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MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

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Development of High Efficiency Window Air Conditioner and Compressor Using Propane (R-290)

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Abstract

According to US EIA, 2009, there were 46.7 million window air conditioners (WAC) in the United States, accounting for 1.5% of the total U.S. residential energy use or about 0.33 quads.

Propane (R-290) is a very efficient refrigerant and promising for small packaged air conditioner and heat pump systems. It is a much less expensive, but more efficient refrigerant than R-22 and R-410A. Propane has negligible environmental impact. However, it is highly flammable, so, system charge must meet safety regulation.

A GE AEC12AV Energy Star WAC was used as a baseline, originally designed for R-410A. The unit uses a slinger wheel to spray condensate water on the condenser surface. It has a condenser with 5 mm tubes, an evaporator with 7 mm tubes and a rated cooling capacity of 12K Btu/hr, EER of 12.0.

Shanghai Highly (previous Hitachi) Electrical Appliances Co. Ltd, a business of Johnson Controls, is the CRADA partner that has developed a high efficiency propane rotary compressor, reaching outstanding efficiency at the target working conditions. The unit was modified by replacing the compressor, with the propane rotary compressor, and the expansion devices optimized for R-290.

Through experimental verification in our environmental chambers, the improved WAC achieved the project goals:

1) 260 grams charge limit proposed by EPA.

2) Same rated cooling capacity as the baseline WAC, and 17% higher EER, i.e. 14.0 EER.

3) Operated up to 131°F ambient temperature, suitable for mid-east, high ambient conditions.

In addition, we conducted system simulations using ORNL Heat Pump Design Model (HPDM) to investigate further potential for charge reduction. With using microchannel heat exchangers to replace the fin-and-tube condenser and evaporator, the system charge can be decreased to 150 grams. Use of the microchannel heat exchangers causes capacity and efficiency degradation. However, it can still achieve a rated EER > 12.0 and cooling capacity > 10,000 Btu/hr at the AHRI A condition, i.e. meeting the Energy Star criteria. It implies that the use of a high efficiency propane compressor makes up the loss due to reducing the heat exchanger volume, to meet the performance targets while satisfying the safety (system charge) regulation.

Statement of Objectives

The U.S. and mid-east residential AC markets have a significant portion of window air conditioners (WAC). Due to the compact size and small refrigerant charge, WACs are most tolerant of flammable refrigerants. Hydrocarbon (HC) refrigerants are natural substances and have significantly better environmental impacts and lower costs than conventional refrigerants like R-410A and R-22. Additionally, they have superior thermodynamic properties for air conditioning applications. The project aimed to develop a WAC prototype using propane (R-290) and verify the performance metrics in the laboratory. The key of developing a WAC using a flammable refrigerant is to control the refrigerant charge under the limit of building safety regulations and meet the Energy Star standard.

To minimize the charge, compact heat exchangers were utilized. To achieve the target efficiency, we collaborated with a leading rotary compressor manufacturer in the world, Shanghai Highly Electrical Appliances Co. Ltd, a business of Johnson Controls, to develop a compressor prototype specifically optimized for R-290, able to achieve superior energy efficiency.

Benefits to the Funding DOE Office's Mission

The overall objective of this project is to support the Building Technologies Office goal to reduce building energy use intensity (EUI) by 30% in 2030 vs. 2010 levels and comply with the Multi-Year Program Plan specific goals for the Emerging Technologies Program.

The project developed an advanced window air conditioner, achieving a superior efficiency of 14.0 EER with a rated cooling capacity of 12K Btu/hr. It is 17% more efficient than the current Energy Star WACs on the market. It can greatly contribute to the goal of enhancing energy efficiency, and result in significant energy savings nationwide.

Commercialization Possibilities

WACs are used widely in the world and AC units using hydrocarbon (HC) refrigerants, like R-290, are quickly emerging in the world. The development of high efficiency WACs using propane will help U.S. manufacturers to catch this trend, promote exports, and create domestic jobs.

For the next step, we will collaborate with U.S. manufacturers to put the high efficiency, propane WAC on the market.

Plans for Future Collaboration

Shanghai Highly Electrical Appliances Co. Ltd, is one of the world's largest compressor manufacturers. Its compressor products cover many applications, e.g. space cooling, heating, water heating, dehumidification, and electronics cooling. Their compressors work for many refrigerants, including HFCs, HCFCs, HFOs, natural refrigerants like propane and CO_2. Shanghai Highly Electrical Appliances Co. Ltd, a business of Johnson Controls, will keep supporting other research projects under the Building Technologies Office of DOE, providing customized and optimized compressor samples, to help the Building Equipment Team of ORNL reach project targets.

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1. Introduction

Window air conditioners (WACs) are low-cost and sold in large numbers internationally to provide cooling and improve comfort in apartment buildings. According to the U.S. Energy Information Administration (EIA) there were approximately 46.7 million WACs operating within the U.S. in 2009, accounting for approximately 1.5% of the total U.S. residential energy use. Due to environment concerns, the international community, e.g. Europe and Mideast, is imposing new rules on refrigerant selections. U.S. manufacturers need to accommodate the requests to grow their product exports and create domestic jobs.

Propane (R-290) is a natural substance which has a significantly better environmental friendliness and creates 700 times less pollution than conventional refrigerants, e.g. R-410A and R-22. Additionally, propane has superior thermodynamic properties for air conditioning applications.

On the other hand, propane is highly flammable and its safety concern must be addressed before being adopted by the market. Due to the compact size and small refrigerant charge volume, WACs are most tolerant of flammable refrigerants and suitable for propane. The keys of designing a WAC using a flammable refrigerant is to control the refrigerant charge under the limit of the U.S. building safety regulations and achieve higher efficiency than the max efficiency on the market to realize the refrigerant's advantage in energy efficiency.

From 2017 to 2018, ORNL researchers, collaborating with Shanghai Highly Electrical Appliances Co. Ltd, developed a window air conditioning unit that uses propane as the refrigerant, cooling the air with 17% higher efficiency than ENERGY STAR® commercial units. The prototype integrates novel, compact heat exchangers, compressor and controls, which are specifically optimized for propane. The unit—the first propane window air conditioner to meet U.S. building safety standards—has exceeded performance targets in laboratory evaluations.

Shanghai Highly Electrical Appliances Co. Ltd, is a subsidiary of Johnson Controls Inc. U.S., a company founded in 1993 and headquartered in Shanghai, China. It is a world leading manufacturer of rotary compressors and produces 18 million compressors annually, covering wide categories of air conditioning, water heating and refrigeration. It has a capital of 1.2 billion U.S. dollars, and multiple manufacturing sites in China and India. In fact, approximately 15% of all compressors in the world are made by the company. The company is our partner developing an advanced R-290 rotary compressor used for the high efficiency WAC.

The system is designed to meet the following three technical goals:

1. System refrigerant charge < 260 g as proposed by U.S. Environment Protection Agency (EPA, 2014).
2. Rated cooling capacity > 10,000 Btu/hr, at the AHRI A condition (95°F ambient temperature, 80°F indoor dry bulb/67°F indoor wet bulb temperature).
3. Rated EER reaches 12.0 (Energy Star), at the AHRI A condition.

