United Technologies Research Center

Final Technical Report

Natural Refrigerant Very High Efficiency Residential AC System

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Executive Summary

United Technologies Research Center (UTRC) in collaboration with the University of Illinois – Urbana Champaign (UIUC) proposed in 2009 to design, develop and demonstrate a safe, very low global warming potential (GWP) very high performance air-conditioning (VHPAC) residential system. The proposed residential ducted system was targeted to reduce system direct GWP by a factor of approximately 200, and had the potential to achieve a net 4% reduction in overall annual energy consumption of residential buildings.

The primary objective was to develop a highly integrated system that not only enables the use of low direct GWP refrigerant, but also provides a significant improvement in system efficiency resulting in reduced annual energy consumption, and thus, reduced indirect GWP. Specifically, the goal of the project was to design, develop and demonstrate a safe low GWP VHPAC 3TR residential ducted system that provides: a 30% reduction in annual energy consumption; approximately 200 times reduction in direct GWP; and a 50% reduction in primary refrigerant charge relative to a current 18 SEER/13.4 EER 2-speed R410A AC system

By end of 2012, UTRC successfully designed, developed, and demonstrated (TRL 2 to 5 in < 24 months) first of a kind safe and highly efficient natural refrigerant-based residential air-conditioning system that can provide >350x direct global warming potential reduction, 31.6% annual energy savings, >77% charge reduction and 25% outdoor unit volume reduction as compared to the high efficiency 18SEER R410A 2-speed baseline system. During the course of project research and development, two prototype systems were fabricated from the state-of-art components and >650hr of safe and reliable operation was demonstrated in UTRC laboratory through system level testing under AHRI standard conditions.

During the course of the project, close coordination with UTC Climate Controls and Security (CCS) engineering and supply-chain teams was maintained and potential cost impact of the proposed system for residential HVAC&R market was established. The accomplished project deliverables supported both DOE and CCS mission to define a practical, safe, and strategic technical path in using low GWP natural refrigerant.

1. Contract Objective and Project Management Plan

Goal - In response to DE-FOA-0000115 and in support of energy, environmental, and economic goals expressed in the Energy Policy Act of 2005, Executive Order 13423, and the Energy Independence and Security Act of 2007, the United Technologies Research Center (UTRC) in collaboration with the University of Illinois – Urbana Champaign (UIUC) shall design, develop and demonstrate a natural refrigerant based very high efficiency residential AC system that provides: a 30% reduction in annual energy consumption; approximately 200 times reduction in direct GWP; and a 50% reduction in primary refrigerant charge relative to a current 18 SEER/13.4 EER 2-speed R410A air-conditioning system.

This project was initiated on Aug. 15th 2010. A detailed Project Management Plan including the risk and mitigation strategy, project milestones, costing profile, and timeline was submitted to DOE in Sept. 2010.

In Oct. 2010, project kick-off meeting was held with DOE Contracting Officer's Technical Representative and with key nanofluid researchers from University of Illinois as sub-contractor. Thereafter, several internal and external project reviews were conducted to go over the details of component/system design, testing, and analyses. On Feb. 25th 2013, the final project review was successfully held at Washington D.C. with key DOE BTO customers.

By the end of this project, the team was able to successfully design, fabricate, and test (> 650hrs) two 3 TR air-conditioning system prototypes using natural refrigerant. The second prototype system with integrated power electronics board prototype (PEB1) and controls successfully demonstrated

- Direct GWP reduction: >350x (Target: 200x)
- Energy saving: 31.6% (Target: 30%)
- Charge reduction: >77% (Target: 50%)
- Size reduction in outdoor unit: 25% (Target: 0%)

compared with an existing 18SEER R410A 2-speed unit as baseline.

Overall, the project spending was kept on track. The final status of the detailed project milestones is highlighted in Table 1.1.

Table 1.1 Project milestone status (2010-2013).

Milestones	Due Date	Status
Kick–off meeting	Nov 14, 2010	Completed
Quarterly report delivered	Oct. 30, 2010	Completed
Quarterly report delivered	Jan. 30, 2011	Completed
Nanofluid screening conducted (UIUC)	Feb.15, 2011	Not used
Component design and analysis completed	Mar.31, 2011	Completed
Quarterly report delivered	Apr.30, 2011	Completed
Component-level controls developed with steady state models	Jun. 30, 2011	Completed

