Annex VII

Promoting Low-GWP Refrigerants for Air-Conditioning Sectors in High Ambient temperature Countries Phase II (PRAHA-II)

2019

Project Report

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UNITED NATIONS ENVIRONMENT - UNEP

UNITED NATIONS INDUSTRIAL DEVELOPMENT ORGNIZATION - UNIDO

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Table of Contents

Disclaimer ... 2

List of Tables ... 6

List of Figures ... 6

ACRONYMS ... 7

Executive Summary ... 8

1. Background and Project Main Elements ... 10

2. Capacity Building through Knowledge Sharing and developing a Technical Platform ... 14

2.1. Track-I: Capacity Building for the Manufacturers of PRAHA-I ... 14

2.2. Track-II: Sharing with the different stakeholders at regional and global events ... 17

2.3. Conclusion from the Capacity Building Element ... 18

3. Optimization of PRAHA-I Prototypes ... 19

3.1. Contracting the Activities ... 19

3.2. Scope of Work ... 20

3.3. Deliverables ... 21

3.4. Matrix ... 21

3.5. Project Monitoring ... 22

3.6. Challenges and Modifications ... 23

3.7. Project Implementation and Findings ... 23

3.7.1. Activity 1 – Analyzing the Design of PRAHA-I Prototypes ... 23

3.7.2. Activity 2 – Design Improvements ... 27

3.7.3. Activities 3, 4, and 5 ... 35

3.8. Conclusion and Recommendations from the Optimization Element ... 40

4. Risk Assessment ... 44

4.1. Background on Risk Assessment ... 44

4.1.1. Flammability Definition and Classes ... 44

4.1.2. Concept of Risk Assessment ... 45

4.1.3. Approach of a Risk Assessment Model ... 46

4.1.4. Outline of a Risk Assessment Model ... 47

4.1.5. Global Risk Assessment Efforts ... 48

4.2. Process of a Risk Assessment Model ... 50

4.3. Example of a Risk Assessment Model ... 58

4.3.1. Simulation of Time Factor and Space factor During Usage Stage 60
4.3.2. Simulation of Time Factor and Space factor During Servicing Stage 62
4.4. Conclusions and Recommendations from the Risk Assessment Element 63
5. Overall Conclusions and Recommendations 65
Annex I – Examples of Risk Assessment Model Data Tables filled 67
Annex II - Safety 69
ANNEX-III: AHRTI Final Report 71

List of Tables

Table 1: PRAHA Participation in International Events 17
Table 2: Matrix of Activities for the Prototype Optimization Element of PRAHA-II 22
Table 3: Matrix of units and new refrigerants to be tested....24
Table 4: Components for units 1, 4, 6, and 10....25
Table 5: Compressor models 31
Table 6: Activity 2 Results.....34
Table 7: Prototype Units for Component Modification and Further Testing 35
Table 8: Unit 6 Modifications for Testing. 37
Table 9: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 35°C. 37
Table 10: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 46°C. 37
Table 11: Unit 10 Modification for Tetsing....38
Table 12: Unit 10 - Performance Test Summary at 35°C. 38
Table 13 : Unit 10 -Performance test Summary at 46°C....39
Table 14: Unit 10 – R447B Leak Test Summary Results.....40
Table 15: Flammability Classification for Refrigerants 45
Table 16 Risk Matrix - Frequency vs. Severity (source JRAIA) 51
Table 17: Determination of tolerable risk levels 52
Table 18: Information for the risk assessment model used by PRAHA team 58
Table 19: Data for the calculation of risk for Usage Stage....60
Table 20: Data for calculation of Risk for Service Stage 62

List of Figures

Figure 1: Main findings from PRAHA-I 11
Figure 2: Outline of PRAHA-II....12
Figure 3: Heat Exchanger First Order Analysis (FOA)....26
Figure 4: P-h and T-s diagrams for HCFC-22 alternatives 28
Figure 5: P-h and T-s diagrams for R-410A alternatives 28
Figure 6: Profiles of refrigerants....29
Figure 7: Illustrative T-s diagram for baseline and improved cycle 30
Figure 8: Diagram illustrating COP Improvement a) Real Cycle, b) Ideal Cycle (Carnot) 30
Figure 9: System design framework 31
Figure 10. Condenser Forms: Unit 1 (left), Unit 10 (center), Unit 6 Cabinet (right). 33
Figure 11. HX Form Examples: L-shape (left), Flat (right). 33
Figure 12: Factors and Probability of Ignition....44
Figure 13: Tolerable vs. Acceptable risk (source: UL) 46
Figure 14: Procedure of Risk Evaluation according to ISO/IEC 51 (source: JRAIA) 48
Figure 15: Selection of equipment and life stage for the risk assessment model 50
Figure 16: Life Cycle range for assessment....52
Figure 17: Risk Map 53
Figure 18: Fault Tree Analysis (FTA) model 55
Figure 19: Probability of Ignition FTA 55
Figure 20: FTA reiterative process 57
Figure 21: FTA for usage stage....61
Figure 22: FTA for servicing stage....63

ACRONYMS

AC	Air Conditioning
AHRI	Air Conditioning, Heating, and Refrigeration Institute
AHRTI	Air Conditioning, Heating, and Refrigeration Technical Institute
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
BTU/hr	British Thermal Unit per Hour
CHEAA	China Home and Electrical Appliance Association
EGYPRA	Promotion of Low-GWP Refrigerants for the AC Industry in Egypt
ETA	Event Tree Analysis
EVX	Electronic Expansion Device
FMEA	Fault Measurement and Effects Analysis
FOA	First Order Analysis
FTA	Fault Tree Analysis
GWP	Global Warming Potential
HAT	High Ambient temperature
HC	Hydrocarbon
HCFC	Hydro Chloro Fluoro Carbon
HFC	Hydro Fluoro Carbon
HFO	Hydro Fluoro Olefin
HOC	Heat of Combustion
HX	Heat Exchanger
IPR	Intellectual Property Rights
JRAIA	Japan Refrigeration and Air Conditioning Industry Association
kW	Kilowatt
lbs	Pounds
LFL	Lower Flammability Limit
MCHX	Micro Channel Heat Exchanger
MOP	Meeting of the Parties
OEWG	Open-Ended Working Group
RACHP	Refrigeration, Air-Conditioning, and Heat Pumps
TFHX	Tube Fin Heat Exchanger
TXV	Thermal Expansion Valve
UA	Thermal Conductance
UFL	Upper Flammability Level

Executive Summary

PRAHA Has Turned into a Process!

PRAHA-I created an awareness about the challenges faced by high ambient temperature (HAT) countries and offered stakeholders in HAT countries support in building their technical knowledge of the alternatives technologies as well as practical support through the building and testing of several prototypes using lower-GWP refrigerants.

PRAHA-I concept of testing prototypes at high ambient temperatures pioneered other testing and research programs which eventually tested more alternative refrigerants than the few refrigerants that were still in the development stage when PRAHA-I was launched. In Addition, PRAHA-I also helped component manufacturers, especially compressors, to start building and testing dedicated compressors for the new alternative refrigerants that are capable of delivering sustained energy efficiency levels at HAT conditions.

The main result of PRAHA is that it went beyond the level of being an individual project with specific planned outcomes and outputs, PRAHA turned to be a **PROCESS** at different levels: governmental, local industry, institutional as well as for the international technology providers.

PRAHA-II is a continuation of the process with specific goals that are aligned with the findings of PRAHA-I. The two main findings of PRAHA-I are that, 1) there are viable alternatives at HAT conditions which need optimized equipment design to perform and deliver the energy efficiency minimum requirements, and 2) that there is a concern about safety of the mostly flammable alternative refrigerants that calls for a special risk assessment model for the HAT countries.

PRAHA-II Elements

PRAHA-II had three main elements: 1) to build the capacity of the local industry in designing and testing products using efficient lower-GWP flammable refrigerants; 2) to evaluate and optimize the prototype built for PRAHA-I; and 3) To build a risk assessment model for the high ambient temperature countries.

Each element has its components and events and was designed to give maximum exposure to the stakeholders, both the industry as well as research institutions and the government, on the latest technology and the developments that are happening worldwide. All three elements were designed to benefit the maximum number of stakeholders.

PRAHA-II Main Findings

PRAHA-II delivered tangible and beneficial results on all three main elements.

- **Capacity Building**: The capacity building element was successful in providing a platform of cooperation between governments, research institutes, industry associations, and the industry in general and became a process for the sharing of information and results among the different stakeholders. The experience of working on PRAHA-I gave UN Environment and UNIDO the confidence that international stakeholders support the goals of the project and that the outcome will be beneficial to all and beyond economic gains. Simultaneous to the efforts by the PRAHA project to create awareness about HAT challenges and the work done through the different symposia held in the HAT countries that were participating in the PRAHA project, the local industry themselves started to directly evaluate and examine long term alternatives which reflect the level of built awareness and attention gained to the wise selection of alternatives.

- **Design Optimization**: The original scope and schedule were modified during the project as new findings and challenges surfaced. The original baseline test data was used for comparison with tests done on the optimized units built according to the modeling work done even though the latter tests included measurements and metrics not typically performed in energy certification tests of the type done under PRAHA-I.
  A resume of the conclusions:
    - For systems operating in considerably higher temperatures (greater than 46°C), the resultant impact on performance must be considered since performance will degrade as compared to operating under more temperate conditions.
    - The design assessment through modeling provided good insights on adequate component design and/or selection for proper system functioning when using novel refrigerants;
    - Rebuilt and tested units exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This indicates the importance of proper compressor selection.
    - Because of the differences in saturation curves from the simulation analysis, refrigerant with wider saturation curves tend to result in systems with higher efficiency and less charge when no modifications to the hardware are made. The results showed however, that by making appropriate component selection, such as compressors with larger displacement volumes and higher mass flow rate, the cooling capacities and overall performance of the other refrigerants were of the same order of magnitude.
    - Refrigerant fractionation as evidenced by the leak tests, does not appear to be a great concern since less than 2% change in cooling capacity was observed after the system's re-charge.
- **Risk Assessment**: The work on risk assessment required resources beyond the traditional RACHP expertise that is allocated for typical conversion/demo projects. The different usage and servicing practices used in the region needed to be considered in order to assess the risk of using flammable refrigerants. The initial concern about the effect of high ambient temperature on the increased risk of ignition was removed and the main focus is on actual practices. The recommendation is for HAT countries to continue the risk assessment based on actual situations and reduce the risk by implementing various measures that are verified such as minimizing ignition probability. In addition, the risk assessments of other stages matching cultural and lifestyle aspects should be studied.

The Way Forward

In general, PRAHA-II outcomes will be of benefit to all 35 countries defined by the Montreal Protocol Parties at the OEWG-37, 2016 as "High Ambient Temperature Countries". A HAT symposium scheduled for March 2020 will convey these results to representatives from those countries. UN Environment and UNIDO intend to transform the PRAHA initiative into a live process with continuous feedback and support to HAT countries.

1. Background and Project Main Elements

Background

The 69th meeting of ExCom approved PRAHA-I with the aim to support assessing the feasibility of lower-GWP refrigerants suitable for high-ambient temperature countries and in particular for air-conditioning applications. UN Environment and UNIDO worked with local industries, international technology providers and national ozone units in these countries to do such assessment through an agreeable independent process that included in its core component building and testing 18 different prototypes and comparing them with respective baseline units which are available from the local industry using mainly HCFC and high-GWP HFC such as R-410A. The process of building and testing the prototypes was completed in 2015 and the final report was released in January 2016. PRAHA included additional components for assessing the technology transfer barriers, energy efficiency implications and economics of alternatives in addition to assessment of district cooling opportunities to reduce dependency on high-GWP alternatives and technologies.

The key finding of PRAHA-I show the potentials and challenges to promote the use of lower-GWP alternatives. Furthermore, many of the non-testing components under PRAHA, like assessing standards and codes and promoting technology transfer, were not thoroughly completed due to two main reasons; the commercial availability of the lower-GWP alternatives in the high-ambient markets and limited resources available to complete the work needed. The findings also pose important queries about what is left to be done in order to make the deployment of low-GWP alternatives possible at high-ambient temperature countries.

PRAHA-I Key Findings

The non-testing components under PRAHA-I assessed technological, economic and energy efficiency aspects in conjunction with high ambient temperature with the following key findings:

I. There are potential alternative refrigerants that are close, or in some cases better, in performance and efficiency compared to baseline refrigerants (HCFC_22 and R-410A) that are worth further investigation. With further product engineering (design and optimization) those alternatives can be strong candidates for replacement of HCFC-22;

II. There is a need to develop the R&D capacity of the local air-conditioning industry in high ambient temperature countries in terms of the design and optimization of products using lower-GWP alternatives with their specifics characteristics, such as flammability, higher operating pressures, temperature glide, etc.;

III. Economic and technology transfer barriers Intellectual Property Rights (IPR) will continue to be issues for some time before international and regional markets stabilize on a limited group of candidates that are sustainable compared to the current long list of options being examined;

IV. Due to the nature of those alternatives and the consequent safety issues, a comprehensive risk assessment model needs to be tailored to the needs of A5 countries, in particular for high ambient temperature conditions. Such a model needs to address manufacturing, placing into market, servicing and the end-of-life of the equipment;

V. There is a lack of institutional programs that address alternative technologies to reduce the dependency on high-GWP alternatives in high ambient temperature countries. This is clearly reflected by the market directions during the phase-out of HCFCs;

VI. The process of improving energy efficiency (EE) standards for air-conditioning application in high ambient temperature countries is progressing at a much quicker pace compared to the process of assessing and selecting alternative refrigerants. A smart approach is needed to jointly consider addressing EE and lower-GWP alternatives in order to avoid promoting higher-GWP alternatives that are commercially available at this stage of time.

Figure 1 summarizes the main findings from PRAHA-I.

Technical Feasibility of low-GWP Alternatives

- *Comparison to commercially existing options i.e. HCFCs & high-GWP HFCs*
- *Relation to Energy Effeceincy programs Directives*

Safety consideration and and Availbility of Materials

- *Risk assessment of A2L and A3 alternatives in industry and service*
- *Availability and cost of alternatives and components*
- *Technology transfer and IPR considerations*

Research and Development Capacities

- *Knowledge and capacity of industry to design with lower-GWP alternatives*
- *Related research programs at global, regional and national levels*

Contribution of District Cooling

- Size and potentiality of district cooling applications
- Use of lower-GWP refrigerants or not-in-kind technologies in district cooling applications

FIGURE 1: MAIN FINDINGS FROM PRAHA-I

The Project

UN Environment and UNIDO approached the Multilateral Fund seeking support for stage-II of PRAHA which is designed to address the priority areas identified in PRAHA-I. The Executive Committee of the Multilateral Fund of Montreal Protocol approved, in its 76th meeting in May 2016, stage-II of the project which is now called PRAHA-II.

The main objective of the project is to maintain the momentum generated by PRAHA-I and advance the technical capacities of stakeholders to enable the adoption and use of lower-GWP sustainable technologies for high ambient temperature countries by supporting the decision-making process related to the acceptance and promotion of lower-GWP refrigerants and advancing the technological capabilities of the local industry to design with those refrigerants.

In consultation with the project stakeholders, several areas were identified that would require further work in order to ensure putting the process of alternative refrigerants' deployment on the right track and address all technical, technological and economic concerns of both industry and policy makers. The areas identified and envisaged to be part of PRAHA-II fall under two components with three distinct elements as shown in **Error! Reference source not found.**. The three elements of PRAHA-II are detailed b elow.

