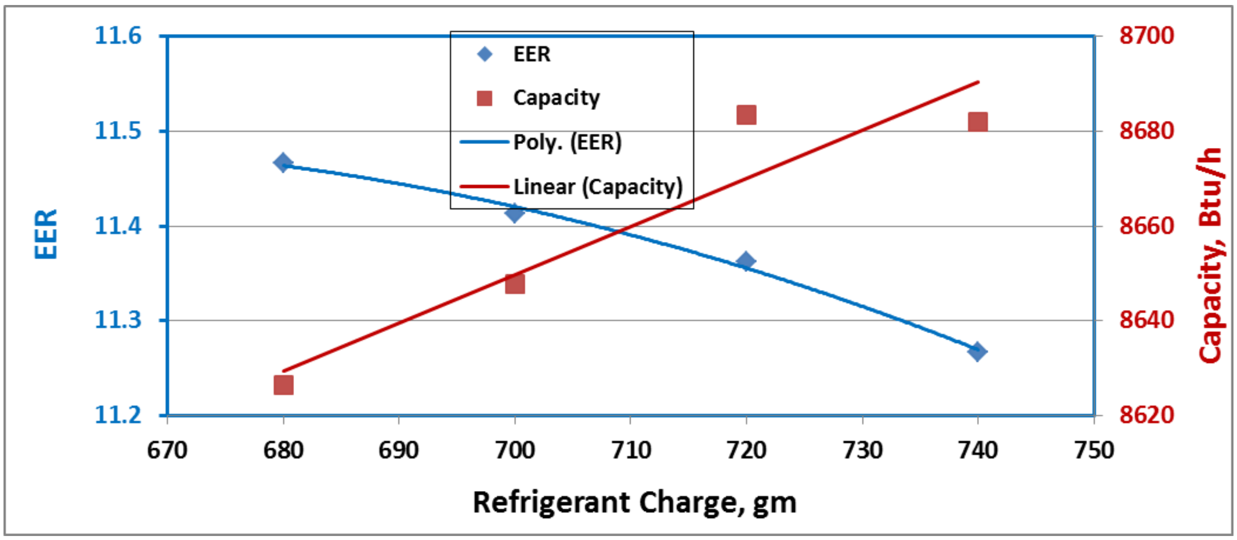

Figure 11: Variation of EER with Charge for the reduced system capacity

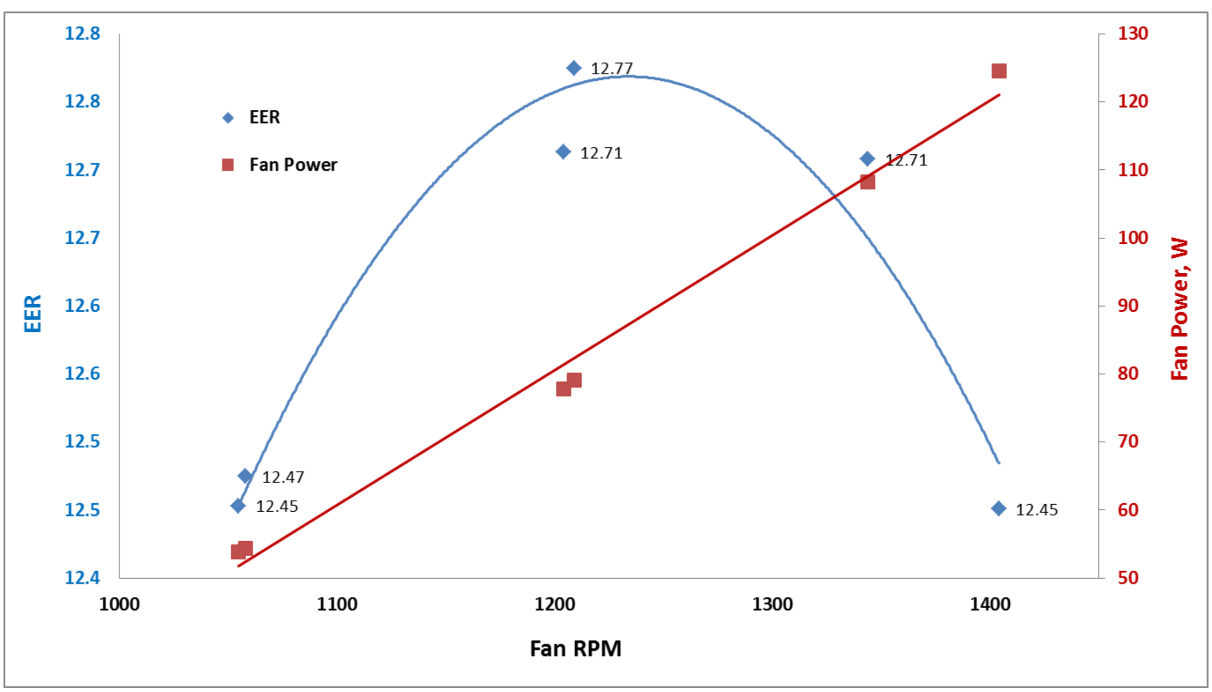

Figure 12: Variation of EER and fan power input for the reduced capacity system with RPM of ECM Fan

System Capacity Reduction, ECM Fan Motor and 85% molar concentration of R32- The refrigerant R410A is a 70/30% (by mole) mixture of R32 and R125. An increase in the concentration of R32 leads to increased system capacity for a given mass flow rate at the expense of the mixture becoming slightly flammable. To explore the effects of the higher R32 concentration, the unit was evacuated and charged with an 85/15 molar mix of the two refrigerants. Due to time constraints, this test was run only with the 8,000 Btu/h compressor and ECM fan instead of starting with the stock compressor and AC fan motor. As before, a charge study was conducted with the optimum charge being found to be 620 gm. Although tests were run at two airflow rates with varying degrees of suction superheat, results are shown only for fan RPM of 1154 (due to brevity of space) in Figure 13. As shown, a maximum EER improvement of 21.3% (i.e. EER enhancement from baseline 10.8 to 13.1) was measured over the baseline unit at a superheat of around 4.8K (8.6°F). The corresponding capacity and the compressor discharge temperature were respectively 8,428 Btu/h and 51.9°C. It may be noted here that the custom blend improved the EER over R410A case by about 3%, which is in agreement with the modelling results of Bansal and Shen (2015).