Component test rig fabricated and commissioned	Jul. 31, 2011	Completed
VSD test rig fabricated and commissioned	Jul. 31, 2011	Completed
Component-level control algorithm implemented	Aug.31, 2011	Completed
Quarterly report delivered	Jul. 31, 2011	Completed
Annual meeting to review progress	Sep.30, 2011	Completed
Quarterly report delivered	Oct. 30, 2011	Completed
Quarterly report delivered	Jan. 31, 2012	Completed
Thermal-hydraulic performance of nanofluid tested and analyzed (UIUC)	Jan.30, 2012	Not used
Preparation of nanofluid for component and system level testing supported (UIUC)	Mar.31, 2012	Not used
Natural refrigerant mechanical sub-system demonstrated	Mar.31, 2012	Completed
Quarterly report delivered	Apr. 30, 2012	Completed
Design of experiments for system testing created	May31, 2012	Completed
1st prototype system rig designed and fabricated with VSD system and control algorithm implemented	Jun. 15, 2012	Completed
Quarterly report delivered	Jul. 30, 2012	Completed
1st prototype system testing completed	Aug.31, 2012	Completed
2nd prototype system rig designed and fabricated	Sep. 30, 2012	Completed
Annual meeting to review progress	Sep.30, 2012	Completed
Quarterly report delivered	Oct. 30, 2012	Completed
2nd prototype system testing completed	Dec.15, 2012	Completed
System testing data analyzed	Dec.31, 2012	Completed
Quarterly report delivered	Jan. 30, 2013	Completed
Final analysis completed	Feb. 28, 2013	Completed
Final report delivered	Mar.31, 2013	Completed
Meeting to review overall progress	Mar.31, 2013	Completed

2. Component Design and Test Rig Commissioning

2.1. Thermal system design and test rig commissioning

In order to achieve performance and cost competitive goals when using the natural refrigerant, a rigorous effort was undertaken to select the SEER18 R410A baseline configuration, collect the variable-speed system design requirements, develop system design models, identify the potential suppliers for the key components, and fabricate/commission the test rigs. Each of the above mentioned activities will be discussed in the following sub-sections.

Baseline system selection

An existing Carrier® 3TR Performance Series mainstream AC product was selected as the 18 SEER/13.4 EER 2-speed R410A baseline system. The selection of unit with the cooling capacity of 3TR was based on its common use and the consequent potential impact on the residential building energy consumption. Table 2.1 lists a few detailed system specifications of the selected baseline system with 3TR unit highlighted in red box.

The cooling mode testing conditions for two-speed system are listed in Table 2.2. The calculation of the annual energy consumption of 2-speed AC system was based on the fraction of the total temperature bin hours, as listed in Table 2.3.

Table 2.1 Specifications of the selected Carrier® 18 SEER 2-speed R410A 3 TR baseline system.

PHYSICAL DATA

3 Ton Unit

UNIT SIZE – VOLTAGE, SERIES	24–30	36–30	48–31	60–31
Operating Weight lb (kg)	204 (93)	204 (93)	306 (139)	316 (143)
Shipping Weight lb (kg)	243 (110)	243 (110)	363 (165)	373 (169)
Compressor Type	2–Stage Scroll			
REFRIGERANT	Puron (R–410A)			
Control	TXV (Puron Hard Shutoff)			
Charge lb (kg)	6.63 (3.01)	6.88 (3.12)	11.63 (5.27)	15.13 (6.86)
COND FAN	Propeller Type, Direct Drive			
Air Discharge	Vertical			
Air Qty (CFM)	3100	3400	4300	4450
Motor HP	1/10	1/5	1/4	1/4
Motor RPM	800	800	800	800
COND COIL				
Face Area (Sq ft)	21.56	21.56	25.15	30.18
Fins per In.	25	25	20	20
Rows	1	1	2	2
Circuits	4	4	7	8
VALVE CONNECT. (In. ID)				
Vapor	3/4	7/8	7/8	7/8
Liquid	3/8			
REFRIGERANT TUBES (In. OD)				
Rated Vapor*	3/4	7/8	1–1/8	1–1/8
Liquid	3/8			

*Units are rated with 25 ft (7.6 m) of lineset length. See *Vapor Line Sizing and Cooling Capacity Loss* table when using other sizes and lengths of lineset.

Table 2.2 AHRI 210/240 2008 version standard test conditions for two-speed air-conditioning system [1].

ANSI/AHRI STANDARD 210/240-2008

Table 7. Cooling Mode Test Conditions for Units Having a Two-Capacity Compressor

Test Description	Air Entering Indoor Unit Temperature				Air Entering Outdoor Unit Temperature				Compressor Capacity	Cooling Air Volume Rate
	Dry-Bulb		Wet-Bulb		Dry-Bulb		Wet-Bulb			
	°F	°C	°F	°C	°F	°C	°F	°C		
A_2 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	95.0	35.0	75.0[1]	23.9[1]	High	Cooling Full-Load[2]
B_2 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	82.0	27.8	65.0[1]	18.3[1]	High	Cooling Full-Load[2]
B_1 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	82.0	27.8	65.0[1]	18.3[1]	Low	Cooling Minimum[3]
C_2 Test – optional (steady, dry-coil)	80.0	26.7	[4]		82.0	27.8	–		High	Cooling Full-Load[2]
D_2 Test – optional (cyclic, dry-coil)	80.0	26.7	[4]		82.0	27.8	–		High	[5]
C_1 Test – optional (steady, dry-coil)	80.0	26.7	[4]		82.0	27.8	–		Low	Cooling Minimum[3]
D_1 Test – optional (cyclic, dry-coil)	80.0	26.7	[4]		82.0	27.8	–		Low	[6]
F_1 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	67.0	19.4	53.5[1]	11.9[1]	Low	Cooling Minimum[3]

Table 2.3 Distribution of fractional hours within cooling season temperature bins in AHRI standard 210/240 [1].