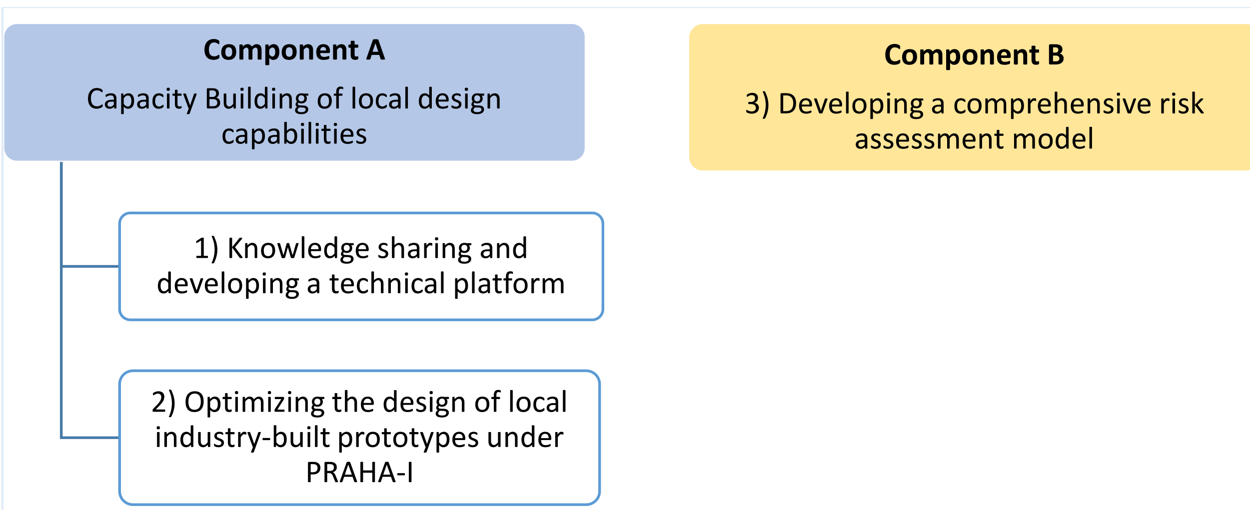


FIGURE 2: OUTLINE OF PRAHA-II

Under Component A, Capacity Building, there are two elements**:**

I. **Building the capacity of R&D sectors in designing with low-GWP alternatives through knowledge sharing and developing a technical platform**
There are three technology schools when it comes to design air-conditioning units, excluding chiller systems, with low-GWP alternatives:

- Designing with HFC-32, which is quite established by the Japanese industry;
- Designing with HC-290, which is at an acceptable level of maturity in China and in other countries;

- Designing with HFO/HFC blends which is just starting to be implemented in different places around the globe.

II. **Optimizing the designs of PRAHA-I prototypes to meet/exceed the baseline designs:** This includes several elements using prototypes of PRAHA-I that had good results and candidate refrigerants that are promising. Prototypes showing unexpectedly poor results will also be evaluated to identify shortcomings.

Component B aims at developing a comprehensive Risk Assessment Model: This includes designing, developing and examining a risk assessment model suitable for use pattern and operating conditions for high ambient conditions and in particular for the GCC region.

2. Capacity Building through Knowledge Sharing and developing a Technical Platform

The concept behind this element is to benefit from the experience of the most advanced industry for each technology in building the capacity of the local R&D in high ambient temperature countries. This includes attending special courses, workshops and conferences discussing these technologies, as well as field visits to manufacturing centers in countries pioneering the technologies.

The three centers of technology for the three main types of lower-GWP refrigerants are Japan, China, and the United States. The Japanese industry is leading in HFC-32 technology for the residential air conditioning sector (apart from Variable Refrigerant Flow -VRF) and is the most proliferated technology in terms of market penetration even though it does not have the lowest GWP. The Japanese market is fully transitioned into using this technology and all Japanese manufacturers are currently producing products using HFC-32. These companies, and other, are building HFC-32 products outside Japan and are marketing them in other markets including some of the HAT countries. The HFC-32 program was conducted in cooperation with Japan Refrigeration and Air Conditioning Industry Association (JRAIA) with input from the Japan Society of Refrigeration and Air Conditioning Engineers (JSRAE) and the industry.

The Chinese industry have an established HC-290 technology and have successfully implemented several conversion projects under UNIDO/UNEP. . Even though the products are not widely available globally, the potential for this technology is very promising due to the many advantages that HC-290 offers in terms of energy efficiency and low-GWP characteristics. The draw-back of high flammability was the main concentration of the capacity building efforts for the stakeholders. The HC-290 program was conducted in cooperation with the Chinese Household and Electrical Appliances Association (CHEAA) and the Chinese industry

The North American industry is leading in the field of unsaturated HFC technology, also referred to a Hydrofluoroolefin (HFO) technology. Although most of the lower-GWP HF0 refrigerants are still not widely available globally, test results have shown some promising alternatives with good performance. The HFO training program which was designed in cooperation with the Air Conditioning, Heating, and Refrigeration Institute (AHRI) which represents the industry in the US with involvement from and the technology providers, i.e., refrigerant and compressor manufacturers.

The capacity building efforts had two tracks: TRACK-I capacity building for the manufacturers of PRAHA-I, and TRACK-II knowledge sharing with different stakeholders at regional and global events

2.1. Track-I: Capacity Building for the Manufacturers of PRAHA-I

The objective of this track is to expose the manufacturers of PRAHA-I to the three technologies through factory visits, study tours, specialized courses, and special events. The purpose is to see firsthand how the technologies have developed in the three centers and how to apply them locally in terms of design, production capabilities, and after sales support.

This track included two study tours, one to Japan and one to China which also included events that were specially designed for PRAHA. In Japan, a risk assessment workshop to explain the Japanese model for A2L refrigerants and the data needed for building the model; and in China a special workshop on A3 refrigerants that attracted input from local and international resources and included participants from other HAT countries.

The HFC-32 study tour objective was to provide participants with a good background about designing and working with lower-GWP refrigerants with A2L low-flammability characteristics. The tour included plants visits, the risk assessment workshop, as well as attending the JRAIA International Symposium on "New Refrigerants and Environmental Technology".

The plant visits took place at Daikin facility in Shiga and the Mitsubishi plant in Shizuoka. Both plants produce HFC-32 based units and have been in production for a couple of years with hundreds of thousands of units installed. The plant visit included explanation of the charging and testing facilities where special precautions are needed. Participants were able to view the special measures taken for the safe handling of flammable refrigerants including storage.

The one-day workshop was conducted by JRAIA at their premises in Tokyo. The subject was risk assessment of A2L refrigerants for residential and commercial equipment. The information provided was detailed and included a review of the risk assessment work conducted by JRAIA; a presentation of key requirements for design; risk assessment for residential & commercial split type air conditioners and VRF during installation and maintenance; and safety guidelines during charging and servicing. Presenters were from Daikin, Panasonic, and Mitsubishi.

The symposium took place in Kobe Dec 1 & 2, 2016: The program provided in-depth information about global efforts to transition to lower-GWP refrigerants including research, regulation, design, safety, components, and energy conservation. The symposium also included a session on new refrigerants and their systems.

The HC-290 study tour objective was to provide in-depth knowledge of HC-290 with visit to a production plant, a building with an HC-290 installation, a special workshop, plus visit to China Refrigeration Expo to attend a one-day roundtable organized by UN Environment and other associations billed as Ozone2Climate Industry Roundtable (O2C).

The visit to AUX factory near Ningbo allowed the participants to view the special measures taken to manufacture equipment working with A3 flammable refrigerants. Factory personnel provided an overview of the R&D work and planning as well as sharing some information on the availability of products and their comparison to those operating with high-GWP refrigerants.

A visit to a facility with more than 1,100 units running on HC-290 was also arranged. The facility is a student dormitory for over 2,000 students in several buildings and all rooms are fitted with mini-splits running on HC-290. The units have been in operation for over two years and no incidents or major problems were reported. Participants were given a presentation by the management and maintenance staff and had the chance to interact with students and gauge their experience living in a facility with units running with an A3 refrigerant.

Workshop on Designing, Production and Installation with HC-290 in the Air Conditioning Industry was organized for PRAHA in collaboration with the Chinese association CHEAA and the Ozone authority of China, FECO. The workshop was enlarged to include other participants from China who joined the expanded PRAHA team. The expanded team included participants from Egypt, Tunisia and Vietnam. The agenda of the two-day workshop included presentations by research facilities, universities, and Refrigeration, Air Conditioning and Heat Pump (RACHP) component and equipment manufacturers.

The workshop focused on risk assessment and other measures related to hydrocarbons and HC-290 in particular. Presentations included a review of international standards and what is needed to enable the new flammable technologies to be adopted by the residential and commercial AC sector; conversion of a production line for the manufacturing of R290-based RAC equipment; and the performance of HC-290 in high ambient conditions. Other presentations discussed the installation and servicing of equipment with flammable refrigerants; reducing charge amount; and a review of R&D work by the manufacturers on A3 refrigerants.

The O2C Roundtable was organized by UN Environment, UNDP, FECO, and CHEAA and covered subjects on policy to promote alternative technologies, global trends, challenges and opportunities for the industry, and solutions for the cold chain and logistics. The PRAHA team presented the challenges in phasing out HCFCs in the countries with higher ambient temperature. Participants had the chance to visit the China Refrigeration Expo in Shanghai, one of the largest for RAC equipment.

The HFO experience in the United States included a course on "*New ASHRAE-Classified Refrigerants to Meet Society's Changing Needs*" by the ASHRAE Leaning Institute (ALI) was offered to several PRAHA stakeholders who were attending the ASHRAE conference and AHR expo. The course discussed the properties of refrigerants and the history of development of synthetic refrigerants and delved into a detailed discussion on flammability and the safe uses of refrigerants. International standards and agreements governing refrigerants and flammability were discussed.

The participants were also invited to a one-day workshop by the Climate and Clean Air Coalition to Reduce Short-Lived Climate Pollutants (CCAC) on *"Sustainable Technologies for Stationary Air Conditioning"* which aimed to familiarize participants with climate-friendly and cost-effective air conditioning technologies which have proven their applicability to replace high-GWP HFCs.

PRAHA-II team presented on "Challenges at High Ambient Temperature" with discussions on the effect of high ambient on the design and operation of air conditioning systems, energy efficiency of refrigerant alternatives, and safety when using flammable refrigerants. The presentation also included highlights from the four research projects testing low-GWP refrigerant alternatives at different temperatures and a comparison of the results. The presentation concluded with a brief description of the work done on PRAHA-II.

The key outcomes from this element of PRAHA-II were in providing information on risk assessment work for both A2L and A3 refrigerants; informing on the availability of new components and new products running with lower-GWP refrigerants; viewing of operating production lines handling A2l and A3 flammable refrigerant; experiencing an actual installation with more than 1,100 HC-290 units installed; and acquiring information form specially designed workshops, seminars, and courses.

2.2. Track-II: Sharing with the different stakeholders at regional and global events

PRAHA-II expanded beyond the original PRAHA-I participants. PRAHA started by inviting members from EGYPRA, the Egyptian Project for Testing Low-GWP Refrigerant Alternatives, to events and study tours. The addition of EGYPRA was a natural one as both projects have similar goals in testing alternative refrigerants on prototypes built by the local industry. EGYPRA participants joined the study tour to Japan in November 2016. The study tour to China in April 2017 was joined by participants from Tunisia and Vietnam; Pakistan was also invited but could not join.

The workshop in Japan was built for the PRAHA and EGYPRA participants. In China, the workshop included, other than EGYPRA, Tunisia, and Vietnam, many participants from China. It also included NGOs, and global researchers. There were close to a hundred participants and the workshop turned into a large forum on the research and development of A3 refrigerants.

Awareness building about HAT and the PRAHA project has been a constant element of PRAHA. The PRAHA-I final report lists the programs and the events which PRAHA launched or participated in. The HAT series of symposia is but one example of the awareness building achievements of PRAHA.

With PRAHA-II, the campaign continued with PRAHA taking advantage of the presence of its managers or consultants to continue the message and update stakeholders, the industry, and the Parties on the developments and the latest technological information related to HAT or to the research at HAT.

PRAHA appears in websites both by UN Environment and UNIDO. Some examples:

https://www.unido.org/our-focus/safeguarding-environment/implementation-multilateral-environmental-agreements/montreal-protocol/finding-climate-friendly-ways-cool-down

PRAHA has truly helped in spreading awareness on HAT challenges and opportunities. The continuous awareness of the challenges and the opportunities of the HAT regions has made HAT a permanent subject to be added to the Decisions of the Parties and is a part of every Task Force study and report. HAT now is a full chapter of the 2018 RTOC Assessment Report.

Table 1 shows events and functions where PRAHA either organized special/program in their margins, joined as keynote presentation or organized a dedicated event about the subject.

Table 1: PRAHA Participation in International Events

#	Date	Event
1	Jan 2016	Special Session at ASHRAE Winter Conference
2	Mar 2016	Special Session at West Asia/Africa Joint Network Meeting
3	July 2016	Special Session at OEWG-38
4	Aug 2016	Training Course at IIR Gustav Lorentzen Conference
5	Sept 2016	Special Session ASHRAE-AUB Efficient Building Design Conference
6	Dec 2016	Special Workshop on Designing with A2L Refrigerants

#	Date	Event
7	Jan 2017	ASHRAE Winter Conference and AHR expo
8	Jan 2017	CCAC Sustainable Technologies for Stationary AC Workshop
9	April 2017	Special Workshop on Designing with A3 Refrigerants
10	Oct 2017	International Workshop on Risk Assessment for HAT
11	Nov 2017	Special Session at CCAC Workshop at MOP-30 on Opportunities, Challenges, and Experiences with Transitioning to Low-GWP Alternatives
12	Jan 2018	Special Session at OzonAction First Interregional Networks' Meeting
	Oct 2018	Flammable Refrigerant Research and Planning Conference
13	Jan 2019	ASHRAE Winter Conference
14	Feb 2019	Special Session at OzonAction Second Interregional Networks' Meetings
15	March 2020 *(Planned)*	6th International Symposium on Alternative Refrigerants for High Ambient Temperature Countries

2.3. Conclusion from the Capacity Building Element

The experience of working on PRAHA gave UN Environment and UNIDO the confidence that international stakeholders support the goals of the project and that the outcome will be beneficial to all beyond economic gains. On the other hand, and simultaneous to the efforts by the PRAHA project to create awareness about HAT challenges and the work done through the different symposia held in the HAT countries that were participating in the PRAHA project, the HAT countries themselves were bringing up the issues at the different meetings of the Parties whether at the Open-Ended Working Group (OEWG) meetings or the Meeting of the Parties (MOP).

The capacity building element was successful in providing a platform of cooperation between governments, research institutes, industry associations, and the industry in general and became a process for the sharing of information and results among the different stakeholders.

3. Optimization of PRAHA-I Prototypes

This component includes several elements using prototypes of PRAHA-I that had promising results. Prototypes that showed unexpectedly poor results will also be examined to identify shortcomings. The exercise includes mainly three stages of work on the prototypes, plus a leak analysis stage:

a. Analyzing the design of PRAHA-I prototypes: a physical inspection and analysis of prior experimental results, plus a first order assessment of component and refrigerant performance.
b. Design optimization of PRAHA-I prototypes including: acquiring performance maps for components (compressors, fans) that are more suitable for the application; evaluating alternate heat exchanger design configurations; performing detailed engineering optimization to match or exceed the baseline unit performance within an acceptable design space set forth by an expert committee. This may include installing new upgraded compressors, for same refrigerants used in PRAHA-I, and which were not available at the time PRAHA-I prototypes were built; or compressors for refrigerants not tested under PRAHA-I; if so required.

c. Testing new refrigerants emerging since PRAHA-I using prototypes of PRAHA-I with change/upgrade of compressors.

d. Analyzing leak-recharge effect on performance for high glide alternatives.

3.1. Contracting the Activities

PRAHA first contact was with Oak Ridge National Laboratory (ORNL) who had performed their own testing at HAT conditions on two units with two different baseline refrigerants.