2. Baseline Window Air Conditioner

We selected a GE AEC12AV WAC, using R-410A, as the baseline. Some general features of the WAC are listed below:

- Power supply: single-phase, 115 V/60 HZ
- Rated capacity: 12,050 Btu/hr
- Rated EER: 12.1
- Fan speed: 3 speeds
- Indoor air flow rate (Maximum): 290 CFM
- Dehumidification rate: 3.4 pts/hour, i.e. equivalent SHR – 68%.
- System charge: 750 grams, i.e. 1.65 lbms

The schematic diagram of the WAC and its P-h diagram are shown in Figures 1 and 2, respectively. The open WAC unit is shown in Figure 3. The condenser fan blade is specially configured 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, cooling the air drawn across the coil and enhance the condenser heat transfer. This feature is called the "sling" effect. Figures 4 and 5, respectively, show the single axis fan and the "slinger ring." A subcooler downstream of the condenser coil is submerged in a water collection pan, which collects condensed water from the indoor evaporator coil.

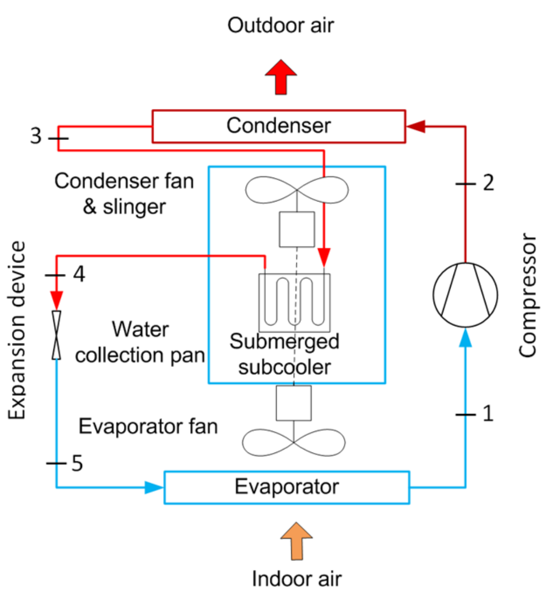

Figure 1: Schematic of Window Air Conditioner

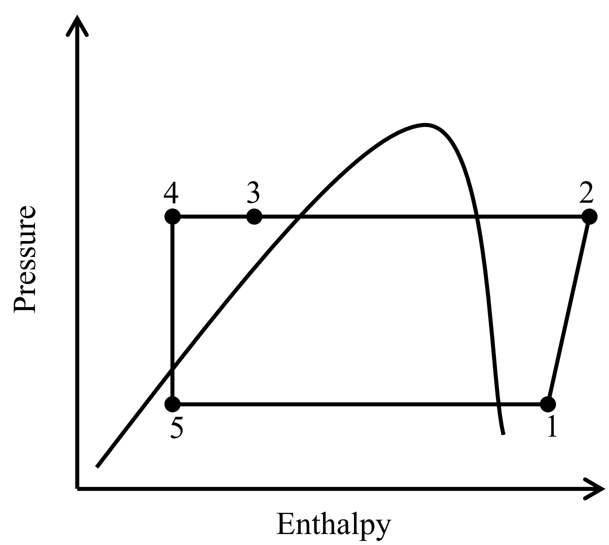

Figure 2: P-h diagram of Window Air Conditioner

Figure 3: Baseline GE Window Air Conditioner

Figure 4: Single axis blower/fan

Figure 5: Slinger on condenser fan

The WAC uses fin-and-tube coils for the evaporator and condenser. Some basic parameters of the heat exchangers are given in Table 1.

Table 1: Condenser and Evaporator of Window Air Conditioner

Parameters	Fin-&-Tube Condenser Coil	Fin-&-Tube Evaporator Coil
Face area [ft^2]	1.44	0.94
Total Tube Number	56	48
Number of rows	3 (cross counter-flow)	4 (cross counter-flow)
Tube outside diameter [in]	0.2 (5 mm)	0.33
Finned tube length [in]	18.2	15.0
Tube pitch [in]	0.6	0.75
Row pitch [in]	0.5	0.7
Fin density [fins/in]	20	22
Fin length [in]	11.4	9.0
Fin depth [in]	1.5	2.8

Tube connection patterns of the condenser and evaporator can be seen in Figures 6 and 7, where green cells represent numbered tubes, arrows indicate refrigerant flow direction, and "JUNC#" means a refrigerant flow intersection port. The condenser coil has seven circuits and five refrigerant flow intersections, as numbered from zero and described in Figure 6. The evaporator coil has six circuits (with two feeds in and four out) and four refrigerant flow intersections as described in Figure 7.

Figure 6: Tube connections in condenser

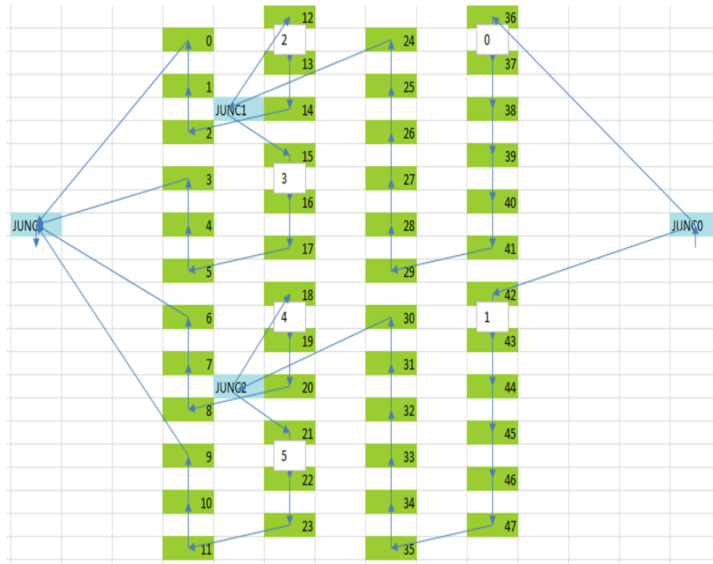

Figure 7: Tube connections in evaporator

The submerged subcooler has eight tubes, having an outside diameter (OD) of 0.28 inch and total length of 12.7 feet. The liquid line is one-foot long and has an OD of 0.28 inch. The discharge line has a length of 3.3 feet and an OD of 0.32 inch. The length and inner volume of the suction line can be ignored as containing only low-density vapor.

It should be emphasized that there are three main reasons for selecting GE AEC12AV as the baseline unit:

1) It uses 5 mm tubes for the condenser where a major part of refrigerant liquid stays. The smallest diameter tubes for fin-and-tube coils on the market is 5mm. That minimizes the system charge.
2) The condenser uses a slinger wheel to spray the condensate water for condenser evaporative precooling, which leads to high condenser heat transfer effectiveness with a relatively restricted volume.
3) The GE AEC12AV is already an ENERGY STAR unit having the rated EER of 12.1, and capacity of 12 K Btu/hr.

3. Compressor and Heat Exchanger Models

The DOE/ORNL Heat Pump Design Model (HPDM) (Shen, 2014) was used for designing the WAC. HPDM is a hardware-based simulation model that uses the Newton-Raphson method to solve simultaneous system equations. The component heat exchanger (HX) models have different levels of complexity which fall into three categories: bulk models, phase-to-phase models, and segment-to-segment heat exchanger models. These are used to build a heat exchanger having arbitrary circuitry, geometry, and represent any boundary conditions. All phase-to-phase and segment-to-segment heat exchanger models can calculate refrigerant charge inventory. For the system modeling, a component-based modeling framework has been developed that allows connecting steady-state component models in any manner.