Bin Number, j	Bin Temperature Range °F	Representative Temperature for bin °F	Fraction of Total Temperature Bin Hours, N_j/N
1	65-69	67	0.214
2	70-74	72	0.231
3	75-79	77	0.216
4	80-84	82	0.161
5	85-89	87	0.104
6	90-94	92	0.052
7	95-99	97	0.018
8	100-104	102	0.004

<u>*Variable-speed system design/testing conditions*</u>

The testing conditions defined in the AHRI standard 210/240 for the variable speed product were selected for the evaluation of the studied natural refrigerant system. The seven AHRI testing conditions with the outdoor ambient temperatures ranging from 67°F to 95°F have been highlighted in Table 2.4. The calculation of the annual energy consumption of variable-speed AC system was based on the same temperature bins listed in Table 2.3.

Table 2.4 AHRI 210/240 2008 version standard test conditions for variable-speed air-conditioning system [1].

ANSI/AHRI STANDARD 210/240-2008

Table 9. Cooling Mode Test Conditions for Units Having a Variable-Speed Compressor

Test Description	Air Entering Indoor Unit Temperature				Air Entering Outdoor Unit Temperature				Compressor Speed	Cooling Air Volume Rate
	Dry-Bulb		Wet-Bulb		Dry-Bulb		Wet-Bulb			
	°F	°C	°F	°C	°F	°C	°F	°C		
A_2 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	95.0	35.0	75.0[1]	23.9[1]	Maximum	Cooling Full-load [2]
B_2 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	82.0	27.8	65.0[1]	18.3[1]	Maximum	Cooling Full-load [2]
E_V Test - required (steady, wet coil)	80.0	26.7	67.0	19.4	87.0	30.6	69.0[1]	20.6[1]	Intermediate	Cooling Intermediate[3]
B_1 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	82.0	27.8	65.0[1]	18.3[1]	Minimum	Cooling Minimum[4]
F_1 Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	67.0	19.4	53.5[1]	11.9[1]	Minimum	Cooling Minimum[4]
G_1 Test - optional (steady, dry coil)	80.0	26.7	[5]		67.0	19.4	–		Minimum	Cooling Minimum[4]
I_1 Test - optional (cyclic, dry coil)	80.0	26.7	[5]		67.0	19.4	–		Minimum	[6]

<u>*System modeling*</u>

A steady-state system model of the proposed 3TR AC system was constructed using UTC proprietary modeling tool. The system modeling indicated that the cyclic degradation coefficient (C_D) plays an important role in the system energy calculation. As stated in AHRI Standard 210/240, the cyclic degradation coefficient (C_D) can be either determined by conducting two optional dry-coil tests or assigned with the default value of 0.25 when the tests have not been conducted. At the component design stage, the default value of C_D =0.25 was used for the proposed system. In order to compare the energy consumption on the same basis, the cyclic degradation coefficient (C_D) of the two-speed R410A baseline was also set to the default value of 0.25. During the actual prototype system performance analysis, both the default and actual cyclic degradation coefficients were evaluated.

With the default value of 0.25 used for C_D, the system simulation results indicated that the proposed cooling system can achieve the targeted 30% reduction in the annual energy consumption under some pre-defined assumptions. A dynamic model of the proposed 3TR AC system was further developed utilizing UTC's proprietary dynamic thermal fluid libraries. This dynamic system model was verified against the steady-state model results at the rated conditions as shown in Tables 2.5 and 2.6. The differences between the steady-state model and the dynamic model were well within ± 10%, which is acceptable for the control design. In order to support the sub-system test rig set up and operation, an additional dynamic model without the work recovery device was also developed.

Table 2.5 Comparison of the steady-state model results and the dynamic model results at AHRI 95 ^{o}F outdoor ambient condition.

	Key Parameters		
	Condenser Capacity (W)	Evaporator Capacity (W)	COP
Steady-State Model	12260	10157	4.16
Dynamic Model	11750	9693	4.04
Error (%)	-4.2%	-4.6%	-2.9%

Table 2.6 Comparison of the steady-state model results and the dynamic model results at AHRI 82 ^{o}F outdoor ambient condition.