Unfortunately, due to legality issues and differences in the contractual practices commonly followed by UNEP, the contract between UNEP and ORNL did not materialize in spite of several attempts to find out solutions.

PRAHA team managed to negotiate and contract with The Air Conditioning, Heating and Refrigeration Technology Institute (AHRTI), the research arm of (AHRI) to take over the task as an internationally independent institute with relevant experience in conducting similar work i.e. AREP project (Alternative Refrigerants Evaluation Programme) and having access to several reputable testing and research centers within North America where the prototypes from PRAHA-I were being stored since end of PRAHA-I project. AHRTI, finally, selected Optimized Thermal Systems (OTS) as the most capable and sound research center for completing the planned work within the required timeline and budget.

3.2. Scope of Work

The scope of work that is covered by AHRTI's contractor OTS includes five activities as follows:

Activity 1: Analyzing the Design of PRAHA-I Prototypes

This task involved the following:

- Physical inspection
- Prior experimental results assessment
- First order assessment of component and refrigerant performance
- Development of validated model
- Detailed assessment of why the performance is "good, i.e. as designed" or "bad, why it did not perform as designed"

Activity 2: Design Optimization

Design optimization study for select units using the heat pump design model for available prototype units. This entailed:

- Acquiring performance maps for components (compressors, fans) that are more suitable for the application
- Evaluating alternate heat exchanger design configurations
- Performing detailed engineering optimization to match or exceed the baseline unit performance within acceptable design space set forth by an expert committee. This may include installing new upgraded compressors, for same refrigerants used in PRAHA-I that were not available at the time PRAHA-I prototypes were built; or compressors for refrigerants not tested under PRAHA-I; if so required.

Activity 3: Prototype Units Fabrication

AHRTI, in coordination with UN Environment, selected a subset of prototype units and modify them as per the design optimization study. This involved heat exchanger modification, compressor replacement, expansion valve fine-tuning, fans and blower replacements, etc. All components were from standard production lines.

Activity 4: Evaluation of the Optimized Prototypes

Optimized prototypes were tested in the multi-zone environmental chamber to evaluate their performance according to ASHRAE Standard 37 at relevant indoor and outdoor conditions (AHRI 210/240 "A" condition, ISO 5151 "T3" condition, hot and extreme conditions)

Activity 5: Analyzing Leak-Recharge Effect for High Glide Alternatives

The impact of leak-recharge effect on the performance of alternative refrigerants with high glide was experimentally evaluated.

Activity 6: Reporting and Data Management

AHRI submitted a peer-reviewed project report prepared by OTS.

3.3. Deliverables

The key deliverables/results to be achieved are:

a) Evaluation of prototypes tested under PRAHA-I
b) Optimized PRAHA-I prototypes: three units chosen
c) Analysis of leak-recharge of high glide alternatives on system performance
d) Report summarizing the project findings.

3.4. Matrix

The work to be done is shown in the matrix Table 2. The work is in five phases:

- Evaluation of the prototypes;
- Optimization of selected prototypes;
- Building some of the units per the optimized design;
- Testing for a number of refrigerants;
- Leakage assessment.

The selection of units for the various activities as well as that of the refrigerants was done the PRAHA team in coordination with the AHRTI based on:

- For Activity 1, all units needed to be evaluated.
- For Activity 2 for the modeling activity of optimization, the team chose one unit from each application, i.e. window, decorative split, and ducted split. An extra decorative split unit running with HC-290 was also added since decorative splits are the most abundant in the market and the team felt it important to have two splits optimized, one with HC-290 and one with alternatives to R-410A. The team also tried to balance the refrigerants choosing both alternatives to HCFC-22 as well as R-410A. At the time of selection, there was no clear trend or indication from the industry as to which refrigerants would be commercialized. One of the refrigerants originally selected had to be dropped at the request of the supplier.
- For Activities 3 & 4, the window unit with HC-290 was chosen to be re-built and tested. These activities for the window unit had to be dropped for reasons mentioned under **Challenges and Modifications**. For the decorative and ducted splits units 6 and 10, the team chose to work with the same refrigerant alternatives as in Activity 2. Activities 3 and 4 finally worked on one decorative split (unit 6) and one ducted split (unit 10).
- For Activity 4, leak analysis, all the zeotropic blends used in activities 3 and 4 were planned to be tested.

For the unit numbering system, units 1 to 3 are window units, units 4 to 9 are decorative splits and units 10 to 12 are ducted splits.

TABLE 2: MATRIX OF ACTIVITIES FOR THE PROTOTYPE OPTIMIZATION ELEMENT OF PRAHA-II

Unit	Type	Activity 1 Phase I data analysis	Activity 2 Optimization	Activity 3 Build per optimization	Activity 4 Test per build	Activity 5 Leak analysis
1	Window	L-20 (R-444B)	R-444B			
			R-454C			
			R-290	HC-290*	HC-290*	
			R-457A			
6	Decorative Split	HFC-32	HFC-32	HFC-32	HFC-32	
			R-454B	R-454B	R-454B	R-454B
10	Ducted	HCC-32	R-447B	R-447B	R-447B	R-447B
			R-452B	R-452B	R-452B	R-452B
4	Split	HC-290	HC-290			
2	Window	R-444B				
3	Window	DR-3 (R-454C)				
5	Split	HFC-32				
7	Split	L-41 (R-447A)				
8	Split	R-444B				
9	Split	R-454C				
11	Ducted	R-444B				
12	Ducted	R-454C				

* Could not be completed due to 1) not fitting the timeline, and 2) the limitation of testing A3 packaged (window)

3.5. Project Monitoring

AHRTI assembled a project committee made up of AHRI members to help monitor and guide the project and set-up biweekly conference calls with OTS and the PRAHA management team. The calls, which started in November 2018, are normally held on the first and third Thursday of every month. As part of the bi-weekly update, OTS reports both on the progress as well as the technical aspects of the project and solving any possible problems that may arise

On such example is the participation of an additional refrigerant supplier in the project through the supply of information and quantities of refrigerant R-459A to test in one of the optimized and rebuilt prototypes. The problem of receiving response from the supplier was raised in one of the calls and the supplier was contacted by the PRAHA team. The supplier advised of its inability to provide R-459A timely and asked to withdraw from the project. R-459A was replaced by R-454B which has been gaining acceptance by the industry lately.

3.6. Challenges and Modifications

The implementation of this portion of the PRAHA-II project came up with some challenges:

The tests that were carried out for PRAHA-I, while sufficient for the purpose of measuring capacity and energy efficiency for the purposes of PRAHA-I, did not have enough essential data to enable a complete cycle evaluation for optimization purposes.

Some key components and specifications, such as compressors and/or compressor maps for HC-290 and heat exchangers, were not readily available to fit in the project timeline.

The scheduling mechanism of the lab for PRAHA I (fixed test window) and testing logistics was not suited for completing of the project within the budge and required timeline. Therefore equipment performance testing was carried out in-house at OTS facility; however, its lab was not equipped to test the window unit of unit 1 working with A3 flammable refrigerant HC-290 (propane) due to safety concerns and requirements. Testing Unit 1 had to be dropped. Alternatively, the optimization of window unit was carried out using modeling approach.

Overall, the analyses presented by the design assessment through modeling provided good insights on adequate component design and/or selection for proper system functioning when using novel refrigerants. The tests in activities 3-5 partially served as validation for the models developed, and as check for previous test data from PRAHA I.

3.7. Project Implementation and Findings

The full AHRTI report is an annex to this report. The summary of findings per activity are given below

3.7.1. Activity 1 – Analyzing the Design of PRAHA-I Prototypes

Activity 1 was comprised of three major tasks including: a) reception of 12 physical units at the OTS facility followed by visual inspection and parts identification; b) review of performance test reports from PRAHA I tests; and c), analysis of data and identification, for units of interest, opportunities for improvement targeting higher performance and minimal charge.

The twelve units are shown in Table 3 with the PRAHA-I test results and the new refrigerants to be used.

Table 3: Matrix of units and new refrigerants to be tested

Category	Unit #	Ref.	Designed Capacity Btu/h	Measured Cap. Btu/h	Voltage	Ref. (New designs)	Ref. (Tests)
Window	**1**	**L-20 (R-444B)**	**18,000**	**19,104**	**208-230/60/1**	**R-444B, R-454C, HC-290, R-457A**	**HC-290**
	2	L-20 (R-444B)	18,000	16,924	208-230/60/1		
	3	DR-3 (R-454C)	18,000	18,063	208-230/60/1		
Decorative splits	**4**	**HC-290**	**24000 (18,000)**	**19,000**	**208-230/60/1**	**HC-290**	**HC-290**
	5	HFC-32	24000 (18,000)	19,328	208-230/60/1		
	6	**HFC-32**	**24,000**	**25,456**	**208-230/60/1**	**HFC-32, R-454B**	**HFC-32, R-454B**
	7	L-41 (R-447A)	24,000	24,830	208-230/60/1		
	8	L-20 (R-444B)	24,000	22,740	208-230/60/1		
	9	R-454C	24,000	14,638	208-230/60/1		
Ducted splits	**10**	**HFC-32**	**36,000**	**35,500**	**220-240/50/1**	**R-447B, R-452B**	**R-447B, R-452B**
	11	R-444B	36,000	36,553	220-240/50/1		
	12	DR-3 (R-454C)	36,000	33,032	220-240/50/1		

Following is a summary of findings from Activity I

A. Analysis of PRAHA-I Test Results:

- **For the window units:** *Evaporator:* The inlet refrigerant temperature and pressure were not measured. The outlet pressure was estimated from suction pressure, a reasonable assumption given the short distance between the evaporator and compressor. The outlet temperature was measured so the superheat was computed. *Condenser:* The inlet refrigerant temperature and pressure were measured. The outlet pressure was not measured, but the outlet temperature was measured.
- **For the decorative splits:** *Evaporator:* The "Inlet Pressure" is the value measured at the service port at the exit of the outdoor unit, after the expansion device (capillary tubes). So, there is significant, but unmeasured pressure and saturation temperature drop between the measurement location and the actual inlet of the evaporator as abovementioned. The "Outlet Pressure" was measured at the service port before entering the outdoor unit. There was an unmeasured pressure drop in the suction line from the evaporator outlet to that measurement location. The inlet and outlet temperature measurements seem like reasonable numbers for the actual inlet and outlet. *Condenser:* The inlet pressure was not measured, the inlet temperature was measured, and the outlet pressure was only measured for Unit 4. The outlet liquid temperature was not measured, rather, the "OD Liq" temperature measurement was likely taken at the liquid service port, near the pressure

measurement. The temperature was much too low to be the actual condenser outlet, but not cold enough to be the evaporator inlet.

- **For the ducted splits**: *Evaporator:* The "Inlet Liquid" temperatures and pressures were taken before the TXV, so they were not actual measurements of the evaporator inlet condition. The outlet temperature and pressure measurements were available so the superheat could be calculated (lab used the compressor suction temperature rather than evaporator outlet temperature to compute superheat.) *Condenser:* The inlet temperature was measured, but the pressure was not. The outlet temperature and pressure were measured, so the sub-cooling was calculated. The sub-cooling computed by the lab ranged between 17 to 18°F, which doesn't correspond to the measured conditions. The calculated sub-cooling for Unit 11, however, was negative for all three tests; as such, it is possible that there was a two-phase refrigerant at the condenser outlet.

B. Hardware Improvement Assessment

This section defines a first order analysis of the effect of hardware assessment for units 1, 4, 6, and 10. A first order analysis is structural analysis that is performed without taking the unit apart or making any changes to. The analysis is made for the different components.

Unit Component Modification Potential

Table 4 shows the detailed existing components for the units of interest for modification.

TABLE 4: COMPONENTS FOR UNITS 1, 4, 6, AND 10

System	Unit 1	Unit 4	Unit 6	Unit 10
Refrigerant	**R444B**	**R290**	**R32**	**R32**
Compressor	HIGHLY SL260DG-C8EU	HIGHLY PSH356DG-C8DU3	GMCC KSG226N1UMT	Copeland ZP42K5E-PFJ-XXX
Condenser	5mm Louver TFHX	9.5mm Wavy TFHX	7mm Louver TFHX	9.5mm Louver TFHX
Expansion Device	Capillary Tube	Capillary Tube	Capillary Tube	Capillary Tube
Evaporator	9.5mm Louver TFHX	7mm Louver TFHX	7mm Slit TFHX	9.5mm Louver TFHX

- **Heat Exchangers (HX):** OTS put as an objective to improve performance while minimizing charge. One way of addressing both objectives is by reducing the tube/channel diameter since heat transfer coefficients are inversely proportional to tube diameters. Pressure drop is also inversely proportional to tube diameter so smaller tubes result in reduced size and reduced internal volume but higher pressure drop.

  A qualitative analysis using values from literature was carried out to demonstrate the relative impact of diameter over abovementioned metrics, specifically: heat transfer coefficient, compactness and overall thermal conductance (UA). The left-hand side plot in Figure 3 shows three curves inversely proportional to the diameter; a 5mm tube can achieve, in this example, 70% greater UA than a conventional 9.5mm, within the same cabinet.

These are further explored to illustrate the impact on a system level. Systems respond to UA of both condenser and evaporators, but for the purposes of this analysis, the condenser is only considered. UA represents the overall thermal conductance, which will impact the approach temperatures in the system (ΔT_{app}). If the heat of rejection is kept constant, the higher the UA, the smaller are the ΔT_{app}'s, thus allowing the condenser to operate in lower pressure levels, which will consequently increase the system performance. An example using a hypothetical HFC-32 cycle with an EER of 12 as base is shown in the right-hand side plot in Figure 3. Performance improvement is limited by the Second Law, when the approach temperatures near zero. In this illustration, the EER has the potential to increase by over 20% with better condenser design alone.

It is imperative to note that the results presented in this section are first order analysis for illustration purposes only. Further in this report it is presented in more detail a re-design framework, applied to the units of interest in this project, using the metrics outlined in this section.

Unit 1 already had a 5mm condenser, which limits the options for HX re-design. Unit 6 had a 7mm HX on both the indoor and outdoor units, which allows some room for improvement if reducing to 5mm. Lastly, both coils for Unit 10 had 9.5mm tubes, thus there is greater potential for charge reduction and performance improvement for that unit in particular.

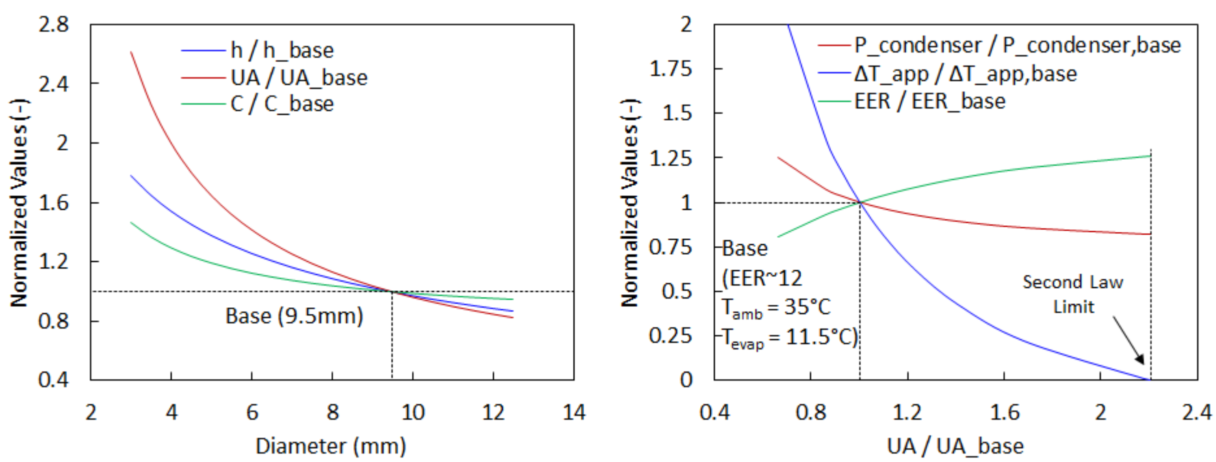


FIGURE 3: HEAT EXCHANGER FIRST ORDER ANALYSIS (FOA)

- **Compressors:** The existing units mostly use compressors sized specifically for R-410A or HCFC-22 and in some cases custom made for the particular application. This presents an opportunity for a better compressor selection when migrating to other refrigerants such as R454B or R447B on Units 6 and 10, respectively. A compressor designed for a particular refrigerant having a higher efficiency rating will result in better energy efficiency performance of the same unit.
- **Expansion Devices:** Expansion devices such as TXV's and EXV's may allow for better control and reduced losses in connecting pipes if located near the evaporator. Some units, such as 6 and 10, have a capillary tube in the outdoor unit, which forces the refrigerant to travel in two-phase

along the connecting pipes, and at lower temperatures, thus increasing pressure drop and heat gain. In some regions, expansion devices are installed in the outdoor units for noise control purposes.