Compressor:

HPDM provides multiple choices per AHRI standard 540 [ANSI/AHRI (2010)] for modeling a single-speed compressor. A 10-coefficient compressor map from the compressor manufacturer has been used here to model the baseline WAC unit using R410A and R-290. It simulates energy balance from inlet to outlet using the calculated power and given heat loss ratio and it also considers the actual suction state to correct the map mass flow predictions.

The compressor efficiencies are defined as isentropic $\eta_{isentropic}$ and volumetric efficiency η_{vol} as given in Equations 1 and 2.

$$m_r = Volume_{displacement} \times Speed_{rotation} \times Density_{suction} \times \eta_{vol} \tag{1}$$

$$Power = m_r \times (H_{discharge,s} - H_{suction})/\eta_{isentropic} \tag{2}$$

Where $\boldsymbol{m_r}$ is compressor mass flow rate; $\boldsymbol{Power}$ is compressor power; $\boldsymbol{\eta_{vol}}$ is compressor volumetric efficiency; $\boldsymbol{\eta_{isentropic}}$ is compressor isentropic efficiency; $\boldsymbol{H_{suction}}$ is compressor suction enthalpy; $\boldsymbol{H_{discharge,s}}$ is the enthalpy obtained at the compressor discharge pressure and suction entropy.

Expansion Device:

Expansion device was modeled as an isenthalpic process where both the degrees of superheat and sub-cooling were specified as inputs.

Fans and Blowers:

For a given airflow rate, the model normally uses the fan curve to simulate static head, power consumption, and calculate air-side temperature increment from inlet to outlet. However, in this study, we did not use a fan curve. Instead, we directly used the air flow rate and the corresponding power consumption measured in the experiment.

Segment-to-Segment Heat Exchanger Models:

A segment-to-segment modelling is used to model evaporator and condenser. The segment-to-segment modeling approach is commonly used to model fin-tube (FTC) heat exchangers, by dividing each single tube into numerous segments. Each tube segment has individual air side and refrigerant side entering states, and considers possible phase transition, as shown in Figure 8. Refrigerant side heat transfer, pressure drop gradient, and void fraction can be considered specific to local quality and flow pattern.

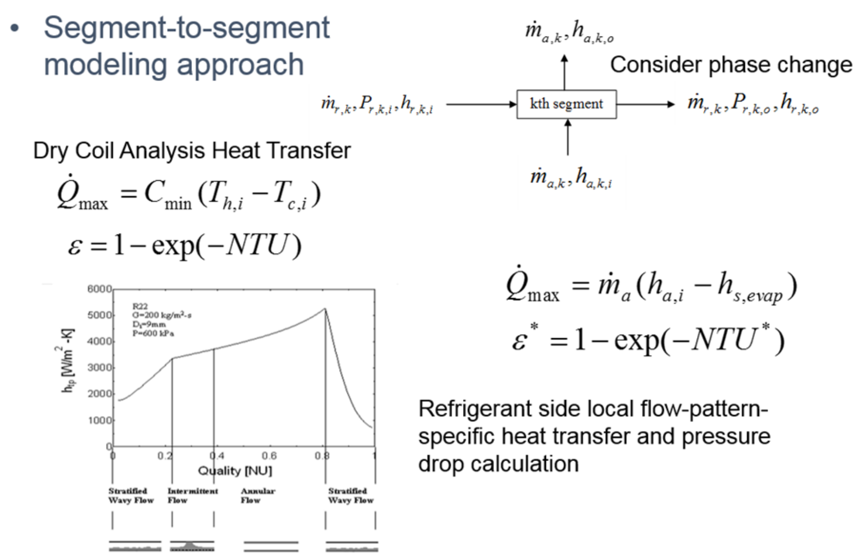

Figure 8: Segment-to-Segment Modeling approach to Simulate Control Volume of a Tube Segment

The heat transfer rate $\dot{Q}$ in each mini-segment is calculated using the $\varepsilon - NTU$ method. The pressure drop of the refrigerant $\Delta P_{r,k}$ is determined by the frictional pressure drop and momentum pressure drop within the port segment. In addition, the refrigerant mass in the tube segment can be determined with the known refrigerant properties and inner volume. The essence of using finite segments is that the substance properties can be treated as constant within each segment. Certainly, smaller segments lead to better accuracy. However, it is at the expense of calculation time.

<u>Refrigerant-to-Air Heat Transfer in Segment-to-Segment Heat Exchanger Models:</u>

For elements where no moisture condenses, the effectiveness correlations for sensible heat transfer are used to calculate the heat transfer between the refrigerant and air. The fin efficiency is calculated as Equation 3:

$$\eta_F = \frac{\tanh(mL_f)}{mL_f} \tag{3}$$

where η_F is the fin efficiency for heat transfer relative to the maximum heat transfer if the whole fin temperature is the same as the fin base temperature. L_f is the fin length in each control volume. In addition, $m = \sqrt{\frac{2h_o}{k_f \delta_f}}$, k_f is the fin thermal conductivity, h_o is the airside heat transfer coefficient, and δ_f is the fin thickness.

The overall efficiency relative to the entire surface area is calculated as,

$$\eta_o A_o = (A_o - A_F) + \eta_F A_F \tag{4}$$

where A_F is the fin surface area of each divided port, and A_o is the exposed port surface area including the port outside surface and the fin surface.

The overall heat transfer conductance UA is composed of three parts: the air side, the tube wall and the refrigerant side.

$$\frac{1}{UA} = R_o + R_{port} + R_i = \frac{1}{\eta_o A_o h_o} + R_{port} + \frac{1}{h_i A_i} \tag{5}$$

where h_i is the refrigerant side heat transfer coefficient, A_i is the port inside heat transfer area, and R_{port} is the tube wall thermal resistance, assuming each port is a mini round tube.

<u>Refrigerant-to-Air Heat & Mass Transfer in Segment-to-Segment Heat Exchanger Models:</u>

In the case of water condensing on an evaporating coil, both heat and mass transfer need to be considered. The effectiveness of heat and mass transfer is still defined as $\varepsilon^* = \frac{\dot{Q}}{\dot{Q}_{max}}$. The approach for modeling a wet coil is based on Braun (1989), which introduced the parameter of the air saturation specific heat at the refrigerant temperature to model wet coils, where $c_s = (\frac{dh_{sat,air}}{dT})_{T=T_{ref}}$. Using the air saturation specific heat, the existing effectiveness relationships developed for sensible heat transfer can also be used for heat and mass transfer, based on the modified definitions for the number of transfer units and capacitance rate ratio. The details follow.

The driving potential for heat transfer in the case of a wet coil is the difference between the enthalpies of the inlet air and the saturated air enthalpy at the refrigerant temperature. Then, the maximum possible heat transfer rate $\dot{Q}_{max}$ is defined by:

$$\dot{Q}_{max} = m_a(h_{a,i} - h_{s,evap}) \tag{6}$$

Heat and mass transfer fin efficiency:

$$\eta_F^* = \frac{\tanh(m^* L_f)}{m^* L_f} \tag{7}$$

where $m^* = \sqrt{\frac{2h_o / C_{p,m}}{k_f / C_s \times \delta_f}}$, $C_{p,m}$ is the specific heat of the wet air, m_a is the air mass flow rate, $h_{a,i}$ is the air inlet enthalpy, $h_{s,evap}$ is the saturated air enthalpy at the refrigerant temperature. The overall heat and mass fin efficiency relative to the overall surface area is calculated as,

$$\eta_o^* A_o = (A_o - A_F) + \eta_F^* A_F \tag{8}$$

Overall heat and mass transfer conductance UA^*:

$$\frac{1}{U_o^* A} = \frac{C_{p,m}}{\eta_o^* h_o A_o} + C_s R_{tube} + \frac{C_s}{h_i A_i} \tag{9}$$

Modified number of transfer units NTU^* for the wet coil analysis:

$$NTU^* = \frac{U_o^* A_o}{m_a} \qquad (10)$$

Modified capacitance rate ratio for the wet coil analysis: $Cr^* = \frac{m_a}{m_r(C_{p,r}/C_s)}$. This parameter is used in a similar way in the heat and mass transfer analysis as C_r is used in the relationships for the effectiveness of the sensible heat transfer analysis.