	Key Parameters		
	Condenser Capacity (W)	Evaporator Capacity (W)	COP
Steady-State Model	12398	10545	4.88
Dynamic Model	11774	10005	4.72
Error (%)	-5%	-5.1%	-3.3%

Key component design and sub-system test rig

Significant efforts were focused on identifying state-of-the-art but commercially available key components and suppliers. Multiple compact heat exchanger suppliers and compressor manufacturers were closely engaged to develop high performance prototypes. Existing UTRC work recovery capability was also utilized to test work recovery component under the AHRI rated conditions. Based on the component testing results, the work recovery device design was dropped for this project application, due to its small impact on the overall system performance (versus the associated increases in cost and refrigerant charge). However, it needs to be noted that the work recovery device performance depends on both the working fluid and the operating condition and there exists potential to increase system efficiencies at high ambient conditions (e.g. 95 ^{o}F and above).

The sub-system test rig, shown in Fig. 2.1, was designed and fabricated using the key prototype components. Instrumentation (temperature, pressure, flow, and power sensors) and DAQ (Data Acquisition) were designed, procured and calibrated. Appropriate power supplies and watt transducers were installed and commissioned with support from UTRC facility engineering.

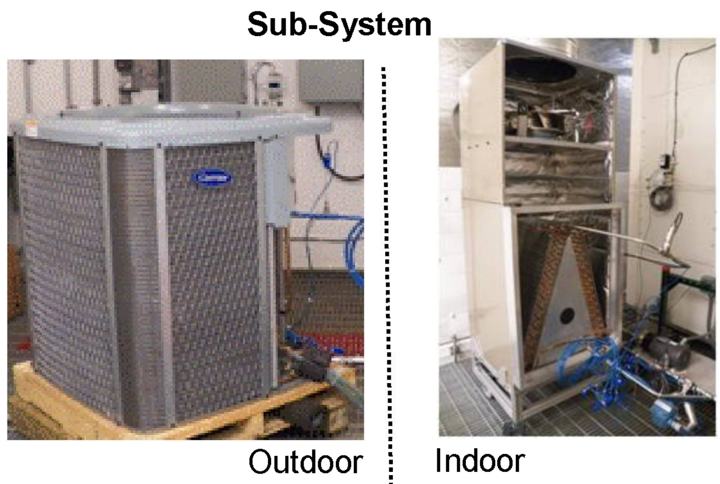

Figure 2.1 Fabricated sub-system.

The sub-system test rig was successfully leak checked via both high pressure and vacuum approaches. The commissioning of the sub-system test rig was completed under partial load and full load conditions. Overall energy balance of <5% was achieved based on the evaporator capacity measurements as shown in Table 2.7. In order to compare the evaporator capacity energy balance, a MicroMotion flow sensor was used to measure the refrigerant mass flow and an air flow code tester (based on ASHRAE standard 41) was used to measure the air volumetric flow. The air-side heat transfer was kept at dry conditions during the energy balance check.

Table 2.7 Energy balance analysis based on evaporator capacity measurement.

Outdoor Ambient	Refrigerant side	Air side	Diff.
67 ^{o}F	2759 W	2898 W	5%
87 ^{o}F	5904 W	6009 W	1.8%
95 ^{o}F	10253 W	10525 W	2.7%

During the initial testing in late 2011, it was observed that there was up to 13psi parasitic pressure loss in the compressor suction line, which caused extra power consumption in the compressor. In early 2012, several improvements were made in the suction pipe and sensor locations. The suction pressure loss was successfully reduced to around 2 psi (85% reduction) in the modified thermal sub-system rig. In order to handle the natural refrigerant testing, a new psychrometric testing chamber was constructed with UTRC internal capital funding excluding UTRC cost share funding provided to the project. After the new psychrometric testing facility was commissioned, the outdoor unit of the sub-system was relocated into the new testing chamber with the indoor/outdoor connection pipe lengths of around 25 feet. The improved thermal sub-system rig was then successfully commissioned with natural refrigerant.

2.2. Power Electronics (PE) component design

The power electronic system for the proposed system consists mainly of motors, power electronic converters for controlling the power flow in the motor drive or the utility, and the intelligence to control the system. In this project, the variable speed drive (VSD) operation of the main compressor was realized for a high efficiency three phase permanent magnet (PM) type motor while taking power from a single-phase ac source at high power factor. To achieve such performance, a single-phase interleaved Power Factor Correction (PFC) unit was integrated with a three-phase inverter. Advanced sensor-less control techniques were implemented for controlling the PM motor. Suitable control techniques were developed for the power factor correction unit.

The power electronics system level design was focused on two areas: (i) design configuration for the PE system, and (ii) choices of the circuit topology for the power factor correction (PFC) unit.