- **Fans and Blowers:** Replacing the fan and blower may be necessary if newly designed HXs offer considerable change in pressure drop over the baseline since the flow rates are kept constant. The lack of test data on pressure drop forces us to rely on predicted values only.

3.7.2. Activity 2 – Design Improvements

OTS developed improved designs for some units, including use with additional refrigerants. The main goals were to maintain capacity while minimizing internal volume (refrigerant charge) and maximizing performance (COP). The exercise in optimizing the improved designs is subject to limitation in component availability from pre-established vendors. The activity involved:

- Developing a cycle simulation model for each of the baseline systems.
- Calibrating the models using the data provided in Activity 1 (relying on the performance test data for the three ambient conditions).
- For each system, evaluating whether the existing compressor and fans are the best fit, or if alternate designs would be preferred.
- Evaluating heat exchanger design options and suppliers for alternative off-the-shelf solutions. As appropriate, conduct a thorough parametric analysis study for the air-to-refrigerant heat exchangers for use with the alternative refrigerants. In addition to heat exchanger type and/or tube diameter and fin pattern, this may include revised circuitry.
- For each of the targeted design cases/refrigerants, evaluating the performance of optimum component selections and quantifying any anticipated performance gains.

Following is a summary of findings from modeling and simulation:

A. Hardware: A first order analysis in Activity 1 showed that moving towards smaller hydraulic diameter tubes can be beneficial from a charge reduction standpoint. Units 4 and 10 use conventional 9.5mm diameter tube condensers making them good candidates for condenser replacement with either a smaller tube diameter or a microchannel heat exchanger (MCHX). The compressors used on Units 1, 4 and 6 do not have available performance maps making it difficult to assess their fitness for the system. The focus of this study is on proper compressor selection and condenser re-design.

B. Refrigerant: HC-290 and HFC-32 have wider saturation regions, as can be seen from Figure 4 and Figure 5 for P/h and T/s, putting them at an advantage since they may operate with smaller superheat and sub-cooling, while benefiting from two-phase heat transfer. Their cycles may get closer to that of the ideal Carnot cycle compared to refrigerants with narrower saturation. Although this appears to be the case, this is not universally true for mixtures since they can exhibit other properties that make them suitable for certain designs.

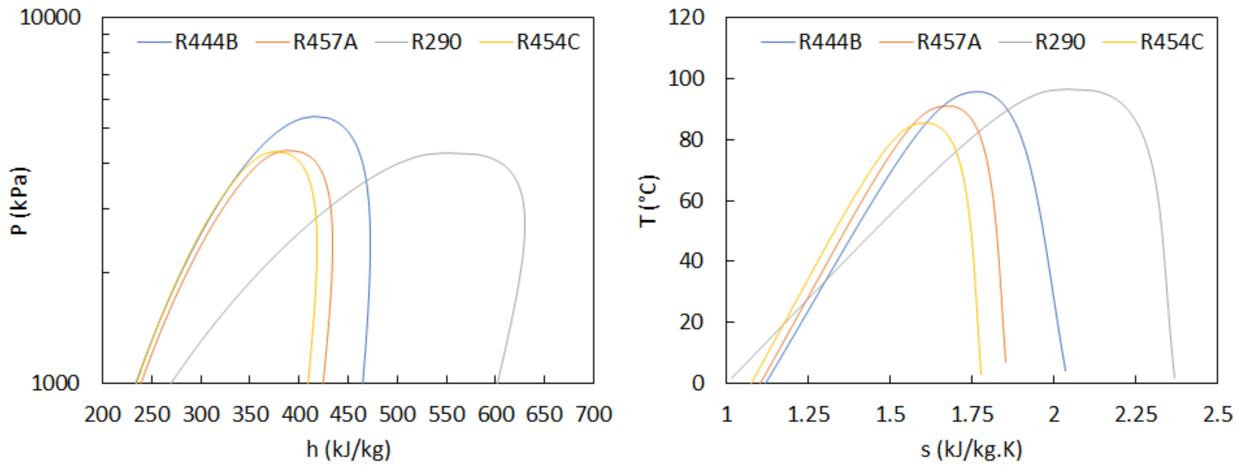


FIGURE 4: P-H AND T-S DIAGRAMS FOR HCFC-22 ALTERNATIVES

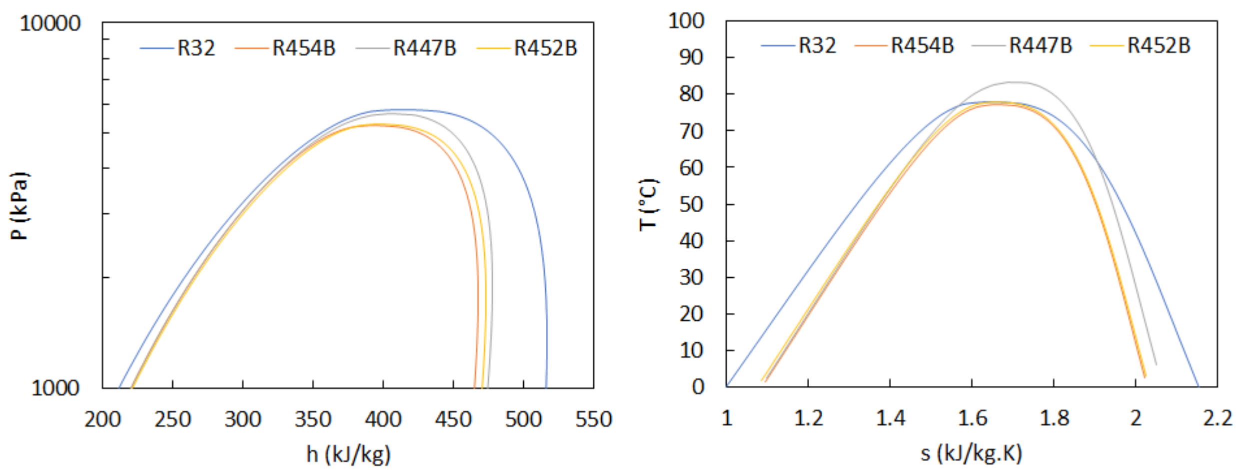


FIGURE 5: P-H AND T-S DIAGRAMS FOR R-410A ALTERNATIVES

Amongst the blends investigated for Unit 1, R-444B has the widest saturation region while also having the highest temperature glide Figure 6 .The latter is typically not beneficial, in particular for evaporators, but it may help the condenser. The glide enables the refrigerant temperature profile to get closer to the air temperature profile without crossing (Figure 6). From a thermodynamic perspective, this means R-444B can have its condensing pressure reduced further, resulting in higher theoretical COP.

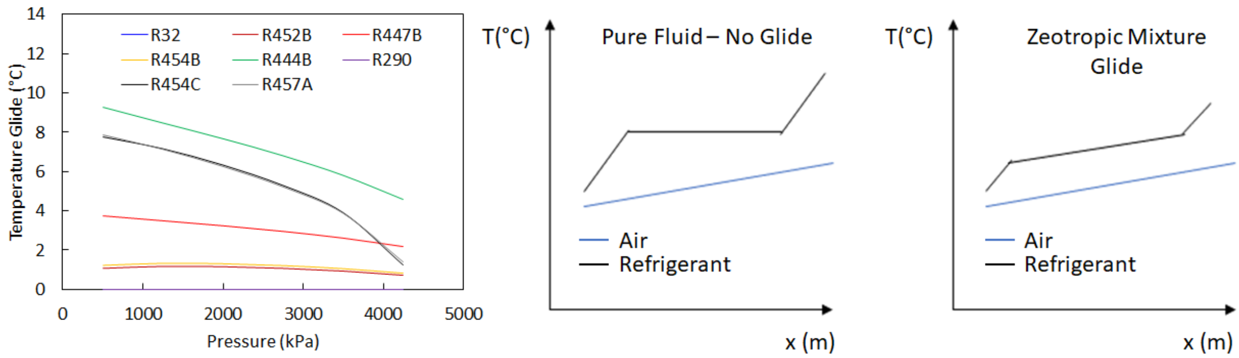


FIGURE 6: PROFILES OF REFRIGERANTS

For Units 6 and 10, the investigated blends, although having narrower saturation than the baseline R32, have similar thermophysical characteristics (Figure 4) with lower temperature glides (Figure 6Figure 7) making them more competitive from a capacity and performance perspective.

C. **System Design Optimization / Modification Framework:** The framework consists of a retrofit of the existing units by properly designing and selecting components that can be replaced with no modification of the cabinets. In other words, any component replaced must occupy the same envelope as the baseline component. The focus of the re-design is on:
   - Compressor
   - Condenser, and
   - Expansion valve

The evaporator designs were not changed for two main reasons: a) some are custom-made wrap-around the blower units, such as in Unit 6, making it hard to quickly find an off-the-shelf option; and, b) the goal is to deliver the same cooling capacity while improving efficiency. For the latter, there is more room for improvement in the condenser by reducing condensing pressure, assuming the evaporator can already deliver the expected capacity.

The fans and blowers were also not considered for change, in part due to the lack of information on the performance curves from the baseline models, but also due to potential high cost and lead time for replacement with secondary impact on performance since 80-90% of the power consumed comes from the compressor.

The first step to assess the level of performance required for each component is to investigate an improved theoretical cycle, which will indicate how much COP improvement can be expected, as well as refrigerant flow rate needs and HX size (UA). To improve the performance of a vapor compression cycle, the pressure lift between evaporating and condensing pressures must be reduced. Consequently, the approach temperatures between air and refrigerant will be reduced as well (Figure 7), thus the thermal capacitance of the heat exchangers must increase. Furthermore, the closer to the saturation region, the closer the cycle reaches the ideal Carnot efficiency (Figure 8).

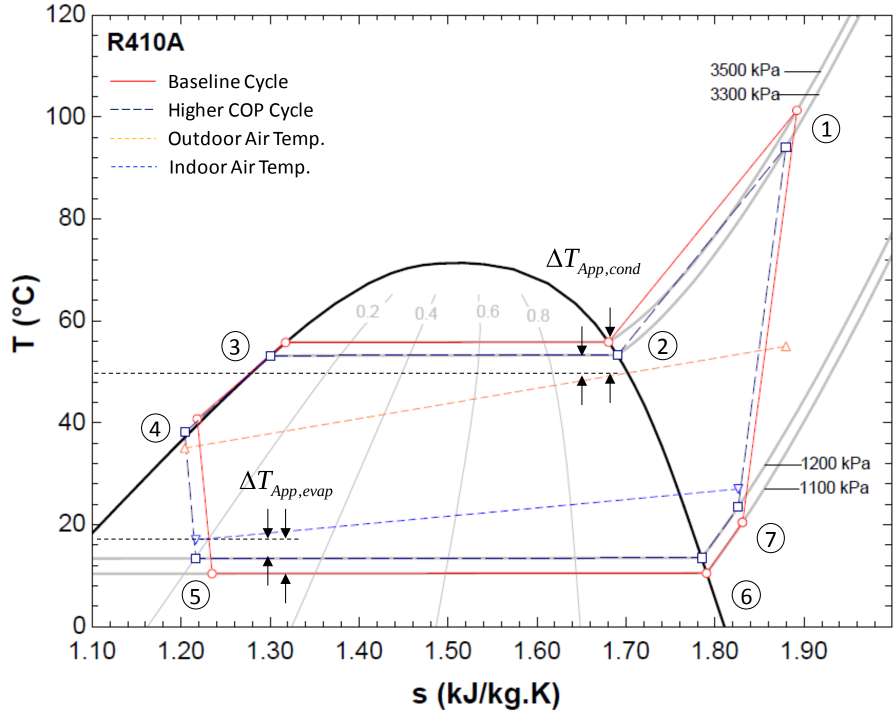


FIGURE 7: ILLUSTRATIVE T-S DIAGRAM FOR BASELINE AND IMPROVED CYCLE

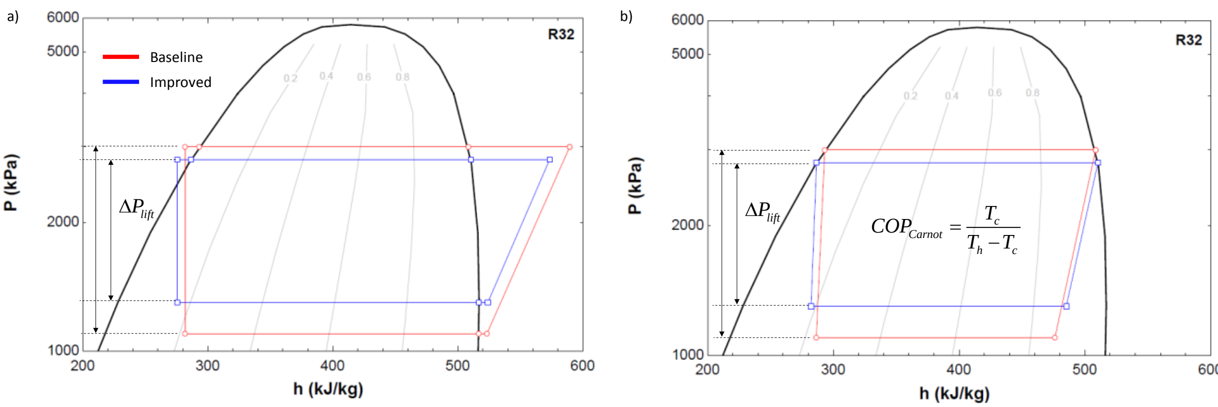


FIGURE 8: DIAGRAM ILLUSTRATING COP IMPROVEMENT A) REAL CYCLE, B) IDEAL CYCLE (CARNOT)

The system design framework is performed according to Figure 9

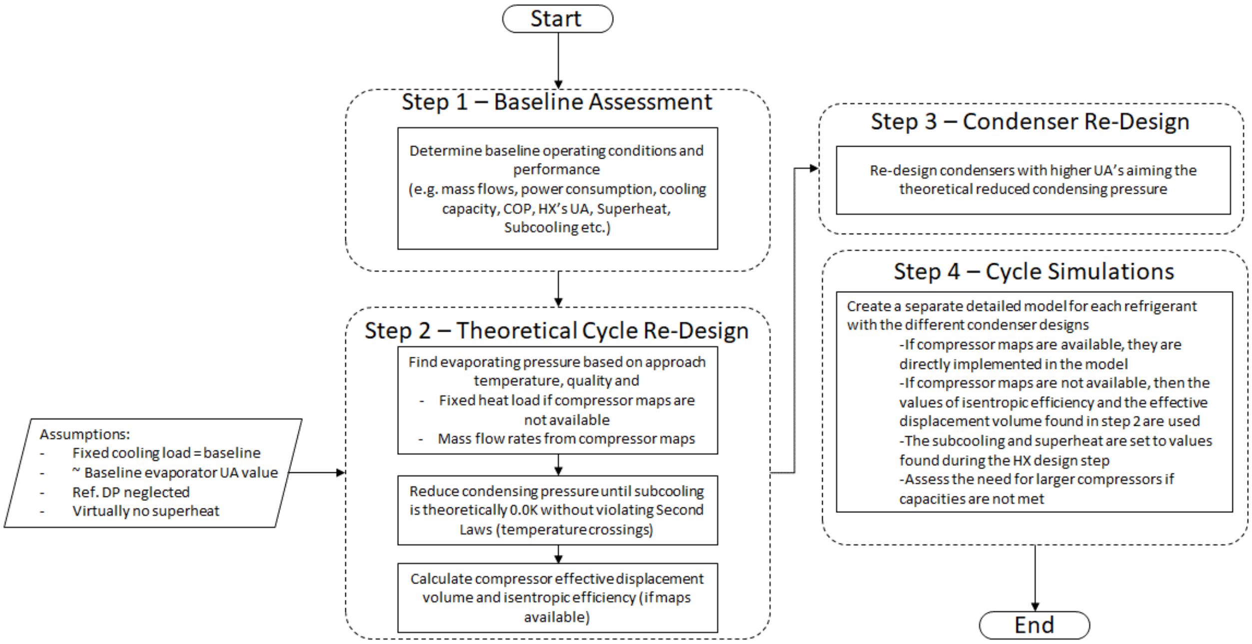


FIGURE 9: SYSTEM DESIGN FRAMEWORK

C. **Compressors**: Modeling compressors are handled in two possible ways, as suggested previously: using performance maps when available or using fixed isentropic efficiency and effective displacement volume. For the larger capacity units (6 and 10), performance maps were provided. Although these compressors were originally designed for R410A refrigerant they may operate – not necessarily optimally – with other refrigerants. Compressor manufacturers supporting this project used proprietary simulation tools, with aid from available empirical data (tests with other refrigerants), to develop theoretical maps for the various refrigerants of interest (Table 5) and made them available to OTS for modeling purposes. It is understood that the predictions are for reference only, and the compressor manufacturer does not guarantee performance for any refrigerants for which the compressors haven't been fully tested.