Relationships of heat and mass transfer effectiveness ε^*: The heat and mass transfer effectiveness are a function of the modified number of transfer units and the modified capacitance rate ratio, $\varepsilon^* = f(NTU^*, Cr^*)$. The existing effectiveness models for sensible heat exchangers can then be used directly for wet coil analysis after using NTU^* to replace NTU, and Cr^* to replace C_r. The heat and mass transfer of a two-phase section is calculated by,

$$\varepsilon^* = 1 - \exp(-NTU^*) \qquad (11)$$

For a single-phase section, the effectiveness correlations for cross-flow with one mixed (refrigerant) and one unmixed (air) fluid are used.

After the heat transfer rate $\dot{Q}$ is determined, the air side parameters can be obtained using the air side analysis. The air enthalpy flowing out of the segment is determined by,

$$h_{a,o} = h_{a,i} - \frac{\dot{Q}}{m_a} \qquad (12)$$

The saturation air enthalpy at the tube surface is

$$\bar{h}_{s,s,o} = h_{a,i} - \frac{h_{a,i} - h_{a,o}}{1 - \exp(-NTU_o^*)} \qquad (13)$$

where $NTU_o^* = \frac{\eta_o^* h_o A_o}{m_a C_{p,m}}$.

The effective tube surface temperature of the wet tube $\bar{T}_{s,o}$ is the saturated air temperature corresponding to the enthalpy of $\bar{h}_{s,s,o}$ at the barometric pressure.

Then the outlet air temperature can be determined using a sensible heat transfer calculation,

$$T_{a,o} = \bar{T}_{s,o} + (T_{a,i} - \bar{T}_{s,o})\exp(-NTU_o) \qquad (14)$$

where $NTU_o = \frac{\eta_o h_o A_o}{m_a C_{p,m}}$. is for sensible heat transfer.

To model a port segment in an evaporator, the analysis can be conducted at first for the wet coil to decide the effective surface temperature $\bar{T}_{s,o}$. If $\bar{T}_{s,o}$ is smaller than the inlet dew point temperature, the coil is considered wet. Otherwise, the heat transfer rate is determined using the dry coil heat transfer analysis.

Microchannel heat exchangers can be handled in a similar way like fin-and-tube heat exchangers, by dividing each single port into mini-segments. As shown in Figure 9, each port is divided to smaller segments. Heat transfer, pressure drop, and charge inventory are calculated segment by segment. Each segmented control volume contains a single port. Heat transfer and pressure drop at the air and refrigerants sides are considered within each control volume. Air flows port-to-port along the micro-channel tube and fin surface.

There are many port shapes, for example, round, rectangular, trapezoidal, triangular, etc. The port hydraulic inside diameter is treated as the characteristic dimension to calculate heat transfer, pressure drop and void fraction. The real shape of a port is used to calculate its cross-sectional flow area for determining the inner volume, charge inventory, and inside heat transfer surface area. Each port's outside heat transfer surface area is calculated using the microchannel tube total outside surface area divided by its number of ports. The fins are treated as straight fins.

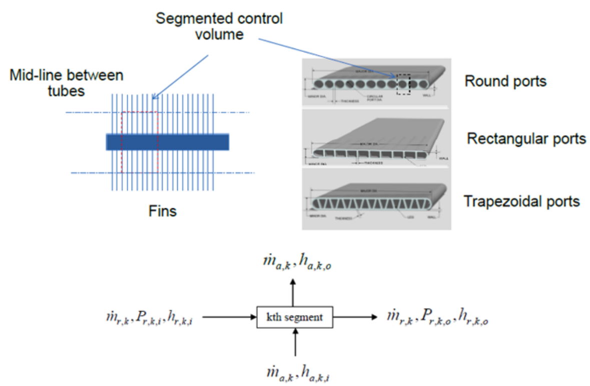

Figure 9: Segmented Control Volume of a Mini-Port

It should be mentioned that Hughmark (1962) void fraction model was used to calculate the two-phase refrigerant density, for fin-and-tube coils and microchannel heat exchangers. For the ports other than round shape, hydraulic diameters are used for the void fraction calculations.

Submerged subcooler:

The subcooler model considers phase transition in the heat transfer section, i.e. allowing two-phase or liquid refrigerant entrance [LBNL (1997)]. It assumes natural convection at the water side. The water pool temperature is a measured input. Effectiveness-NTU method is used to calculate energy transfer rate between the refrigerant and water.

The "slinger" effect:

The slinger sprays water droplets into the air stream flowing over the condenser coil surface. Instead of modeling the heat and mass transfer process, a simple approach was adopted here to treat the slinger effect as an air side heat transfer enhancement factor from the experimental data.

Experiments were performed in the psychrometric chamber, with strictly controlled indoor condition at 80ºF DB/67ºF WB (26.7ºC/19.4ºC). The slinger effect is modeled [LBNL (1997)] as a function of the water condensate amount sprayed on the condenser coil. For this single-speed WAC, the only factor impacting the water condensate amount is the outdoor air temperature. Thus, for comparing the slinger effect, the outdoor air temperature was varied from 90ºF to 110ºF (32.2ºC to 43.3ºC) in the outdoor chamber, which resulted in different amounts of water condensate, and hence varying sling effect. Figure 10 compares the model predicted air side heat transfer enhancement multipliers due to the sling effect to laboratory data deduced heat transfer multipliers, as a function of the ambient temperature. The laboratory data deduced heat transfer multipliers were obtained by adjusting air side heat transfer coefficient of the condenser model to match the measured performance, assuming no sling effect. Since there is a large dispersion in the laboratory measurements, deviations between the laboratory deduced and model predicted heat transfer multipliers can be up to 30%. However, the average multipliers are close; with laboratory data deduced multiplier being 1.33, and the model predicted average multiplier being 1.24.