(i) Design of suitable configuration for the PE system

The design of the power electronic system was realized on a printed circuit board (PCB). The main motor drive unit for the compressor motor with the power factor correction unit and the electro-magnetic interface (EMI) filter was integrated on a single printed circuit board, labeled PEB1 (Power Electronic Board 1).

(ii) Choices of the circuit topology for the power factor correction (PFC) unit. Various components of the motor drive system were powered from a single phase utility supply (220V, 60Hz). The total maximum power that would be drawn from the utility was about 4kW. In order to meet the desired harmonic levels while achieving high efficiency and low cost, appropriate PFC circuit topology was selected. Furthermore, suitable control techniques had to be deployed for achieving the best performance of the system. Studies were conducted to select the circuit topology for the single-phase PFC unit, suitable for the estimated power rating of the HVAC system. Two circuit topologies were down-selected for final consideration. The first topology was an interleaved double boost converter with single-phase rectifier. The second topology was based on a dual boost converter configuration. Though both topologies had the potential to achieve high efficiency, a thorough comparison was conducted to down-select the option that fit the overall system requirement in terms of both performance and cost.

2.3. Nanofluid design

Development of the nanofluid for this project was lead by University of Illinois at Urbana-Champaign (UIUC). UIUC literature search showed that a number of researchers have studied modification in thermal properties, convective heat transfer and pool-boiling heat transfer of nanofluids [2]. Analytical approach modifying Maxwell's theory of conduction through suspended particles to account for particle shape, distribution, concentration, particle-shell structure and contact resistance has yielded quite a few equations. Recently a comprehensive study for water-based nanofluids reproduced thermal conductivity experimental data by classical effective medium theory for well dispersed particles [3]. While majority of the data available in the literature have shown enhancement in thermal conductivity and convective heat transfer of nanofluids, enhancement in pool-boiling heat transfer are not as consistent. The experimental data on flow-boiling heat transfer of nanofluids are limited and show enhancements [4-6]. No experimental data for natural refrigerant based nanofluids are available in literature.

To investigate thermal properties of refrigerant-based nanofluids, a pressure vessel apparatus integrated with viscometer and thermal conductivity sensors was designed. The viscometer and thermal conductivity sensors were capable of measurement in the range from 0.1 to 500 mPa-s (cP) and 0.01 to 2 W/m-K, respectively. The relative accuracy on both sensors was ±1% and the temperature and pressure range was from -30°C to 150°C and 0 to 7 MPa, respectively. Thermal conductivity sensor was also capable of measuring thermal diffusivity, enabling calculation of specific heat of the nanofluid. The pressure vessel was also equipped with a magnetic stirrer drive for nanofluid stirring, heating/cooling coil for variation in temperature, and two glass windows for visualization.

A closed re-circulating loop was used to measure flow-boiling heat transfer coefficient (HTC) of nanofluids. Nanofluid was circulated using a gear pump (Micropump Inc.) coupled with a 0.33 HP TENV motor (Baldor Inc.). A VS1ST series variable frequency drive, VFD (Baldor Inc.) was used to control motor rpm, thus controlling the mass flow rate of nanofluid in the loop. A coriolis type ELITE series flow sensor (Micro Motion Inc.) was used along with 2700 series transmitter to measure the mass flow rate and density. The test section was made up of smooth copper tubing, 6.23 mm ID and 1.8 m long. Thermocouples were attached at seven different locations on the test section dividing it in six sub-sections, each 250 mm long. The sub-sections were centered on the test section. Two thermocouples, one at top and other at bottom, were placed at each of the seven cross sections. Thermocouple beads were attached to the surface of the tube into small slots using epoxy. The slots were 1 x 0.5 x 0.5 mm in dimension. Two Omegalux rope heaters (Omega Inc.), 5 mm in diameter and 3.05 m in length, were used as heat source for the test section. Each rope heater was rated at 500 watts with 240 volts voltage input. Power input to the rope heaters was supplied using N5771A dc power supply (Agilent technologies) rated at 1500 watts. The test section along with heat source was used as evaporator in the closed loop. A schematic of the test section is shown in Fig. 2.2. A similar configuration to that of evaporator test section was used as preheating section before the test section. No surface thermocouples were used in preheating section. A compact plate heat exchanger was used as condenser in the loop. The condenser exchanged heat with ethylene glycol solution circulated from laboratory chiller system. A turbine flow meter and transmitter (Flow Technology) was used to measure the mass flow rate of the glycol solution. Sheath thermocouples were used to measure fluid temperature in the chiller loop at the inlet and exit of the condenser. A sample cylinder (Swagelok), one liter in internal volume, was used as a receiver in the closed loop. The flow circulated through exit of the pump, flow sensor, preheating section, evaporator test section, condenser, and receiver back to the pump inlet. Temperature of the fluid was measured using sheath thermocouples at the inlet and exit of preheating section, inlet and exit of the evaporator test section, inlet and exit of the condenser, and exit of the receiver.