TABLE 5: COMPRESSOR MODELS

Model	Capacity (BTU/hr)	Frequency (Hz)	Refrigerants
ZP20K5E-PFV	24,000	60	HFC-32, R-454B, R-410A
ZP21K5E-PFV	24,000		
ZP31K6E-PFV	36,000	50/60	R-447B, R-452B, R-454B, R-410A
ZP34K6E-PFV	36,000		

For the smaller units (1 and 4), which were re-designed using HC-290 (Propane), compressor performance maps were not available. The approach for these units then was to set a target isentropic efficiency of 0.7 (baseline data suggests that the compressor efficiencies ranged from 0.55 to 0.65). The required mass flow rate is calculated based on capacity in the theoretical cycle model described above. From there, the effective displacement volume can be determined by the equation below[1]. The latter serves to determine whether a system can use the same compressors for different refrigerants.

$$V_{eff} = \eta_{vol} \cdot V_{disp} = \frac{\dot{m}_{required}}{f \cdot \rho_{suction}}$$

D. **Heat Exchangers:** The condensers design procedure takes into consideration the following:

- **Face area**: baseline face area must be preserved or at most reduced. Furthermore, the aspect ratio must also match that of the baseline so the HX can be drop-in replaced in the same cabinet.
    - o Find the number of tube rows and tube length to match as closely as possible to tube face area and aspect ratio

- **Airside pressure drop and flow rate**: the test data from reports contain only air flow rate measurements, while no information on pressure drop is provided. Additionally, the fan performance curves are also not available, which limits the ability to find the exact operating condition. The baseline models provide an estimate prediction for the pressure drop, which is used as reference.

- **Thermal performance**: this step must be iteratively conducted with the previous step, as such for each design change the air flow rate and capacity are evaluated under the new conditions found in the theoretical cycle re-design.
    - o Gradually increment the condensing pressure until attainable performance is achieved. This process is done iteratively using the theoretical cycle model, to find new expected operating conditions for evaporating pressure, superheat, sub-cooling and refrigerant flow rate.

- **HX Form**: as indicated previously, the HX design is constrained by cabinet dimensions as well as form. In the case of units 1 and 4, the condensers are flat coils placed 90° inside the cabinet (Figure 10), which makes it simpler for drop-in replacement as long as new designs have the same overall dimensions. For units 6 and 10, however, the condensers are L-shaped inside the cabinet (Figure 10). Forming coils is widely done, however, for custom coils it may be a challenge, in particular for MCHX. For this reason, the MCHX designs for units 6 and 10 are sized for a full-face area, assuming the coil can be formed, and a second design that is a single flat slab placed in longer side of the "L" shape (Figure 11).

[1] See Nomenclature at the end of this chapter

FIGURE 10. CONDENSER FORMS: UNIT 1 (LEFT), UNIT 10 (CENTER), UNIT 6 CABINET (RIGHT).

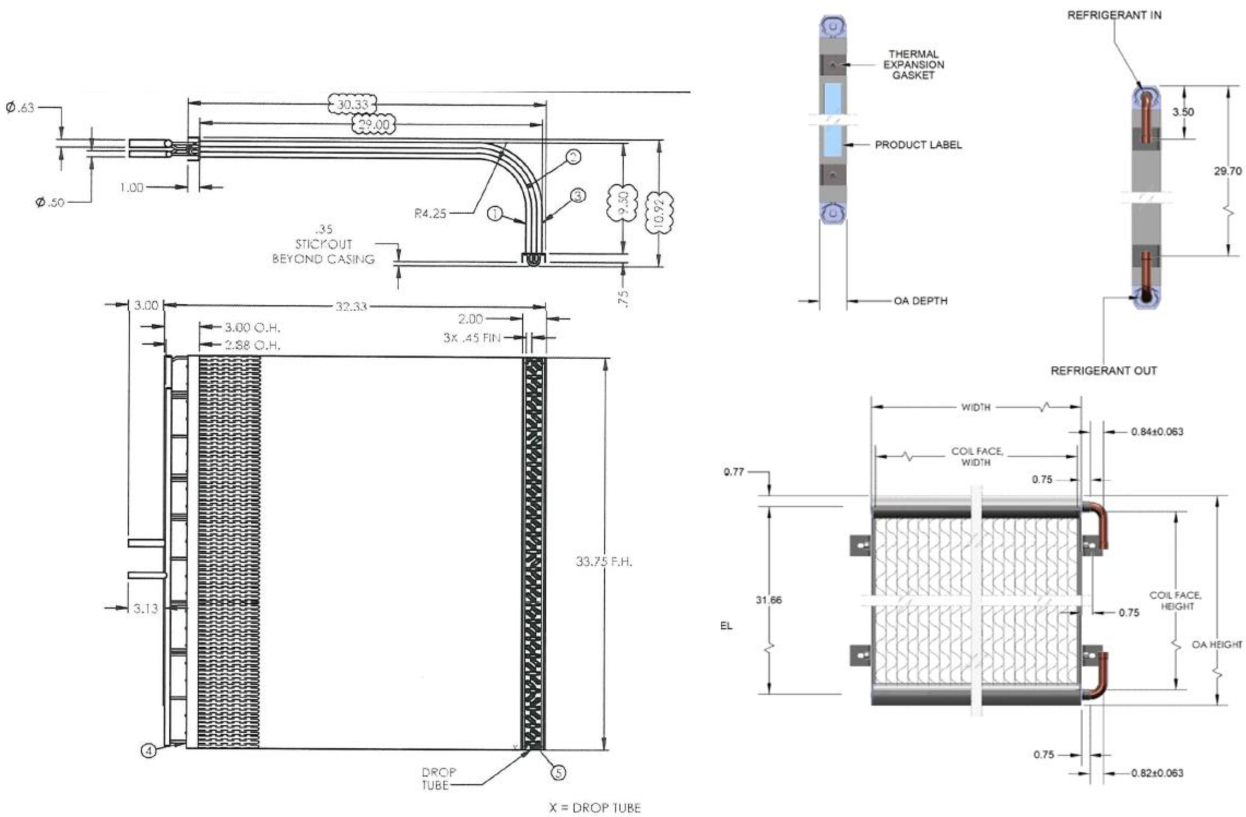


FIGURE 11. HX FORM EXAMPLES: L-SHAPE (LEFT), FLAT (RIGHT).

Summary of Results for Activity 2

Table 6 shows the summary from the design simulation activity

Table 6: Activity 2 Results.

	General Information			Hardware					Performance	
System	**Rated Capacity (@35°C)**	**System Configuration**	**Refrigerant**	**Compressor**		**Condenser**		**Exp Device**	**CC @ 46°C**	**EER @ 46°C**
-	**BTU/hr**	**-**	**-**	**Effective Disp. Vol. (cm^3)***	**Efficiency (-)**	**Type**	**Effectiveness (-)**	**Type**	**%**	**%**
Unit 1	18000	**Baseline**	**R-444B**	**19.8**	**0.66**	**Tube-Fin (5mm Tube)**	**0.20**	**Passive**	**0.00%**	**0.00%**
		Alternate 1	HC-290	25.9	0.70	Same as Baseline	0.35	Active (EXV)	1.40%	8.20%
		Alternate 2	R-454C	24.8	0.69		0.26		4.00%	-1.30%
		Alternate 3	R-444B	19.6	0.70		0.23		4.20%	9.90%
		Alternate 4	R-457A	25.3	0.68	MCHX	0.24		2.00%	3.10%
Unit 4	24000	**Baseline**	**HC-290**	**26.4**	**0.61**	**Tube-Fin (9.5mm Tube)**	**0.24**	**Passive**	**0.00%**	**0.00%**
		Alternate 1	HC-290	26.3	0.70	Tube-Fin (5mm Tube)	0.26	Active (EXV)	1.20%	21.40%
		Alternate 2	HC-290	37.9	0.70		0.20		34.40%	-10.60%
Unit 6	24000	**Baseline**	**HFC-32**	**16.0**	**0.60**	**Tube-Fin (7mm Tube)**	**0.12**	**Passive**	**0.00%**	**0.00%**
		Alternate 1	HFC-32	16.9	0.65	Tube-Fin (5mm Tube)	0.15	Active (EXV)	3.00%	11.20%
		Alternate 2	R-454B	18.4	0.67		0.19		-1.00%	14.80%
		Alternate 3	R-452B	19.0	0.70		0.17		2.50%	13.50%
Unit 10	36000	**Baseline**	**HFC-32**	**19.6**	**0.44**	**Tube-Fin (9.5mm Tube)**	**0.13**	**Passive**	**0.00%**	**0.00%**
		Alternate 1	R-447B	22.3	0.65	Tube-Fin (5mm Tube)	0.25	Active (EXV)	5.10%	47.50%
		Alternate 2	R-452B	23.0	0.67		0.25		6.20%	60.70%
		Alternate 3	R-454B	23.3	0.67		0.25		6.20%	56.50%

** Product of displacement volume and volumetric efficiency*

The General Information describes the baseline unit with the alternate refrigerants used, while the Hardware describes the Compressor, Condenser and the Exp. (expansion) Device for each alternative.

The performance at 46°C is given as a percentage of the baseline performance for the cooling capacity (CC) and Efficiency (EER).

For unit 1 (window unit), the optimized design with the same refrigerant as the baseline can improve EER by 9.9% and using HC-290 can lead to an improvement in the EER by up to 8%.

For unit 4 (decorative split with HC-290), the baseline unit which was supposed to be a true 24,000 Btuh unit had an 18,000 Btuh (26.4 cm^3 effective displacement) compressor with a 24,000 Btuh coils. Optimizing the unit with an 18,000 Btuh compressor would lead to 21.4% improvement in EER, while if a 24,000 Btuh compressor (37.9 cm^3 effective displacement) is used, the EER drops by 10.6%.

The other decorative split (unit 6) running with HFC-32 shows an improvement in EER for all alternative refrigerants.

The unusual results for unit 10 (ducted split) showing a 50% increase in EER is due to using bigger condensers (0.25 effectiveness vs 0.13 for the baseline.

3.7.3. Activities 3, 4, and 5

A. Scope and Implementation of activities

Activity 3: Prototype Units Fabrication

Using design decisions made in Activity 2, OTS constructed two prototypes out of the three that were targeted (see section 4.6 Challenges and Changes). The two units are outlined in Table 7.

TABLE 7: PROTOTYPE UNITS FOR COMPONENT MODIFICATION AND FURTHER TESTING

Category	Unit	Refrigerant(s) for Prototype Development
Decorative Split	6	HFC-32 R-454B
Ducted Split	10	R-447B R-452B

This activity involves modifying the existing prototypes to include the new components while making additional changes, such as adding valves, to enable leak testing in Activity 5.

Activity 4: Evaluation of the Optimized Prototypes

This activity involves physically testing performance of the modified units for at least two ambient conditions:

- ❖ Measurement points include:
  - a. Refrigerant Side
    - i. Compressor suction – temperature, pressure
    - ii. Compressor discharge – temperature, pressure
    - iii. Expansion valve inlet – temperature, pressure
    - iv. Evaporator Inlet – temperature, pressure
    - v. Evaporator Outlet – temperature
  - b. Air Side
    - i. Environmental chamber ambient temperature, relative humidity
    - ii. Condenser incoming air temperature

iii. Condenser exhaust air temperature
iv. Evaporator incoming air temperature
v. Evaporator exhaust air temperature
vi. Evaporator pressure drop
vii. Indoor air flow rate

c. Power

i. Compressor
ii. Fans
iii. Any additional controls or electrical components

- ❖ Conduct troubleshooting measures, as needed, to confirm operation prior to start of testing.
- ❖ Charging the unit was conducted at 35°C (95°F) in the outdoor unit environmental chamber. Conduct a charge optimization to assess the most appropriate refrigerant charge given the test set-up. This will include testing the unit at three different charge amounts to determine the charge that produces the best possible result (COP) at the rating condition. Conducting this step ensures appropriate charge levels and good measurement values.
- ❖ Tests repeated at the high ambient condition T3 (46°C outdoor).
- ❖ Test data analyzed and compared against the modeling predictions from Activity 2. Any system modifications that have potential to improve performance, including further adjustments to the refrigerant charge, were identified.

Activity 5: Analyzing Leaks of Alternatives

In addition to addressing the performance of the individual systems, analysis on refrigerant leakage is needed to meet Project Objective #3. Additional testing were conducted following the performance tests

Results

The detailed outcomes and test data can be found in the OTS report which is attached to this report. The following is a summary of the results:

Unit 6

Some modifications were made to Unit 6 to improve its efficiency. The baseline compressor was replaced with alternate models to account for the change in refrigerant and to improve efficiency. The compressor used with R-454B had a higher displacement volume than the one used with HFC-32. Furthermore, the capillary tubes were replaced with a manual throttling valve simulating the TXV that was installed directly at the evaporator inlet to increase the cooling capacity of the evaporator. A summary of the design modifications evaluated for Unit 6 is listed in Table 8.

Tables 9 and 10 show the performance of Unit 6 for baseline and modifications at 35°C and 46°C ambient, respectively. There is a discrepancy in the measurements from condenser outlet to expansion inlet in the baseline case, since the capillary tube (removed in the modified systems) was located in the outdoor unit. The expansion causes the refrigerant to flash in the liquid line thus compromising the readings at the expansion device. For calculation purposes, the condenser outlet enthalpy was used instead of the expansion inlet.

TABLE 8: UNIT 6 MODIFICATIONS FOR TESTING.