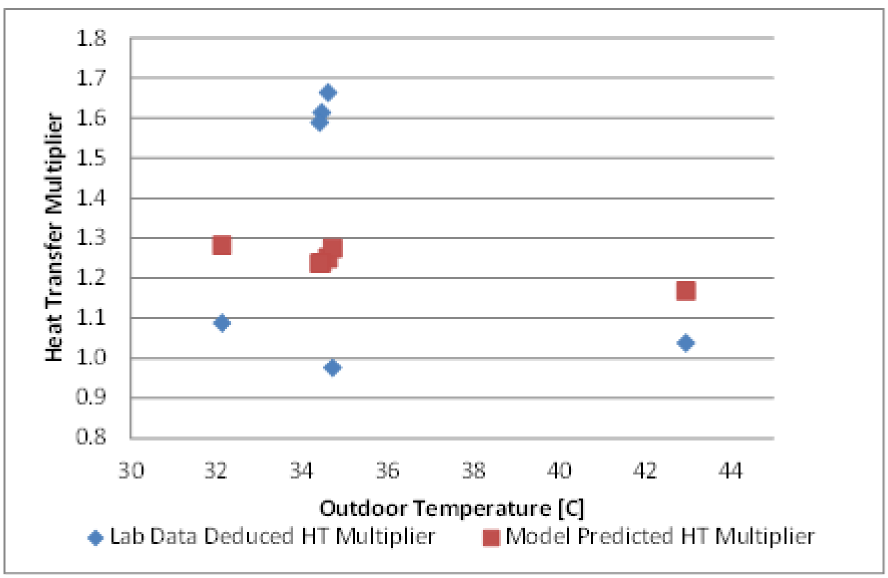

Figure 10: Heat transfer enhancement ratio due to 'sling effect' vs. outdoor temperature

4. Model-Based Design Improvements

We used the DOE/ORNL Heat Pump Design Model to model the GE baseline WAC using R-410A and calibrated the system to match the available product data. Table 2 compares the simulation with the product data, at the AHRI A condition.

Table 2: R-410A WAC model validation at the AHRI A condition

	Predicted by calibrated model	Published product data
Cooling Capacity [Btu/hr]	12,069	12,050
EER [Btu/hr/W]	12.1	12.1
SHR [%]	69%	68%
System charge [grams]	732	750
Evaporating temperature [F]	50.0	N/A
Condensing temperature [F]	124.0	N/A
Evaporator exit superheat degree [R]	10	N/A
Condenser subcooling degree [R]	15	N/A
Subcooler temperature drop [R]	6.4	N/A

Uncertainty of system charge prediction is 732 grams (prediction) versus 750 grams (product), i.e. 2.5% under-predicted. The baseline WAC uses a single-speed, rotary compressor, made by Shanghai Highly Electrical Appliances Co. Ltd, with the brand name of "Highly". We selected an R-290 compressor available on the market from the same manufacturer to achieve the similar capacity and efficiency. A side-to-side comparison between the two compressors is given in Table 3.

Table 3: Comparing Compressors

Model No	ASD100HW-H6KUN (Highly)	PSD162XW (Highly)
Refrigerant	R-410A	R-290
Power supply	115 V/60 HZ/single-phase	115 V/60 HZ/single-phase
Displacement [ml]	10.0	16.2
Amount of oil charge [ml]	230	270
Space volume of inner case [ml]	1175	1175
Rated capacity* [Btu/hr]	9,827	10,014
Motor input* [W]	970	894
Rated COP* [W/W]	2.97	3.28
Volumetric efficiency* [%]	87%	91%
Isentropic efficiency* [%]	67%	67%
Estimated refrigerant mass in oil [g]	40	40

*the rated values were obtained at a standard test condition: evaporating temperature 7.2°C (45°F); condensing temperature: 54.4°C (130°F); liquid temperature entering expansion valve: 46.1°C (115°F); compressor return gas temperature: 35°C (95°F); ambient temperature: 35°C (95°F).

As described in Table 3, R-290 requires 62% larger displacement volume for the similar cooling capacity. The two compressors have the same rated isentropic efficiency, i.e. 67%, but the rated COP of the R-290 compressor is 10% higher than the R-410A compressor, because R-290 is a more efficient refrigerant based on its thermodynamic cycle properties.

Using the same heat transfer calibration factors as the simulation in Table 2, the system model was used to model the R-290 WAC, by changing the refrigerant to R-290 and replacing the compressor with Highly

PSD162XW. For the R-290 compressor, we specified the known displacement volume and set the volumetric efficiency as 94% and the isentropic efficiency as 70%, i.e. a similar isentropic efficiency as the R-410A compressor. Table 4 below gives the predicted values with and without the submerged subcooler, at 15°R condenser subcooling and 10°R evaporator exit superheat.

Table 4: R-290 system predictions with/without the submerged subcooler at the AHRI A condition

	With submerged subcooler	Without submerged subcooler
Cooling Capacity [Btu/hr]	11792	11415
EER [Btu/hr/W]	12.5	12.0
SHR [%]	70%	71%
System charge* [g]	328	263

* The system charge predictions were adjusted by adding 2.5% uncertainty.

It can be seen from Table 4, to match the charge target below 260 grams, the submerged subcooler must be eliminated. Without the submerged subcooler, we can still achieve an EER >= 12.0, and cooling capacity > 10,000 Btu/h. The predicted charge is 263 grams, slightly higher than required.

In summary, three major improvements were made to convert the R-410A baseline unit to a R-290 WAC within the charge limit.

1) Replaced the R-410A rotary compressor with a high efficiency R-290 compressor.
2) Optimized the expansion device, i.e. two parallel capillary tubes for the R-290 system. As a result, the optimized capillary tube length for R-290 was about half of the ones used for R-410A having the same inside diameter.
3) Removed the submerged subcooler. It can decrease the system charge by 65 grams, while only degrading the efficiency minimally.

5. Compressor Improvement

We rely on using a high efficiency R-290 rotary compressor to achieve the target EER. The baseline WAC uses a single-speed, R-410A rotary compressor, made by Shanghai Highly Electrical Appliances Co. Ltd. We established a collaboration relationship with the same manufacturer and they made a specifically optimized, high efficiency R-290 prototype compressor to achieve similar capacity and higher efficiency than the baseline R410A compressor. Based on the PSD162XW on the market, they made improvements via optimizing the compressor structure for R-290 and using a higher efficiency motor to improve the compressor overall efficiency. The manufacturer tested the improved compressor prototype in a calorimeter and developed a compressor map covering suction saturation temperatures from 45 to 55°F, and discharge saturation temperatures from 110 to 130°F. The compressor demonstrated outstanding performance. Figure 11 shows the measured compressor isentropic efficiency as a function of the suction and discharge saturation temperatures. The improved R-290 compressor has approximately 80% isentropic efficiency, which is the top level on the market. Its efficiency is about 14% higher than the baseline R-410A compressor (ASD100HW-H6KUN) and the original R-290 compressor (PSD162XW) on the market.

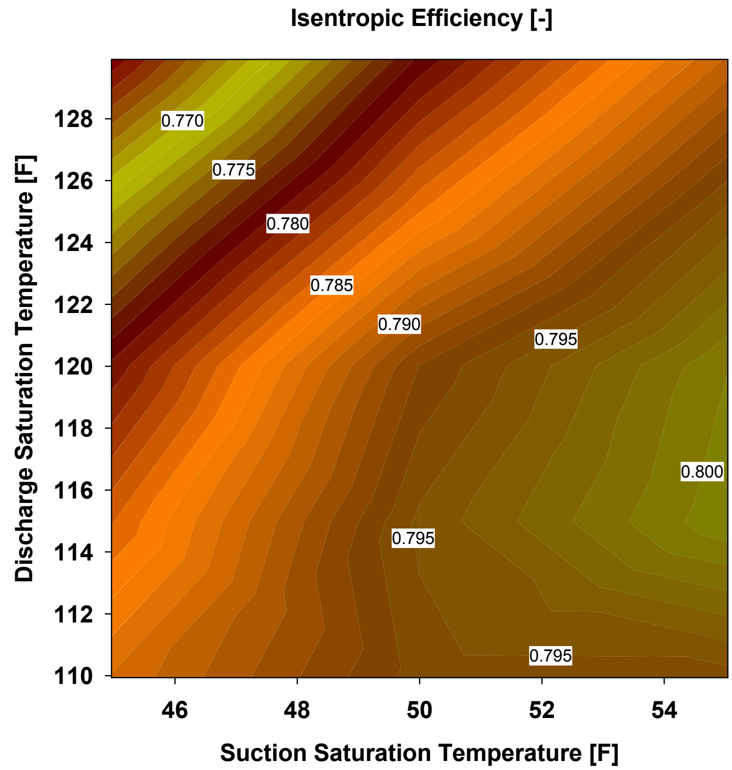

Figure 11: Isentropic Efficiency Map of R-290 Sample Compressor

6. Experimental Verification

We made three modifications described in Section 4, i.e. optimized the expansion device (cutting 50% shorter the original two-parallel capillary tubes); eliminated the submerged subcooler; used an improved R-290 compressor prototype with 14% efficiency enhancement than the baseline. The WAC was installed in our environmental chambers to verify the performance experimentally.