Absolute pressure in the system was measured at the inlet and exit of the evaporator test section and that of condenser. A bypass was used in between exit of the pump and inlet of the receiver and operated using a ball valve to establish very low mass flow rates in the closed loop. Data from flow sensors, dc power sources, pressure transducers, and thermocouples were recorded using current and thermocouple cDAQ modules (National Instruments).

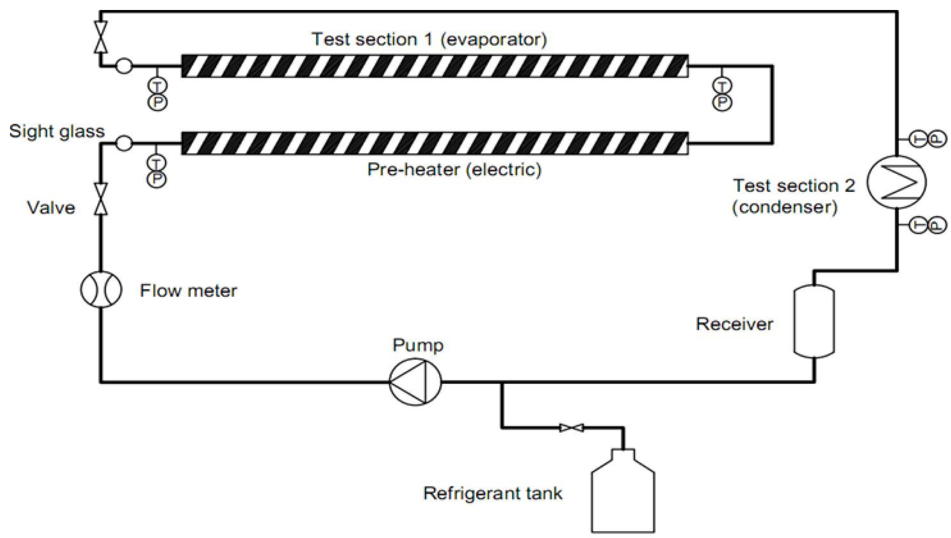

Figure 2.2 Schematic of the flow-boiling apparatus.

3. Component Testing and Analysis

3.1.Thermal sub-system test rig testing

The functionality testing of the natural refrigerant thermal sub-system test rig (without the power electronics prototype and controls) was successfully conducted at the UTRC Psychrometric Facility in Q2 2012. The thermal sub-system showed stable performance over the AHRI outdoor temperature range of 67^{o}F-95^{o}F with the compressor speed manually adjusted. Step change tests were also conducted to provide system transient behavior data for control development. After the completion of sub-system functionality tests, the thermal sub-system was further integrated with the power electronics board prototype (PEB1) and implemented with the developed controls.

3.2. Power Electronics (PE) component testing

Several tests were conducted with the power electronics prototype before its final integration with the rest of the system. The details are provided as below:

Functionality testing of PFC and sensor-less controls

The initial testing of the control algorithms for Power Factor Correction (PFC) and sensor-less control of permanent magnet motor drive (PMMD) was carried out on the first Power Electronics Board (PEB1) prototype. Closed loop control of the power factor correction unit was implemented with a resistive load connected at the DC bus of PEB1. The sensor-less control algorithm was developed for permanent magnet motors and successfully tested under no-load condition.

To eliminate the rotor position sensor, and hence to reduce the component count and cost of the system, sensor-less control techniques for the motor drive were evaluated. Preliminary tests for the sensor-less technique were carried out under no-load condition. The dynamic responses of the inner loop d-axis and q-axis current controller are presented in Fig. 3.1(a). The successful sensor-less operation of the motor is shown in Fig 3.1(b), where the speed reference was increased by a positive ramp and then decreased by a negative ramp.

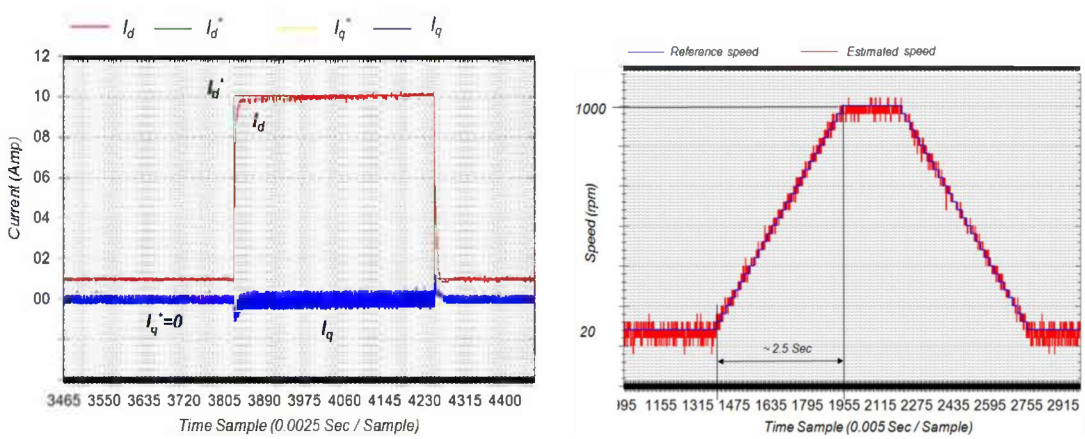

Figure 3.1 Sensor-less control: (a) current control, (b) sensor-less control speed tracking capability.