System	Unit 6		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R32	R454B
Compressor	GMCC KSG226N1UMT	Copeland ZP20K5E	Copeland ZP21K5E
Expansion Device	Capillary Tube (outdoor unit)	Manual valve[2] (indoor unit)	Manual valve (indoor unit)

Cooling capacity for the modified unit with either refrigerant was consistently lower by 6-12% than the baseline. The modified HFC-32 system reportedly showed lower mass flow rate than expected, likely the main cause for the lower-than-expected thermal performance. The R4-54B system resulted in a lower performance but was less sensitive to ambient temperature than its R32 counterpart - i.e. cooling capacity was near the same at both 35°C and 46°C, while for HFC-32 there was a ~2,000 BTU/hr reduction with the temperature increase. It is also possible that there is a mismatch between thermophysical property library and actual refrigerant properties for R454B which can happen with newer fluids. The libraries need periodic update as more test data become available.

TABLE 9: UNIT 6 - PERFORMANCE TEST SUMMARY FOR R32 BASELINE (OTS) @ 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	-	HFC-32	HFC-32	R-454B	-	-
Charge	lbs.	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	25,192	23,585	21,966	-6.4%	-12.8%
Energy Balance	%	-2.28%	-4.66%	-3.06%	-	-
Compressor Power	kW	2.11	1.79	1.77	-15.1%	-16.2%
Fan Power	kW	0.32	0.33	0.33	2.2%	2.2%
Total Power	kW	2.43	2.12	2.10	-12.8%	-13.5%
EER	BTU/hr. W	10.37	11.12	10.44	7.2%	0.68%

TABLE 10: UNIT 6 - PERFORMANCE TEST SUMMARY FOR R32 BASELINE (OTS) @ 46°C.

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	-	HFC-32	HFC-32	R-454B	-	-
Charge	lbs.	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	23,390	21,450	21,821	-8.3%	-6.7%
Energy Balance	%	-1.78%	-4.42%	-7.61%	-	-
Compressor Power	kW	2.71	2.32	2.25	-14.2%	-16.6%
Fan Power	kW	0.40	0.42	0.42	5.3%	5.3%
Total Power	kW	3.10	2.74	2.67	-11.7%	-13.8%
EER	BTU/hr. W	7.55	7.84	8.17	3.8%	8.2%

[2] A manual valve was used to mimic a TXV or EXV; recommended as component modification in these systems.

Unit 10

Applying what was learned in the initial modifications to Unit 6, modifications to Unit 10 were limited to include the compressor and expansion device only. Unlike Unit 6, however, the re-test of the baseline system was not successful; refer Appendix D of the OTS report for additional information. However since Unit 6 baseline re-test showed good reproducibility from original data, it is assumed that the Unit 10 original baseline will act similarly.   A summary of the design modifications evaluated for Unit 10 is listed in Table 11. The detailed test data is presented in Appendix E of the OTS report.

TABLE 11: UNIT 10 MODIFICATION FOR TETSING

System	Unit 10		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R447B	R452B
Compressor	Copeland ZP42K6E	Copeland ZP34K5E	Copeland ZP31K5E
Expansion Device	Orifice	Manual Valve	Manual Valve

At 35°C the modified units exhibited almost 20% less cooling capacity with 10% less power consumption, resulting in up to 11% less EER (Table 12). These results were not unexpected since the modified units were re-designed using the 46°C temperature, when the baseline system's performance showed a great degradation of performance. At 46°C condition, the tests confirmed exhibited 2-5% greater cooling capacity with up to 12% less power consumption compared to the baseline, which was equivalent to 13-17% greater system performance.

In Activity 2 the compressor power consumptions were underestimated, as well as the total fan power consumption, leaving the impression the overall performance improvement would considerably be greater than the observed. The cooling capacity, on the other hand, was predicted with less than 2% deviation from test data, validating at least the models created.

TABLE 12: UNIT 10 - PERFORMANCE TEST SUMMARY AT 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	-	HFC-32	R-447B	R-452B	-	-
Charge	lbs.	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	35,543	32,195	28,128	-9.42%	-20.86%
Energy Balance	%	---	7.52%	-3.29%	-	-
Compressor Power	kW	-	2.67	2.4	-	-
Fan Power	kW	-	0.95	0.98	-	-
Total Power	kW	3.761	3.62	3.38	-3.75%	-10.13%
EER	BTU/hr. W	9.451	8.894	8.322	-5.89%	-11.94%

TABLE 13 : UNIT 10 -PERFORMANCE TEST SUMMARY AT 46°C

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**HFC-32**	**R-447B**	**R-452B**	**-**	**-**
Charge	lbs.	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	29,633	31,073	30,292	4.86%	2.22%
Energy Balance	%	---	4.21%	1.21%	-	-
Compressor Power	kW	---	3.18	2.93	-	-
Fan Power	kW	---	0.95	0.97	-	-
Total Power	kW	4.466	4.13	3.9	-7.52%	-12.67%
EER	BTU/hr. W	6.64	7.52	7.76	13.33%	16.95%

Leak Tests

In the interest of time the leak tests were conducted only on Unit 10 for R447B. The choice of refrigerant was based on temperature glide, where R447B exhibits the highest glide amongst the refrigerants evaluated between Unit 6 and Unit 10 (refer to Figure 6). The leak tests were conducted to closely represent field operation. The procedure applied include the following steps:

1- Run unit until steady-state is achieved (repeat 46°C performance test), monitoring capacity and sub-cooling
2- Gradually remove refrigerant from vapor line until capacity is reduced to approximately 50%, if possible
3- Store and weigh removed refrigerant
4- Re-charge with new refrigerant until same sub-cooling is achieved
5- Compare cooling capacities; if more than 5% deviation is observed, repeat steps 1-4, however in step 2, reduce capacity to 25% only
6- Repeat steps 1-5 for the liquid line

The comparison herein presented refers to a leakage of approximately 30% of charge, while reducing capacity in approximately 50% based on airside only. The leak tests showed less than 2% deviation in cooling capacity after re-charge from both vapor and liquid lines (Table 14). Since the capacity deviation was less than 5%, no further testing for 25% capacity reduction was conducted. The results suggest little impact due to fractionation.

TABLE 14: UNIT 10 – R447B LEAK TEST SUMMARY RESULTS.

System			*Liquid Line Leak*		*Vapor Line Leak*	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	**-**	**R-447B**	**R-447B**	**R-447B**	**R-447B**	**R-447B**
Charge	lbs.	6.625	4.27	6.625	4.23	6.77
Cooling Capacity	BTU/hr	31,073	14,216	30,865	15,171	30,587
Energy Balance	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Compressor Power	kW	3.18	2.93	3.18	2.94	*
Fan Power	kW	0.95	0.98	0.98	0.98	0.98
Total Power	kW	4.13	3.90	4.16	3.92	*
EER	BTU/hr. W	7.52	3.64	7.42	3.87	*

*Compressor power consumption was not properly recorded for this test; the error was identified after the fact and the team was unable to retrieve that information. While that compromises the assessment of the overall system performance, the deviations are expected to be marginal. The leak test on liquid line suggest minimal impact on power consumption after re-charge, while cooling capacity was reportedly fully recovered after recharge on both leak tests.

3.8. Conclusion and Recommendations from the Optimization Element

The original scope and schedule were modified during the project as new findings and challenges surfaced. The data analysis and processing from the tests conducted in the PRAHA-I project showed that more testing parameters and instrumentation would have been needed to support the optimization and/or redesign process within the scope of PRAHA-II since PRAHA-I was designed to conduct testing and comparison of cooling capacity vs. EER for the prototypes against the baseline units from same manufacturers. This affected the evaluation of the units' performance and consequently in building the baseline models.

The Conclusion from **Activity 1** is that for systems operating in considerably higher temperatures (greater than 46°C), the resultant impact on performance must be considered since performance will degrade compared to operating under more temperate conditions. Furthermore, the discharge temperature should be considered when selecting alternative refrigerants.

The key components for performance improvement identified were the compressor, condenser and expansion device.

- At higher temperatures, the saturation temperatures and refrigerant density at the compressor suction port can be very different than that from the rated conditions. Larger displacement volumes and efficiency curves optimized for higher pressure lifts might be required. Therefore, the proper selection of the compressor is paramount.

- A better performance condenser will reduce the approach temperature between refrigerant and air, reducing discharge pressure.

At high ambient conditions, the system is forced to operate in higher pressure lift than at rated conditions, but still requires a certain refrigerant mass flow rate. Passive devices such as capillary tubes and orifices may not be able to provide enough expansion to allow the system to operate in higher temperature conditions. An active expansion device such as Electronic expansive valve (EXV) can adequately control operating conditions and maintain design superheat.

The analyses presented in **Activity 2** (design evaluation through modeling) provided good insights on adequate component design and/or selection for proper system functioning when using alternative refrigerants. The tests in activities 3-5 partially served as validation for the models developed, and as check for previous test data from PRAHA I. The key conclusions and recommendations are:

I. HC-290 and HFC32 have wider saturation regions allowing the system to operate with smaller superheat and sub-cooling, while benefiting from two-phase heat transfer.
II. Refrigerants with high temperature glide may require new heat exchanger (HX) designs, namely condensers. The original designs proved to be sufficiently effective to allow for most systems to operate with the different refrigerants; however, better designs would allow for higher system efficiency and potentially less charge. HX designs are severely constrained by allowed envelope dimensions. A complete system re-design would provide an opportunity for designing HX's with even higher efficiency.
III. The results of this analysis suggest that for an effective use of alternate low-GWP refrigerant, a proper compressor selection must be done. Higher isentropic efficiencies are desired for higher temperatures, but most importantly, the displacement volume requirements can vary from one refrigerant to another.
IV. It is also imperative that having an active expansion device (preferably an EXV) to not only allow for more controlled superheat, but also to enable the unit to run with different refrigerants with very different thermophysical properties.

For **Activities 3, 4, and 5**

I. Unit 6 re-tested baseline exhibited similar performance to that found in PRAHA I testing. It should be stressed that the baseline unit by design had its capillary tube located in the outdoor unit. This would cause liquid refrigerant leaving the outdoor unit to flash. The refrigerant enthalpy at the condenser outlet state was used to calculate the refrigerant-side capacity assuming an isenthalpic expansion without heat loss in connecting pipe. This is different from the modified systems of which the capillary tube was removed, and a manual expansion valve was placed at the inlet of the indoor unit. For modified systems, the enthalpy at the expansion valve inlet was used to calculate the refrigerant-side capacity.
II. The Unit 6 modified systems had lower performance than expected from the Activity 2 models. The R32 system configuration exhibited more than 10% less flow rate than anticipated due to performance maps over prediction, which corresponded to 10% lower capacity. The R454B configuration exhibited a deviation of 5% between model and test due also in part to a 3% flow rate over prediction in the model.
III. Unit 10, on the other hand, exhibited an excellent agreement to the models with less than 2% deviation in cooling capacity.

IV. Unit 10 exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This also indicates the importance of proper compressor selection.

V. The higher-than-expected power consumption in the Unit 10 baseline tests is also evidenced by the fact that even with zeotropic mixtures (R-447B and R-452B), Unit 10 had higher cooling capacity and efficiency than the baseline for the 46°C test condition, as projected in activity 2.

VI. Because of the differences in saturation curves from the Activity 2 analysis, HFC-32 tends to result in systems with higher efficiency and less charge when no modifications to the hardware are made. The results showed however, that making appropriate component selection, such as compressors with larger displacement volumes for the other refrigerants, cooling capacities and overall performance were of the same order of magnitude.

VII. Refrigerant fractionation as evidenced by the leak tests, does not appear to be a great concern since less than 2% in cooling capacity was observed after the system's re-charge.

VIII. The model validation adds confidence in the numerical simulation findings and recommendations provided in activity 2.

The **recommendations** for future development are:

- Establish a baseline system by conducting comprehensive testing including measurements and metrics not typically performed in energy certification tests.
- Replacing refrigerants is viable and can be competitive to presently used refrigerants but doing so requires proper component design and selection; compressor and expansion device particularly. Drop-in replacement without hardware change is never recommended.
- It is recommended to always perform numerical simulations, and to conduct at least some level of "soft" optimization analyses that will provide information for an educated system re-design / retrofit at much lower costs than gradual trial-and-error changes.
- Always test the modified systems in the same test setup as the baseline, with the same instrumentation.

Nomenclature

COP	Coefficient of Performance	-
D_o	Tube Outer Diameter	mm
f	Frequency	Hz
FPI	Fins per Inch	1/in
h	Enthalpy	kJ/kg
h_t	Tube Height	mm
HX	Heat Exchanger	-
$\dot{m}$	Mass Flow Rate	kg/s
MCHX	Microchannel Heat Exchanger	-
P	Pressure	kPa
P_l	Tube Longitudinal Pitch	mm
P_t	Tube Transverse Pitch	mm
s	Entropy	kJ/kg.K
T	Temperature	°C
TFHX	Tube-Fin Heat Exchanger	-

UA	Thermal Conductance	kW/K
V	Volume	m^3
w_t	Tube Width	mm
η_{vol}	Volumetric Efficiency	-
ρ	Density	kg/m^3

4. Risk Assessment

This component includes designing, developing and examining a risk assessment model suitable for the use pattern and operating conditions at high ambient conditions and in particular for the Gulf Cooperation Council (GCC) region. The plan was to coordinate with local institutes and experts in HAT countries to build a special risk assessment model that suits the countries' local needs and operating conditions. This process was to be conducted through the following elements:

I. Developing comprehensive terms of reference for building the local risk assessment model;
II. Analyzing the needs of local technical and research institutes to implement the risk assessment model including the technical capacities of personnel and laboratories;
III. Examining the risk assessment model and validating its applicability at levels of manufacturing, installations, operation and servicing.

Each of the above elements was to be led by a local research institute in consultation and cooperation with international associations partnering in this project. This chapter explains what was achieved given the large scope of this component of PRAHA-II.

4.1. Background on Risk Assessment

The concept of risk assessment in RACHP applications is fairly new as it was introduced with the advent of flammable refrigerants. A brief background is presented in this section to explain the concept and the different terms.

4.1.1. Flammability Definition and Classes

Flammability

For a fire to happen there needs to be three elements: a rapid leak of the flammable gas, a concentration higher than the lower flammability level, and a source of ignition as shown in figure below. Figure 12 shows the probability of ignition as the resultant of these three elements. Lower Flammability Limit (LFL), usually expressed in volume per cent, is the lower end of the concentration range over which a flammable gas can be ignited at a given temperature and pressure.

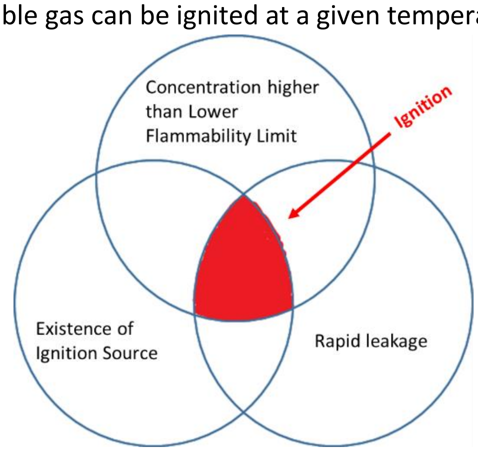


FIGURE 12: FACTORS AND PROBABILITY OF IGNITION

Probability = [rapid Leakage] x [High Concentration] x [Ignition Source]

This report does not aim to cover all aspects of flammability such as the ignition source energy and speed of propagation etc.

Flammability Classification for Refrigerants: Table 15 shows the classes of flammability as defined in ISO 847 and ASHRAE 34.

TABLE 15: FLAMMABILITY CLASSIFICATION FOR REFRIGERANTS

Class	
1	No flame propagation when tested at 60°C and 101.3 kPa
2	Flame propagation and LFL > 0.1 kg/m3 and HOC < 19,000 kJ/kg
2L	Same as 2 except Burning Velocity < 10 cm/s
3	Flame propagation and LFL <= 0.1 kg/m3 and HOC >= 19,000 kJ/kg

Refer to Annex II for a discussion on safety and standards.