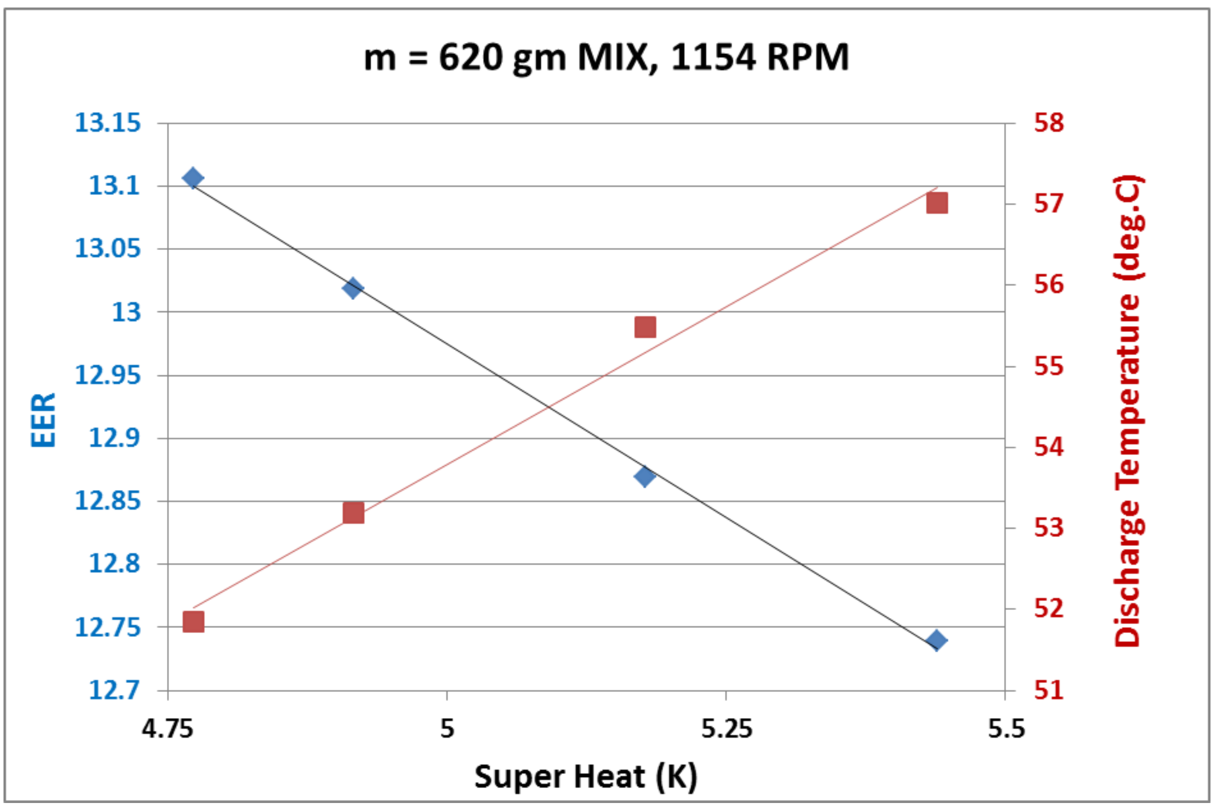

Figure 13: Variation of Efficiency Enhancement vs. Superheat, with ECM Fan (at 1154 RPM) and R32/R125 (85/15) Mixture

Overall Summary and Discussion- Table 2 summarizes the tests conducted and the percent improvement of each over the baseline performance. Figure 14 displays the incremental EER enhancements and the corresponding system capacity for each option from stock (base unit) to the mixture of R32 (85% by mole) and R125 (15% by mole).

Table 2: Summary of Test Results

Tests	ECM Fan	Reduced Capacity	Increase of R32 molar concentration to 85%	% EER Increase over stock unit
Baseline/Stock	no	no	no	0%
Test 1	yes	no	no	8.1%
Test 2	no	yes	no	6.2%
Test 3	yes	yes	no	18.2%
Test 4	yes	yes	yes	21.3%