Laboratory Instrumentations

Figure 12 illustrates laboratory instrumentations, which includes three refrigerant pressure transducers at the compressor suction, discharge and liquid line; two inserted probe thermocouples to measure the refrigerant temperatures at the compressor suction and liquid line; five wire thermocouples were soldered on tube surface and covered with thermal insulation, one at the compressor discharge, and four placed on exits of the four circuits in the evaporator, to indicate the refrigerant flow distribution. As shown in Figures 13 and 14, six wire thermocouples were placed uniformly to measure the average condenser inlet air temperature; six wire thermocouples were used to measure the average indoor return air temperature; three wire thermocouples to measure the average supply air temperature; one relative humidity (RH) sensor was placed at the return side and the other RH sensor was placed at the supply side, to measure the return and supply humidity. The air side capacity was calculated using the measured air enthalpy differential from the return to the supply, multiplied by the indoor air flow rate given in the product specification, i.e. 295 CFM; the refrigerant side capacity was calculated using the measured refrigerant enthalpy difference from the liquid line to the compressor suction, multiplied by the refrigerant mass flow predicted by the compressor map. A watt transducer was used to measure the total power consumption, including the compressor and fan power.

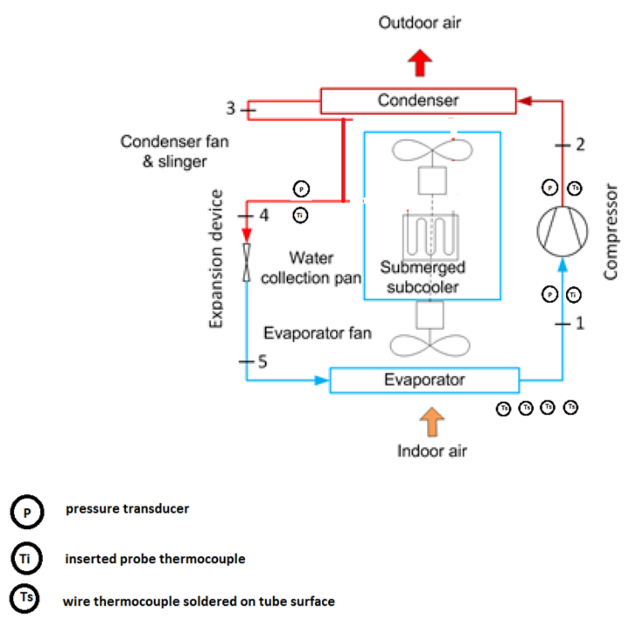

Figure 12: Laboratory Instrumentations

Figure 13: Indoor Side of WAC in Environmental Chambers

Figure 14: Outdoor Side of WAC in Environmental Chambers

Test Conditions

We installed the prototype WAC in ORNL's environmental chambers and evaluated it under extensive operating conditions. Table 5 provides a summary of the set points used for the outdoor and indoor chambers. The "Hot" and "Extreme" conditions are developed to evaluate the performance of the unit at climates like those experienced in the hot climate zone countries.

Table 5: Test Conditions

Test condition	Outdoor Dry-Bulb Temperature °C (°F)[a]	Indoor Dry-Bulb Temperature °C (°F)	Wet-Bulb Temperature °C (°F)
AHRI B[b, †]	27.8 (82)	26.7 (80.0)	19.4 (67)
AHRI A[b]/AHRI	35.0 (95)	26.7 (80.0)	19.4 (67)
T3*[c]	46 (114.8)	26.7 (80.0)	19 (66.2)
T3[d]	46 (114.8)	29 (84.2)	19 (66.2)
Hot	52 (125.6)	29 (84.2)	19 (66.2)
Extreme	55 (131)	29 (84.2)	19 (66.2)

[a] There is no specification for the outdoor relative humidity as it has no impact on the performance.

[b] Per AHRI Standard 210/240

[c] T3* is a modified T3 condition in which the indoor settings are similar to the AHRI conditions - only used for mini-split AC units.

[†] Only used for to evaluate the performance of the mini-split AC units.

[d] Per ISO 5151 Standard

Laboratory Measured Performance Results

The prototype WAC was charged with 260 +/- 5 grams of propane (R-260), with the optimized capillary tubes (two parallel) and improved compressor for R-290. The measured performance results, under the conditions of Table 5, are given in Table 6. The measured results were recorded as average values during one-hour steady-state tests, after the WAC reached steady-state operation for half hour. Figure 15 presents the measured air side cooling capacity and EER as a function of the outdoor air temperature.

Figure 15 and Table 6 demonstrate;

1. The WAC reached outstanding performance at the AHRI A condition, i.e. achieved 14.0 EER and 12K Btu/hr air side cooling capacity. These have exceeded the project goals, with 17% higher EER.
2. The energy balance between the air and refrigerant side cooling capacities was checked at the AHRI A condition, using the compressor-map predicted refrigerant mass flow rate to calculate the refrigerant side cooling capacity. The energy balance is 0.8%. The predicted compressor power by

the compressor map was 716.2 Watts, i.e. 2.0% smaller than the measured power. The consistencies to the compressor map confirm the measurement accuracy.

3. We charged the system with 260 +/- 5 grams of R-290 and re-optimized the length of the two parallel capillary tubes. It led to reasonable system balance points, having compressor suction superheat degree of 17.4 R and condenser exit subcooling degree of 11.7 R.
4. The WAC passed the extreme condition at 131°F, having an EER of 7.5 and 75% cooling capacity relative to the AHRI A condition. The resultant discharge temperature is 183°F, well below the compressor's high temperature limit, e.g. 210°F. This proves the unit capable of the hot climate zone countries.
5. The prototype WAC reached the outstanding efficiency for three reasons: 1) The R-290 rotary compressor delivers top efficiency, having isentropic efficiency around 80%; 2) R-290 has better thermodynamic cycle efficiency than R-410A and R-22; and 3) the sling effect of the baseline WAC, i.e. spraying water condensate on the condenser coil, enhanced the heat transfer.
6. In comparison to the baseline unit, the new design uses less material, i.e. removing the submerged subcooler, and half length of the original capillary tubes. The compressor is a single-speed, rotary compressor from the same supplier, and the fan is the same. Consequently, the R-290 WAC should have comparable or even lower cost than the baseline R-410A WAC.