4.1.2. Concept of Risk Assessment

The concept behind risk assessment is to define what is an acceptable risk given the conditions for ignition in a particular location. To begin with, a definition of risk is agreed upon and a matrix of probability vs. severity is built. For this purpose, this report adopts the work done by JRAIA in Japan.

Definition of Risk

Risk is a combination of the probability of concurrence of harm and the severity of that harm. Tolerable risk is the level of risk that is accepted in a given context based on the current acceptable values by a community. Residual risk is the risk remaining after reduction measures have been implemented. Safety is freedom from risk which is not tolerable.

The risk levels depend on the severity of injury, the amount of damage to the environment, the frequency at which people are exposed to the danger and the duration of exposure.

Tolerable risk is determined by the search for an optimal balance between the ideal absolute safety and the demands to be met by a product. The factors influencing risk are the practicality and means to reduce risk, the benefit to users, cost effectiveness, and social conventions.

The concept of tolerable vs. unacceptable risk was introduced based on the probability of harm and the severity of harm as per Figure 13.

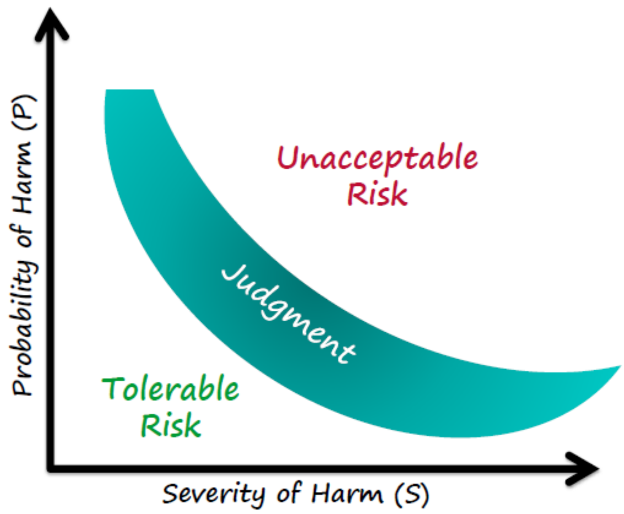


FIGURE 13: TOLERABLE VS. ACCEPTABLE RISK (SOURCE: UL)

The sources of risk start with manufacturing all the way to the end of life of the refrigerant and the equipment. It includes transport and storage, installation and service, operation, as well as removal and dismantling.

4.1.3. Approach of a Risk Assessment Model

The following is part of the process to build a model:

- An outline of the methodology and the components that are the basis for the risk assessment model;
- A model of what data can be collected;
- Information on the regulatory regime and the enforcement mechanisms;
- International standards play a role in the next step of risk assessment in the form of recommendations for local standards; however, the intention is to build a model, not convert it into regulation. Rigorous regulations as those adopted in other regions must be adapted to HAT countries.
- Stakeholders: governments and local research institutions, industry and private sector, and UN Environment & UNIDO;

To determine the outline of the risk assessment model, PRAHA organized a roundtable meeting in cooperation with The Japanese Refrigeration and Air Conditioning Industry Association (JRAIA), and the Air Conditioning, Heating, and Refrigeration Institute (AHRI) as international partners.

The roundtable briefly reviewed the research and testing projects on lower-GWP alternatives for HAT countries as well as the research projects conducted in the United States on A2L refrigerants such as ASHRAE and AHRTI research on flammable refrigerants. Underwriters Laboratory (UL) presented the work that is being done on safety standards and KISR presented a glimpse of their research projects. The industry was also represented in the proceedings and presented their own research and R&D on flammable refrigerants.

A review of the adoptability of flammable refrigerants globally shows the four regions where refrigerants are accepted to varying degrees. Work still needs to be done on HAT regions.

4.1.4. Outline of a Risk Assessment Model

A special expert meeting was held in Cairo in August 2018 focused on the first step of building a risk assessment model through collecting local data and assumptions needed for drafting the model. The meeting aimed to discuss, review and comment on the data collection methodology designed. The meeting was attended by selected experts from the air-conditioning servicing and firefighting sectors, including participation of two members from the Montreal Protocol Refrigeration Technical Options Committee and members of the Halons Technical Options Committee, as well as research institutes' experts, servicing sector expert and National Ozone Officers from Egypt and Kuwait.

JRAIA experts joined the meeting through web-conferencing during the two days. The meeting built clarity and better understanding about the model suggested by JRIAA and included the following:

- Quick Overview of PRAHA-II and First Roundtable Meeting
- JRAIA Risk Assessment Model (Via Web-Meeting) - Brief Introduction to Risk Assessment Concept - Risk Scenarios for installation, use and service of split A/Cs - Explanation of field data/assumptions needed for building the model
- Discussion on Risk Assessment Datasheet and Compilation of Enquiries and Clarification needed from JRAIA - JRAIA Risk Assessment Model - Risk Scenarios for installation, use and service of split A/Cs - Field data/assumptions needed for building the model - Work plan for Data Collection, Review and Validation

The process that will be used is outlined in Figure 14.

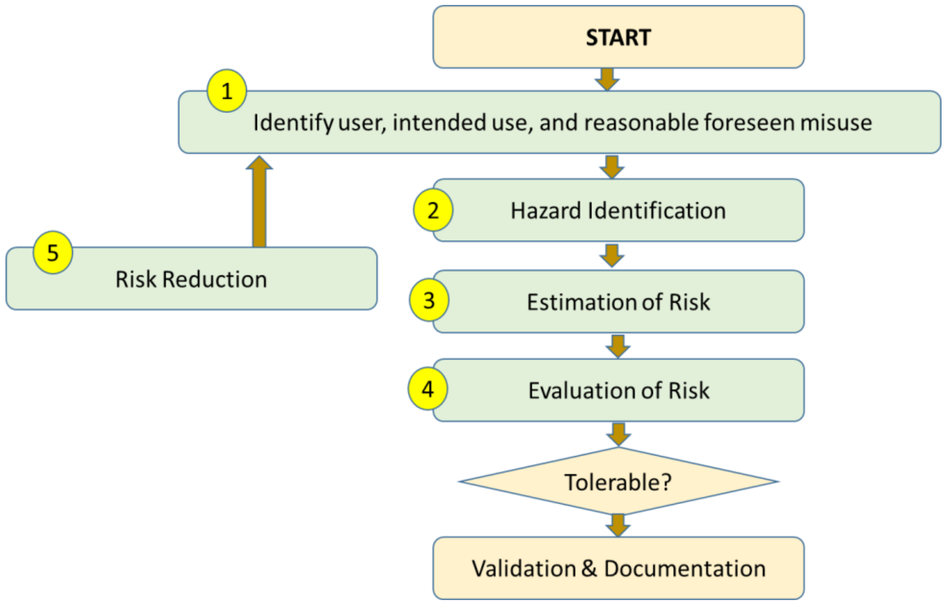


FIGURE 14: PROCEDURE OF RISK EVALUATION ACCORDING TO ISO/IEC 51 (SOURCE: JRAIA)

The experts also discussed the application for the model for which data and information which will be collected. Several applications were suggested with size and use of the room and the sources of ignition. One application will be chosen.
An example of the data tables to be filled before the workshop is shown in **Annex I**.

For more info about the Cairo meeting, please refer to:
https://www.unenvironment.org/ozonaction/news/editorial/un-environment-and-unido-help-countries-high-ambient-temperatures-assess-risk

4.1.5. Global Risk Assessment Efforts

The purpose of this section is not to present a comprehensive background on all the work that has been done globally, but to review those efforts that were presented or shared during the different PRAHA-II events. The PRAHA team is aware of risk assessment efforts done in Columbia and India, among others, some done with the help of implementing or bilateral agencies. Similarly, Chinese associations and industry built their own local risk assessment for the use of A3 refrigerants in unitary air-conditioning applications.

The following is a brief review of research projects that were reviewed both at the International Workshop on Risk Assessment for HAT in Kuwait in Oct 2017 and the Flammable Refrigerant Research and Planning Conference in Chicago in Oct 2018:

Note: AHRTI is the research arm of AHRI in the United States, ASHRAE is the Association of engineers and NFPA is the National Fire Protection Association:

- AHRTI-9007 to conduct refrigerant leak and ignition testing and investigate the control limits and safety factors proposed for IEC 603325-2-40 for air conditioners and 60223-2-89 for refrigeration;
- AHRTI-9009 refrigerant leak detector long-term reliability assessment, to conduct a thorough review of sensor technologies that can detect A2L refrigerants;
- AHRTI-9008 investigation of hot surface ignition temperatures for A2L refrigerants in order to establish a standard;
- ASHRAE-1806 to determine the severity of ignition events using computer modeling;
- ASHRAE-1808 to determine leak rates through mechanical joints;
- NFPA evaluation of fire hazard of A3 refrigerants

AS an example of the work done on A3 refrigerants, the project “Benchmarking Risk by Whole Room Scale Leaks and Ignitions Testing of A3 Refrigerants” conducted by AHRTI conducted leak and ignition testing for HC-290 (propane) under whole room scale conditions to develop data and insight into the risks associated with the use of Class A3 refrigerants. This included parametric testing to investigate how key variables (refrigerant charge amount, release rate and height etc.) influence the ‘ignition event’ under whole room scale scenarios. It involved releasing liquid HC-290 refrigerant into spaces with a variety of viable ignition sources present. The testing scenario simulated a Packaged Terminal Air Conditioner (PTAC) and a mini-split air conditioner (AC) in a typical motel room plus a single door reach-in cooler and a three-door reach-in cooler in a convenience store. The testing scenario was according to the existing requirements or proposed requirements in the IEC Standards 60335-2-40 (for air-conditioning products) and IEC 60335-2-89 (for commercial refrigeration products), and their equivalent North American version published by Underwriters Laboratory (UL).

UL in the US has done work in developing requirements for flammable refrigerants applicable to both air conditioning and refrigeration equipment, as well as the requirements for testing and evaluation of flammable refrigerants including A2L refrigerants. As a result of the work, Standards were published for air conditioners recommending three times the Lower Flammability Limit (3xLFL) under UL 484. For refrigeration, Standard UL 250 for household refrigerators published a 57 gram limit, while UL 60335-2-24 published a 150 gram limit for commercial refrigerators. The transitioning to IEC standards 60335-2-40; 60335-2-24; and 60335-2-89 is now complete.

JRAIA developed a comprehensive risk assessment model for A2L refrigerants. The JRAIA model was used by the PRAHA-II team in the risk assessment work and studied in detail in this chapter. PRAHA-II collaborated with JRAIA to build a model that suits the HAT countries usage and servicing practices.

Initially, it was hoped to cover models for both A3 and A2L. UN Environment and UNIDO were planning to build another parallel model for HAT countries addressing flammable (A3) refrigerants in cooperation with China, given China’s expertise and knowledge about hydrocarbon refrigerants, HC-290 in particular. The work which was planned to be with the Chinese association CHEAA.

4.2. Process of a Risk Assessment Model

The following is a step-by-step outline of a Risk Assessment model based on the workshop that was held in japan in April 2019. Experts from Kuwait and Egypt were invited along with the representative of the national Ozone unit of Kuwait to a two and a half days of workshop and lab visit in Tokyo. The agenda covered a reintroduction of the risk assessment model of Japan with focus on minis-splits as well as the introduction of Japan's experience in data collections methodology. The rest of the workshop was dedicated to the study of a risk scenario prepared by the PRAHA team.

A Step-by-step approach to the case study by the PRAHA team is outlined below:

I. **Selection of equipment type and application**: From residential to refrigeration as per figure below identified by JRAIA. The work on VRF and refrigeration assessment by JRAIA is completed. The PRAHA-II team chose residential air conditioning as it is the most used type in number of units and where the risk might be greatest. The team also identified servicing of the indoor unit as the most relevant for the model.

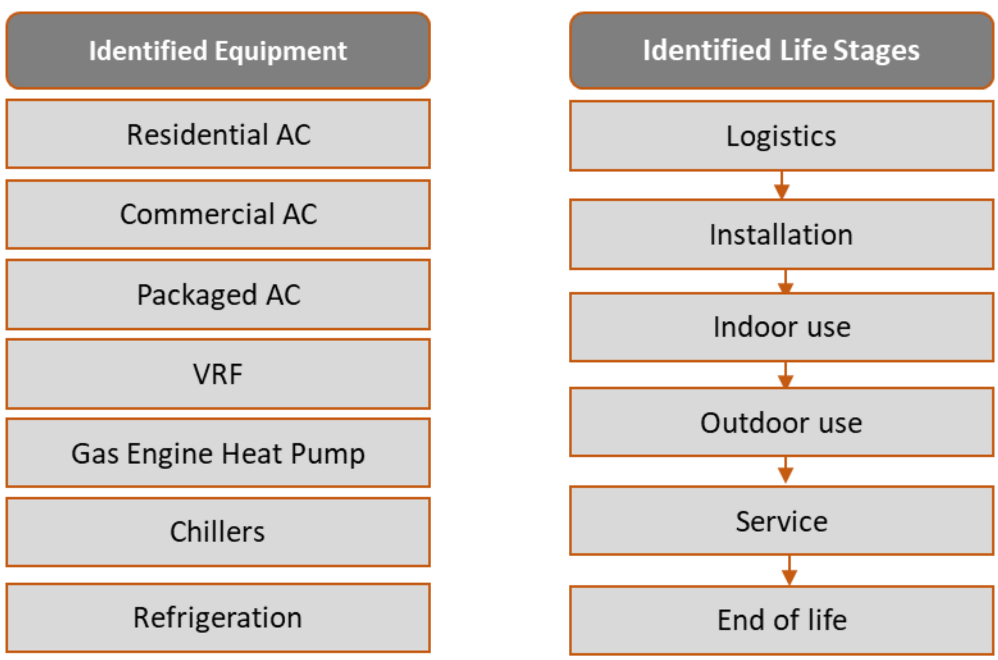


FIGURE 15: SELECTION OF EQUIPMENT AND LIFE STAGE FOR THE RISK ASSESSMENT MODEL

II. **Identify Acceptable and tolerable risk:** Tolerable risk depends on the number of units in the market of the product identified. Tolerable risk depends on the frequency and severity of the accident.

JRAIA defines risk in terms of probability and frequency vs. severity. A low risk is where the probability of an accident is lower and the severity is least. An extreme risk is where the probability is high and the severity is also high.

Table 16 shows the frequency of accidents vs. severity. Frequent accidents leading to catastrophic events are the least acceptable; while improbable of incredible (as in incredibly low frequency) with the least severity are socially acceptable.

TABLE 16 RISK MATRIX - FREQUENCY VS. SEVERITY (SOURCE JRAIA)

	None	**Negligible** (slight injury)	**Marginal** (need for outpatient treatment)	**Critical** (serious injury or need to be hospitalized)	**Catastrophic** (death)
Frequent	C	B3	A1	A2	A3
Probable	C	B2	B3	A1	A2
Occasional	C	B1	B2	B3	A1
Remote	C	C	B1	B2	B3
Improbable	C	C	C	B1	B2
Incredible	C	C	C	C	C
A = Unacceptable risk levels: 1=least, 3= highest		B= Risk levels should be reduced 1= least, 3= highest		C= Socially acceptable risk levels	

III. Analyze Product Cycle

It is necessary to classify the air conditioners into groups and assess the individual risk of each group. If the classification is very narrow, the risk assessment becomes complicated, and data common to different groups cannot be collected because the risk assessment needs to be performed on an individual basis.