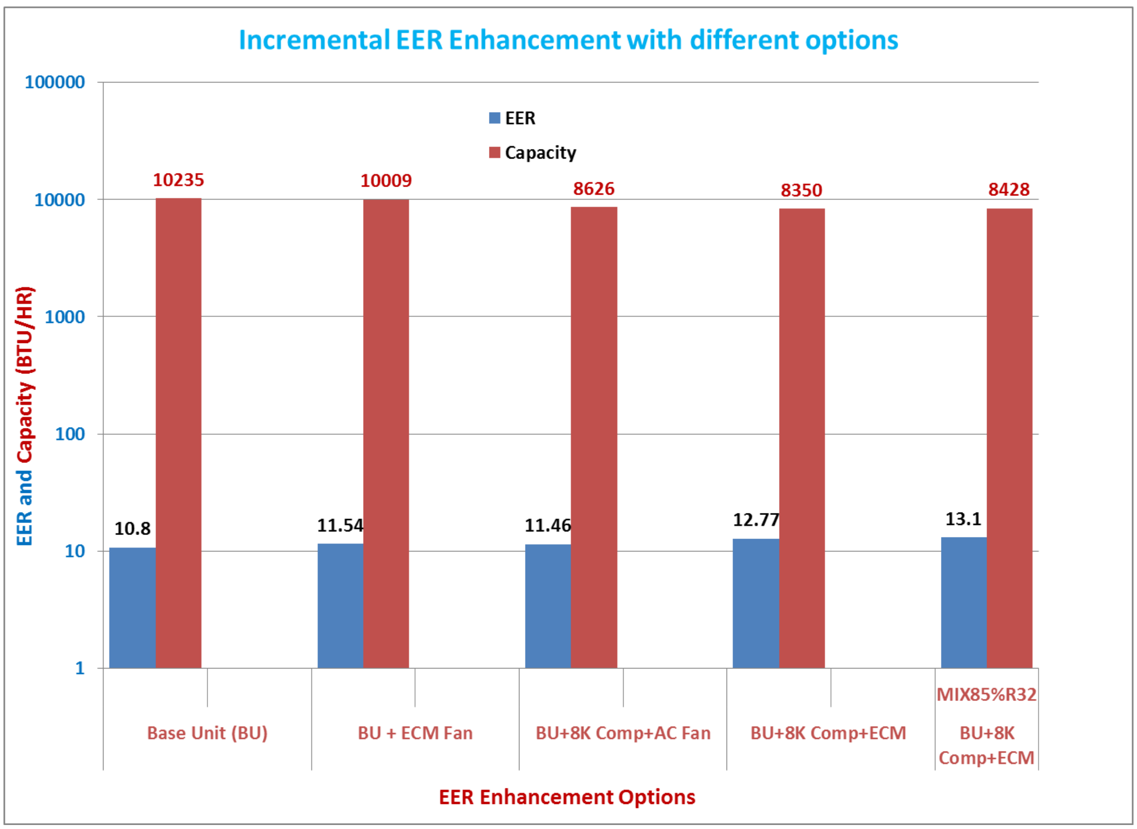

Figure 14: Incremental enhancement in EER along with the corresponding system capacity *(1 Btu/h = 0.293 W)* with different options

Each of the modifications tested here under laboratory conditions, comes with its own incremental cost and engineering challenges with regards to the eventual goal of producing a marketable, higher-efficiency WAC. The ECM fan motor was found to be a major improvement over the AC induction motor both for its main effect of using less power as well as the secondary effect of not dumping as much heat into the condenser section of the unit. From an engineering perspective, the integration of the ECM was straightforward as the dimensions of the motor used were well matched to the existing shroud assembly. For the cases tested, the increase in the R32 concentration of the refrigerant was found to give an incremental improvement over the standard R410A mixture. This blended mixture of R32/R125 has about a 34% lower GWP compared to R410A (1,377 vs. 2,079). The new design offers the technical potential of 0.1 quads of energy savings per year in USA alone and significantly more in other countries where WACs are used more widely such as in Asia, Africa and Middle East [Bansal (2015)]. This results in a number of additional advantages including the reduction in its operating cost, electrical demand, and the environmental impact.

6. CONCLUSIONS

This paper presented the design and development details of a high efficiency novel window air conditioner. A number of novel features were evaluated in the study including an ECM fan, a high efficiency compressor, large heat exchangers of the base unit and a ‘drop-in’ refrigerant mixture of R32/R125 (85%/15% molar concentrations). The use of the ECM fan motor alone in the baseline unit led to 8% EER improvement. This, when complimented with the high efficiency (but less capacity) compressor, led the EER to improve by 18.2%. Finally, the combination of the ECM fan motor, high efficiency compressor and the blend of R32/R125 (85/15 molar concentrations) enhanced the system EER by 21.3%. The new design offers the technical potential of 0.1 quads of energy savings per year in USA alone and significantly more in other countries where WACs are used more widely such as in Asia, Africa and Middle East. The potential improvement in WAC efficiency demonstrated an exemplary approach that would lead to considerable efficiency and environmental advantages, and its usefulness to the society.

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APPENDIX A

Table A1: Results of the design point uncertainty analysis

Base Measurements	
Measurement	**Estimated Combined Logger and Sensor Uncertainty**
Temperature	± 0.1°C
Refrigerant Pressure (low)	± 1.3 kPa
Refrigerant Pressure (high)	± 4.1 kPa
Mass Flow Rate	± 0.1 g/s
Electrical Power (fan)	± 2 W
Electrical Power (compressor and system total)	± 10 W
Atmospheric Pressure	± 0.06 kPa
Relative Humidity	± 1.4 %RH
Calculated Values	
Parameter	**Estimated Uncertainty**
EER	±.12 Btu/Wh
System Capacity	± 16.7 W
Condenser Heat Transfer Rate	± 8.2 W
State 5 Quality	± 0.002
State 1 Superheat	± 0.16 K
State 4 Subcool	± 0.16K
Refrigerant Evaporator Pressure Drop	± 2 kPa
Refrigerant Condenser Pressure Drop	± 5.5 kPa
Average Air Temperatures	± 0.05 K