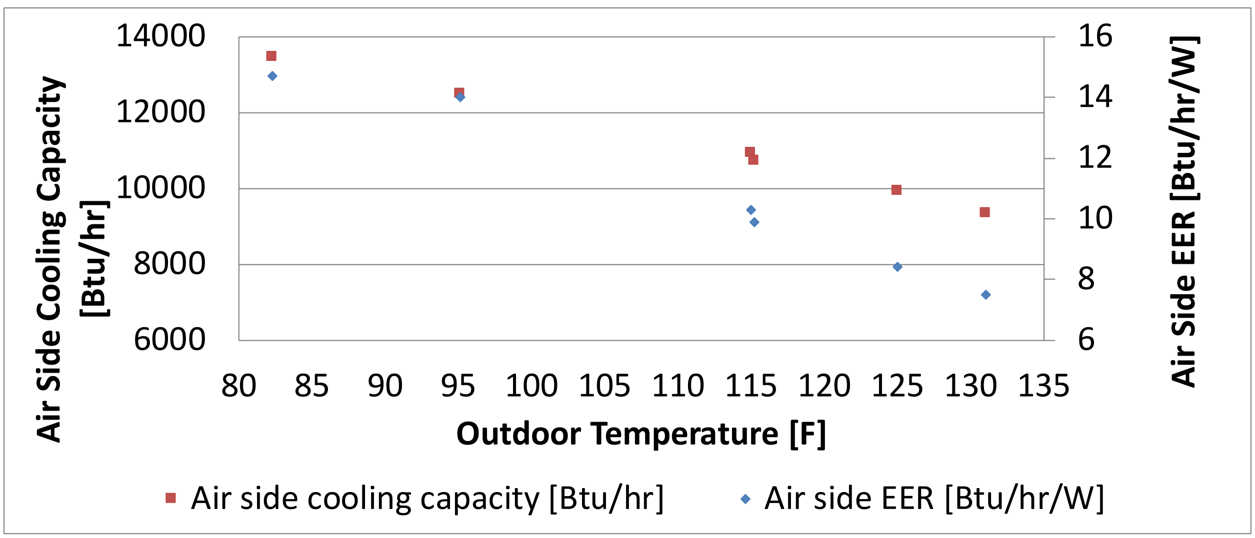

Figure 15: Measured air side cooling capacity and EER vs. outdoor air temperature

Table 6: Test Results

Parameters	AHRI A	AHRI B	T3*	T3	Hot	Extreme
Suction saturation temperature [F]	51.3	50.2	52.4	52.9	54.1	54.8
Discharge saturation temperature [F]	115.9	104.0	133.2	134.9	N/A[1]	N/A[1]
Discharge temperature [F]	141.8	128.6	162.4	165.8	176.7	182.8

Compressor suction superheat degree [R]	17.4	17.9	16.9	18.4	18.1	17.8
Condenser exit subcooling degree [R]	11.7	14.7	6.1	5.2	N/A[1]	N/A[1]
Barometric air pressure [psia]	14.8	14.8	14.8	14.8	14.8	14.8
Indoor return air temperature [F]	80.2	80.1	80.4	84.3	84.4	84.4
Indoor return air relative humidity [%]	53.2%	53.5%	51.6%	41.2%	42.0%	42.1%
Indoor return wet bulb temperature [F]	67.8	67.8	67.5	67.3	67.6	67.7
Indoor supply air temperature [F]	56.0	54.7	57.6	57.8	59.3	60.2
Indoor supply air relative humidity [%]	90.0%	91.0%	89.0%	87.4%	86.9%	86.5
Outdoor air temperature [F]	95.1	82.3	115.0	115.2	125.0	131.0
Air side cooling capacity [Btu/hr]	12477	13452	10938	10732	9926	9337
Sensible heat ratio [%]	64%	62%	68%	80%	82%	84%
Refrigerant side cooling capacity [Btu/hr]	12373	N/A[2]	N/A[2]	N/A[2]	N/A[2]	N/A[2]
Energy balance (refr - air)/air [%]	-0.8%	N/A[2]	N/A[2]	N/A[2]	N/A[2]	N/A[2]
Total power consumption [W]	891.5	789.5	1064.2	1082.4	1182.1	1240.6
Compressor Power consumption [W]	731.5	629.5	904.2	922.4	1022.1	1080.6
Air side EER [Btu/hr/W]	14.0	14.7	10.3	9.9	8.4	7.5

1. At hot and extreme conditions, the discharge pressure exceeded high limit (300 psia) of the pressure transducer, and thus, no high side pressures and subcooling measurements were available.
2. The compressor map was not given beyond the AHRI A condition, and thus, no mapped refrigerant mass flow rates and refrigerant side cooling capacities were available for the other conditions.

To summarize the experimental study, we tested the prototype R-290 WAC under extensive operation conditions from 82°F to 131°F outdoor temperature. At the rated AHRI A condition, the unit reached 14.0 EER and 12K Btu/hr air side cooling capacity. These have exceeded the project goals, with 17% higher EER and the target cooling capacity of 12K Btu/hr, within the charge limit of 260 grams. The unit also passed the extreme condition, indicating it capable for hot climate zone countries like Mideast. Additionally, the prototype uses less material than the baseline R-410A WAC and should have comparable cost.

7. Further Charge Reduction Opportunities

Since R-290, a A3 class refrigerant, is highly flammable, it is necessary to seek all possible opportunities to reduce the charge. We achieved a measured EER, 17% higher than the ENERGY STAR rating. There is room to reduce the system charge at the expense of slightly decreasing the performance. Two additional ways are considered for the charge reduction: one is to minimize the liquid section, i.e. the subcooling degree in the condenser; and the other uses microchannel heat exchangers to replace the fin-and-tube heat exchangers.

Microchannel heat exchangers can achieve similar heat transfer effectiveness as fin-and-tube heat exchangers with less than half the internal refrigerant volume. They have great potential in charge reduction.

However, they are not used for small capacity air conditioners and heat pumps, because the cost effectiveness can't compete with conventional fin-and-tube coils, when using HFC and HCFC refrigerants. Secondly, use of a microchannel evaporator may cause refrigerant mass flow mal-distribution due to two-phase refrigerant entering hundreds of mini-ports, and the tube and fin structure may impede the condense water removal. These issues still need further investigations. On the other hand, for a R-290 system, it is more important to seek charge reduction than lower the cost of the heat exchangers.

We conducted simulations using microchannel heat exchangers to replace the fin-and-tube condenser and evaporator, to realize the potential charge reduction. Table 7 describes geometrical details of the microchannel heat exchangers.

Table 7: Microchannel Condenser and Evaporator of Window Air Conditioner

Parameters	Microchannel Condenser	Microchannel Evaporator
Face area [ft^2]	1.44	0.94
Number of rows	1	1
Tube Thickness [in]	0.01	0.01
Finned tube length [in]	19.0	16.0
Tube pitch [in]	0.375	0.375
Flat tube width [in]	1.0	1.0
Fin density [fins/in]	20	22
Fin length [in]	0.287	0.287
Fin depth [in]	1.0	1.0
Micro-port shape	Square	Square
Port hydraulic diameter [in]	0.0315	0.0315
Number of ports	24	24
Total Tube Number	30	24
Number of tubes in condenser	24	N/A
Number of tubes in subcooler	6	N/A

We simulated three combinations of heat exchangers with the improved R-290 compressor, which are; 1) the baseline fin-and-tube heat exchangers (FTCs); 2) replacing the fin-and-tube condenser using the microchannel condenser (MHXCond); and 3) replacing both the condenser and evaporator using the microchannel heat exchangers (MHXs). It should be noted that we assumed the same fan flow rates and power consumption, and the same evaporator exit superheat degree for the comparison.