Performance testing of packaged PFC and sensor-less controls

The Power Factor Correction (PFC) and motor control algorithms were further developed and tested in the first power electronics board (PEB1) prototype at the rated power. The algorithms were then integrated and tested at full load. A state machine was added to the embedded software to allow for auto start-up and stand alone operation, while fine tuning of the state machine was carried out during the fully integrated variable speed system testing.

Signal interfacing hardware was developed to provide communication capability between the PEB1 local controller and the LabVIEW interface, which operated the variable speed thermal sub-system prototype. PEB1 was successfully integrated with the rest of the prototype system and supported the compressor smooth operation during the entire range of operation.

Performance testing of PEB1 integrated with thermal system

The Power Electronic Board (PEB1), containing a single-phase Factor Correction (PFC) unit and a three-phase motor drive unit, was integrated with the control board. The integrated system along with suitable auxiliary components was assembled in a metallic enclosure. With proper power outlets, the packaged PE sub-system was successfully commissioned with the thermal system. Furthermore, successful communications between the local controller of the PE sub-system and the LabVIEW controller of the thermal sub-system were established to control the PEB1, and hence to operate the compressor motor.

Additional tuning of the sensor-less motor control was carried out to address the issue of instability under the high speed range operation. The designed PE sub-system was able to successfully follow the input signals, start and stop command from the LabView controller.

3.3. Nanofluid testing

A number of nanofluid thermal property and flow boiling tests were conducted at UIUC to fundamentally understand the potential of nanofluid thermal performance improvement. It was observed that the stable dispersion of nanoparticles into natural refrigerant was very challenging. During the execution of this project, the nanofluid research effort conducted at UIUC did not lead to any promising nanoparticle candidate for the base fluid of interest. Nevertheless, UIUC is continuing to independently work on understanding underlying physics that could lead to stable suspensions and heat transfer improvements with the use of enhanced nanoparticle-based fluids. It is suggested that an extensive study of multiple parameters such as particle material, size, mass fraction, surface treatment, and base oil shall be conducted and their effect on flow boiling HTC shall be analyzed. This could help investigate novel physics behind nanofluids in addition to the use of best possible nanofluid in various energy efficient systems.

4. System Design, Testing and Analysis

4.1. 1st prototype system testing and analysis

The first prototype system was assembled by integrating the thermal sub-system with the variable speed power electronics board prototype. Testing of the prototype system with the key components was conducted with the natural refrigerant under the AHRI rated conditions. As stated in the AHRI 210/240 standard, cyclic degradation coefficient (C_D) of the new system can either be experimentally determined through G2/I1 tests of Table 2.4 or assigned with the default value of 0.25. For the DOE proposal, a default C_D value (=0.25) was used. When C_D is set as 0.25 (default value), the normalized annual energy consumption is estimated to be 821 W.hr/(Cooling season hr) and 633 W.hr/(Cooling season hr) for the baseline system and the first prototype system, respectively. As a result, the first prototype system demonstrated 22.9% annual energy saving with C_D =0.25.

4.2. 2nd prototype system testing and analysis

Based on the findings of the first prototype system, a second prototype system was fabricated with design improvements in outdoor air flow distribution, heat exchangers, and fans. Compared with the baseline system, the first and the second prototype systems had 25% reduction in the outdoor unit volume. Also, more than 77% reduction in the refrigerant charge was demonstrated with the second prototype due to the use of compact heat exchanger designs. Figure 4.2 shows the 1st and 2nd outdoor prototype units placed next to the R410A baseline unit. Experimental data show that the second prototype system demonstrated 31.6% annual energy saving compared with the R410A two-speed baseline (with default C_D = 0.25).

It should be noted that the definition of C_D depends on the type of AC system (e.g. single-speed, two-speed, or variable speed). Therefore, the impact of C_D value on each type of system will be different. It would be inappropriate to directly compare the C_D values between systems of different type. In this report, the C_D is primarily used as a key parameter to determine the annual energy consumption of each system.

Over the course of the prototype system testing, more than 100 tests were conducted to thoroughly investigate the impacts of various design factors (e.g. system charge, control parameters, heat exchanger geometry, and fan selection) on the overall system performance. Close attention was also paid to the refrigerant to air energy balance to ensure the high quality of test data. Figure 4.2 shows the energy balance between the refrigerant and the indoor air to validate the cooling capacity measurement of the 2nd prototype system. For most cases, the energy balance was kept within ± 5% error range. For some cases with high loads (~10kW), the energy balance was relaxed to ± 10% error range, as the uncertainties in the latent heat transfer played an important role in the total cooling. As shown in Table 4.1, the propagation of uncertainty from humidity measurement to cooling capacity calculation became inevitably significant.

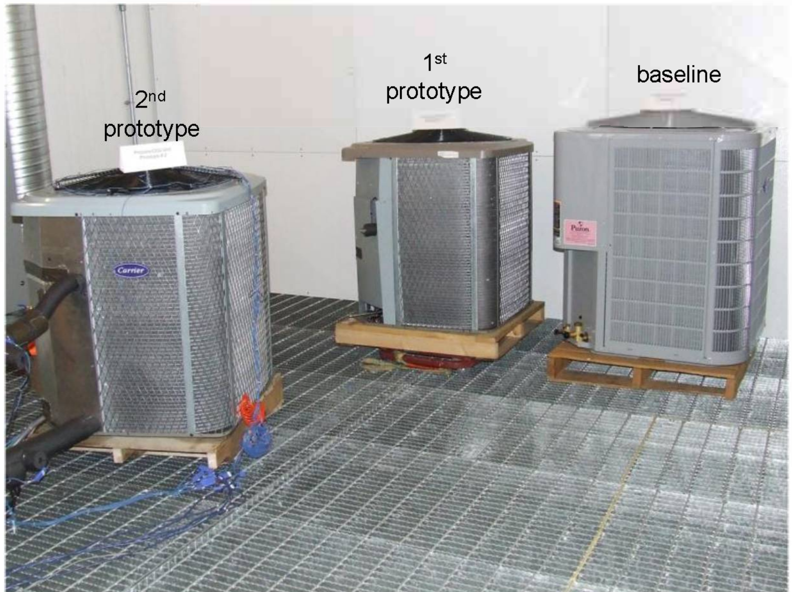

Figure 4.1 Size comparison of the 1st and the 2nd prototype outdoor units with the baseline outdoor unit.

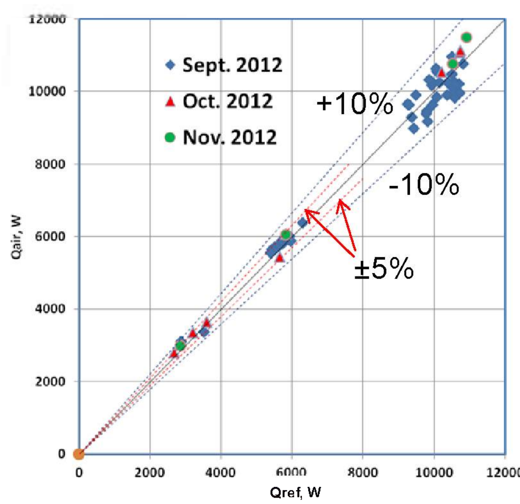

Figure 4.2 Energy balance between refrigerant and indoor air for the 2nd prototype system testing.

Table 4.1 Sensor and cooling capacity uncertainties.

	Parameters	Sensor Uncertainty	Capacity Uncertainty
Evaporator Air Side	Tin	±0.2F	±0.9%
	Tout	±0.2F	±0.9%
	Tdew_in	±0.36F	±2.3%
	Tdew_out	±0.36F	±2.0%
	V_air	±0.4%	±0.4%
Propane Side	T_cond	±0.2F	±0.2%
	T_suct	±0.2F	±0.1%
	P_cond	±0.75psi	~0%
	P_suct	±0.5psi	±0.05%
	m_dot	±0.25%	±0.25%

5. Final Analysis and Summary

The overall performance of the developed system prototypes is highlighted in Table 5.1. As shown, all the key performance deliverables set forth at the beginning of the project were successfully met/exceeded. Additionally more than 650hr of safe operation was demonstrated in UTRC laboratory through system level testing under AHRI standard conditions.

During the course of the project, close coordination with CCS engineering and supply-chain teams was maintained and potential cost impact of the proposed system for residential HVAC&R market was established. The accomplished project deliverables supported both DOE and CCS mission to define a strategic, safe and a practical technical path in using low GWP natural refrigerant.

Table 5.1 Performance summary of prototyped system.

Summary	DOE BTO Target	Demonstrated
Direct GWP Reduction*	200x	>350x
Annual Energy Savings*	30%	31.6%
Outdoor Unit Size Reduction*	0%	25%
Primary Charge Reduction*	>50%	>77%
* compared with the existing high efficiency 18SEER 2-speed R410A AC baseline		

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