Figure 16 compares predicted system charges of the three heat exchanger combinations under AHRI A condition. Using all microchannel heat exchanger decreases the charge level to 150 grams. Only using a microchannel condenser reduces the charge moderately because the baseline WAC already uses 5-mm tubes for the fin-and-tube condenser. Figure 17 presents the cooling EER changing with the subcooling degree. Using the microchannel condenser decreases the EER to around 13.0 and using both the microchannel heat exchangers degrades the EER to 12.3. These values are still above the ENERGY STAR rating. Figure 18 illustrates the cooling capacity. Using the microchannel condenser reduces the cooling capacity by about 4%, because the limited microchannel condenser elevates the condensing pressure and the enthalpy entering the evaporator. Using both the microchannel heat exchangers further decreases the capacity by 10%, because the restricted microchannel evaporator lowers the compressor suction pressure and leads to a smaller refrigerant mass flow rate.

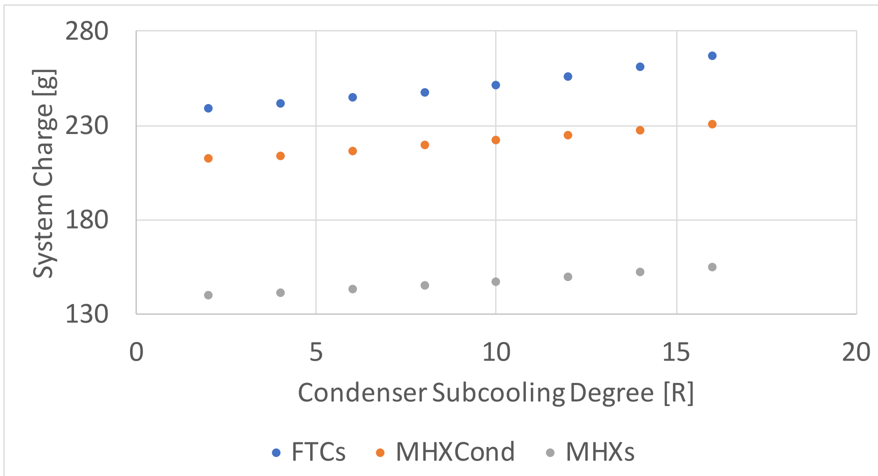

Figure 16: Predicted system charge of R-290 as a function of the condenser subcooling degree

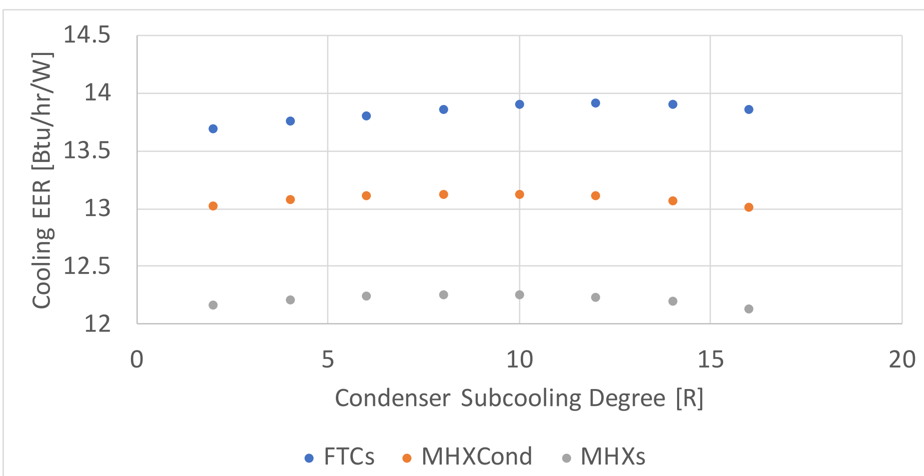

Figure 17: Predicted cooling EER as a function of the condenser subcooling degree

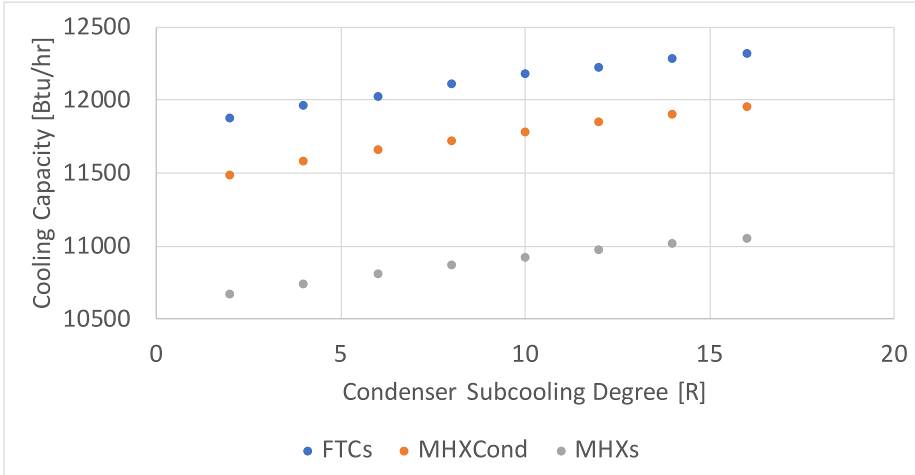

Figure 18: Predicted cooling capacity as a function of the condenser subcooling degree

8. Conclusions

Propane (R-290) is a very efficient refrigerant and promising for small packaged AC/HP systems

- According to U.S. EIA, 2009, there were 46.7 million window air conditioners (WAC) in the United States, accounting for 1.5% of the total U.S. residential energy use or about 0.33 quads.
- Propane is a much less expensive, but more efficient refrigerants than R-22 and R-410A.
- Propane has negligible environmental impact.
- Propane is highly flammable, so, system charge must meet safety regulation.

We started with an ENERGY STAR WAC as the baseline, i.e. GE AEC12AV originally using R-410A. The unit uses a slinger wheel to spray condensate water on the condenser surface, a condenser having 5 mm tubes, and an evaporator having 7 mm tubes.

We modified the unit by replacing the compressor and expansion devices optimized for R-290. Shanghai Highly Electrical Appliances Co. Ltd, a business of Johnson Controls, is our CRADA partner, helped develop a high efficiency propane rotary compressor, which could reach isentropic efficiency up to 80% at the target working conditions.

By experimental verification in our environmental chambers, the improved WAC can achieve the project goals:

1) 260 g charge limit proposed by EPA.
2) Same rated cooling capacity as the baseline WAC, and 17% higher EER, i.e. 14.0 EER.
3) Operated up to 131°F ambient temperature for Mideast, high ambient applications.

Additionally, we conducted system simulations using HPDM to investigate further charge reduction opportunities. Utilizing microchannel heat exchangers to replace the fin-and-tube coils, the charge can be decreased to 150 grams. Use of the microchannel heat exchangers at the reduced heat exchanger volumes causes capacity and efficiency degradation. However, it can still achieve a rated EER > 12.0 and cooling capacity > 10,000 Btu/hr at the AHRI A condition, meeting the ENERGY STAR requirements. That implies that the use of a high efficiency propane compressor makes up the loss due to reducing the heat exchanger volume and meet the high efficiency standard with satisfying the safety (system charge) regulation.

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