The most important considerations for HAT relate primarily to the installation and servicing issue and technicians' skill levels. The temperature has no direct effect on the risk, it is the practice that matters. The question of whether to build a model from scratch or adopt an international model is moot since there is a need to know the status of doing things in the countries that built similar models in order to plug into the locally built model, i.e. level of service, frequency of service, types of installation etc. The team decided to build a model from scratch.

The life cycle range for assessment is shown in Figure 16. Each stage has to be assessed separately and added together to get to the total risk.

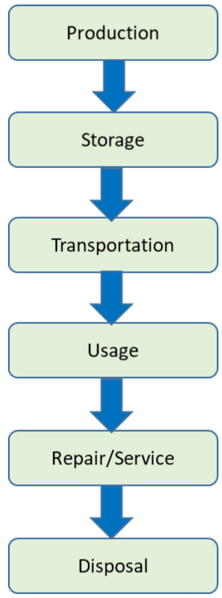


FIGURE 16: LIFE CYCLE RANGE FOR ASSESSMENT

The determination of tolerable risk depends on the population of products in the country. The example from Japan is in Table 17:

TABLE 17: DETERMINATION OF TOLERABLE RISK LEVELS

Product/System	Unit Population	Tolerable risk Usage stage	Tolerable risk Service stage
Residential AC	1×10^{8}	1×10^{-10}	1×10^{-9}
Commercial AC	7.8×10^{6}	1.3×10^{-9}	1.3×10^{-8}
VRF	1×10^{7}	1×10^{-9}	1×10^{-8}
Chillers	1.34×10^{5}	7.5×10^{-7}	7.5×10^{-7}
Condensing units	1.46×10^{5}	6.9×10^{-8}	6.9×10^{-7}

The PRAHA team used the JRAIA approach to set the tolerable risk for residential units at the following levels:
For the usage stage = 1 / 100 x unit population
For the service stage = 1 / 10 x unit population

And the risk map becomes as in Figure 17:

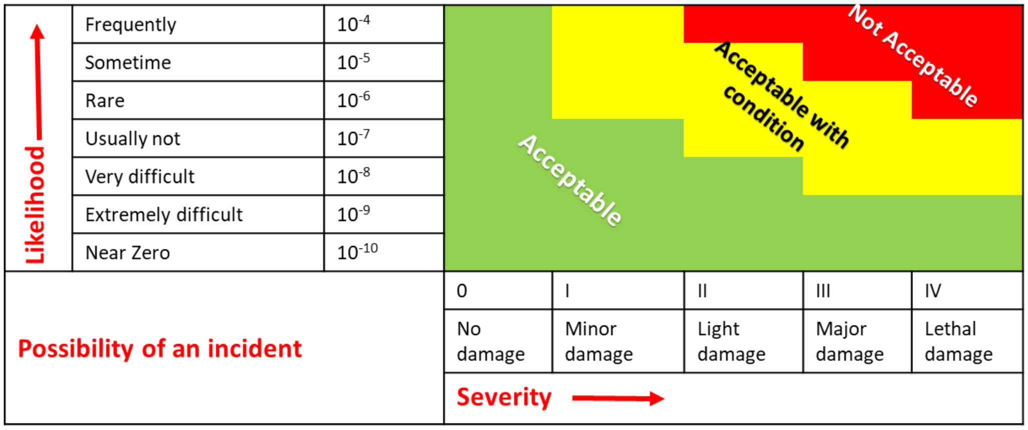


FIGURE 17: RISK MAP

IV. Risk Scenarios

A critical stage of the risk assessment is to identify those scenarios in which an ignition source is present in conjunction with a flammable concentration of leaked refrigerant. To better understand these scenarios, one must consider the various triggering events which could cause refrigerant to be released, the location of the release, and the specific type of person that might be present (*i.e.*, a worker, repair person or customer) at the time of the release. It is important to note that, during normal operations, the refrigerant will be contained within the system, and thus there is no risk of adverse events associated with these refrigerants during regular use. However, if refrigerant leaks from the equipment and is not dispersed prior to accumulating to a flammable concentration and a sufficient energy source is present, refrigerant ignition could occur (AHRTI 8009)

The first step in a risk analysis is to select a risk assessment method. There are three known methods used: Event Tree Analysis (ETA), and Failure Modes and Effects Analysis (FMEA), and Fault Tree Analysis (FTA). ETA is based on binary logic, in which an event either has or has not happened or a component has or has not failed. FMEA is a structured approach to discovering potential failures that may exist within the design of a product or process. Failure modes are the ways in which a process can fail. Effects are the ways that these failures can lead to harmful outcomes for the user. The goal of FTA is to provide an order of magnitude estimate of the likelihood that the outcome in question will occur (US NRC, 1981).

The team chose the fault tree analysis in line with JRAIA. Refer to item VII for FTA description.

The risk assessment of flammable refrigerants considers two individual phenomena: the presence of an ignition source and the generation of a flammable volume. The risk scenarios that were considered were:

A. Refrigerant leak during maintenance work on the indoor unit during brazing and due to pipe breakage by corrosion with an ignition source caused by live wire, static electricity, or electric tool such as screw drivers;

B. Refrigerant leak during brazing of outdoor unit with leakage caused by prior maintenance work or during maintenance work and an ignition source from the brazing torch;
C. Refrigerant leakage during normal home use caused by pipe breakage through corrosion, external pressure or natural causes such as earthquakes with an ignition source of an open flame, electric spark or static electricity.

V. Select Risk Analysis Sources

The input into the model is taken from data tables for the type of application and usage of the equipment that are being studied. Source for input into the volume of the flammable cloud can be taken from research done for the type of gas. Data for source and time of ignition can sometimes be available from the fire department.

VI. Data Collection

Data collection takes into consideration the following:

a) Select the stages of the life cycle of the air conditioners. Choose the manner of classification of manufacturing, transportation, use, service, and disposal of an air conditioner into separate stages for evaluation. The evaluation of the manufacturing stages of each product is normally the responsibility of the manufacturer;
b) Investigate the conditions of installation of the selected air conditioner to determine the conditions to be evaluated during the risk assessment;
c) Determine the severity of the hazard focusing on the damage caused by flammability;
d) Set tolerance levels. Set socially acceptable probability of harm for the air conditioner;
e) Investigate refrigerant leakage rate, speed, and amount based on surveys conducted with air conditioning service companies. The initial leakage location and leakage concentration should also be determined;
f) Determine flammable time volume through CFD or calculations. For the conditions set as per point (b), the flammable time volume can be calculated by CFD simulation based on the leakage amount, speed, and concentration of the refrigerant as per point (e).
g) Consider ignition sources. Distinguish the ignition properties depending on whether the ignition source is a spark (for example, electrical contacts, lighter, and/or static electricity), or an open flame (for example, candles, matches, and/or combustion equipment).

VII. Fault Tree Analysis (FTA)

It utilizes a "top-down" approach, starting with the undesired effect as the top event of a tree of logic. Fault trees (FTs) consist of various event boxes, which reflect the probability or frequency of key events leading up to a system failure. The event boxes are linked by connectors (gates), which describe how the contributing events may combine to produce the system failure. Events may be combined in different ways: in cases where a series of events must all occur to produce an outcome (e.g., ignition source and sufficient oxygen to support combustion), the probabilities or frequencies of the individual contributing events are multiplied via an "AND" gate; in cases where only one of a series of events is needed to produce an outcome (e.g., a strong spark, open flame, or a hot surface all possibly leading to refrigerant ignition), the probabilities are usually added via an "OR" gate. (AHRTI 8009, 2015).

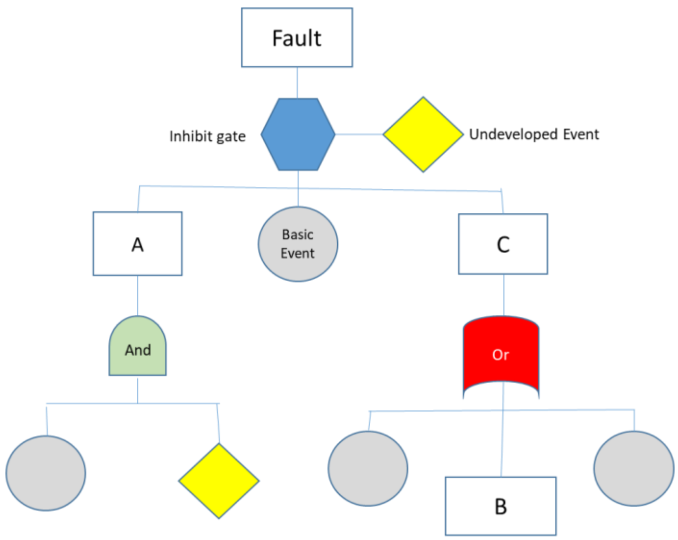


FIGURE 18: FAULT TREE ANALYSIS (FTA) MODEL

In the case of flammability, the probability of leakage is combined with ("and" gate) the possibility that the length of time that flammable cloud exits covered area would lead to ignition in case of the existence of an ignition source (another "and" gate).

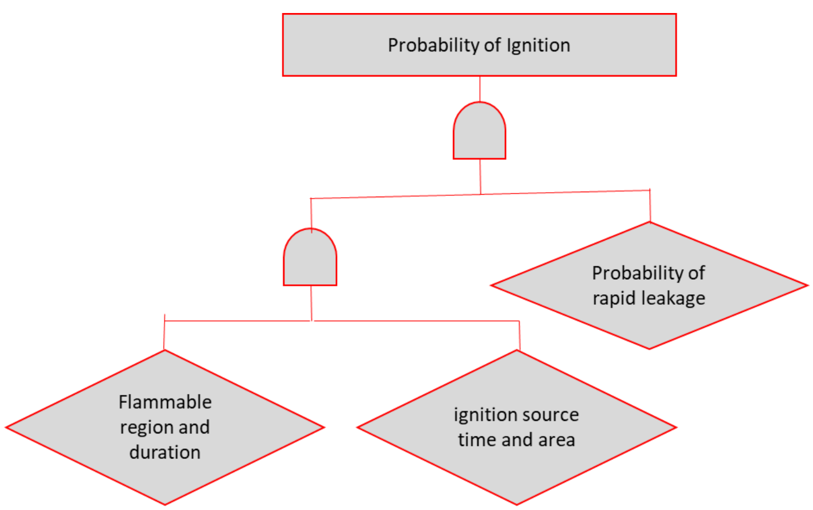


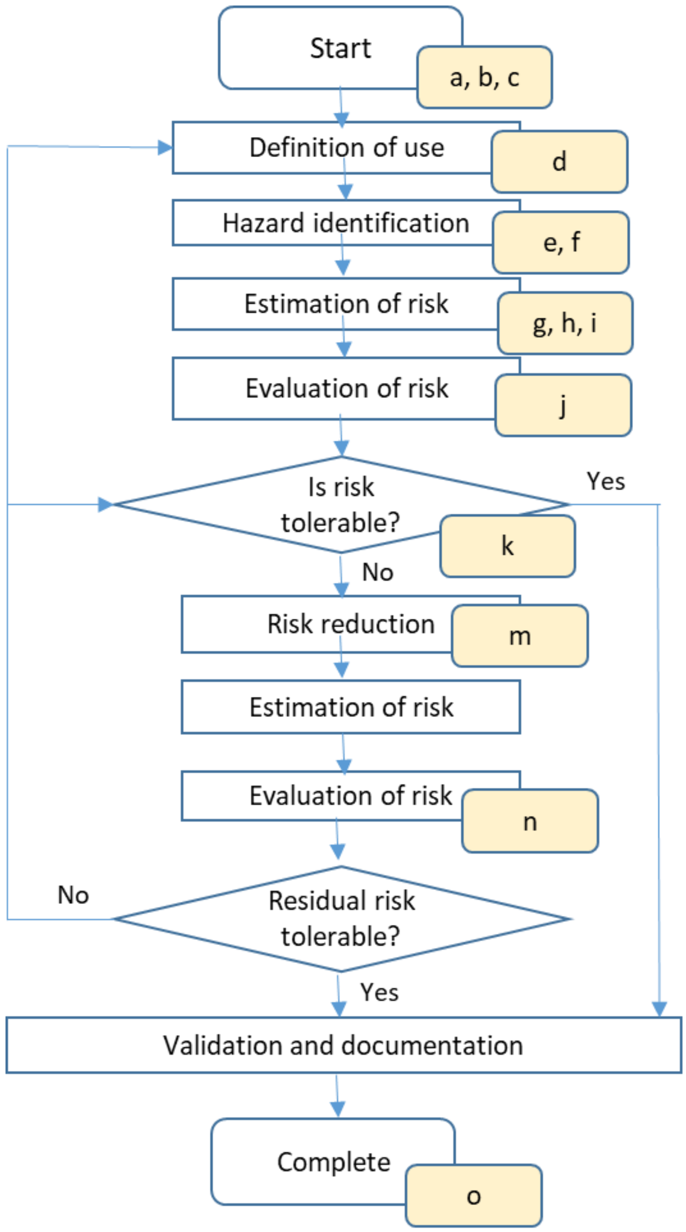
FIGURE 19: PROBABILITY OF IGNITION FTA

In the development of FTA for flammability, the presence of the flammable region and the ignition source correspond to independent trees. Then, their probabilities are multiplied in the final step to calculate the accident probability.

When the contents are reviewed, the risk is evaluated against the risk map in item III above and the calculated accident probability is compared to the acceptable probability in the risk map. The risk tolerance propriety is then determined.

VIII. Suggest Measure to Mitigate Intolerable Risk

When the tolerance from the risk evaluation in the steps above is satisfactory, the risk assessment ends. If the risk exceeds the tolerance, countermeasures to reduce the risk should be taken. These countermeasures include the implementation of regulations and other measures like introducing safety procedures in order to reduce the risk of accidents. In some instances, it might be necessary to revise laws and regulations in order to ensure that they cover the accepted probability. The reiterative process, which is explained in Figure 20, is as follows:



a) Select risk assessment method
b) Select product
c) Select stages of the product life, i.e. usage or service etc.
d) Investigate installation circumstances
e) Determine severity of hazard
f) Set tolerance levels
g) Investigate refrigerant leak rate, speed and amount
h) Determine flammable time volume
i) Consider ignition sources
j) Develop FTA
k) Compare against tolerance
l) Evaluate risk against tolerance
m) Reduce risk with countermeasures
n) Redevelop FTA
o) Confirm and publish

FIGURE 20: FTA REITERATIVE PROCESS

IX. Recommend Standards and Codes

Once the countermeasures have been introduced, the FTA factors are reviewed and these countermeasures are added in the appropriate position of the tree. A new calculation can then be made and repeated until the calculations confirm the accepted tolerance according to the risk map. The results can then be released to the public and standards and codes can be drawn.

4.3. Example of a Risk Assessment Model

The team chose a case study of an office space in a government building during the usage phase when the equipment is running and during the repair/service stage. The target product is a 5.3 kW split system using an A2L refrigerant. The team selected the Fault Tree Analysis method which is described under item VII below. The target product and the indoor and outdoor conditions plus the service case are shown in the tables below.

At the workshop in Tokyo in April 2019, the PRAHA team worked with the JRAIA experts to do two case studies using the information provided by the PRAHA team. The two case studies are:

- During usage of an air conditioner in a government office. The sources of ignition are extreme including charcoal and lighter used for incense burning, an aroma candle, as well as cigarettes and lighters as smoking is still allowed.
- During the repair stage during brazing with sources of ignition including the brazing burner, a cigarette and a lighter.

Table 18 lists the equipment as well as the indoor and outdoor conditions

TABLE 18: INFORMATION FOR THE RISK ASSESSMENT MODEL USED BY PRAHA TEAM

Target Product	Value
Model number	CS-PC36JKF
Type(cooling / HP)	HP
Capacity(kW)	10.5
Refrigerant type	A2L
Refrigerant amount(kg)	2.7
Alternative refrigerant type	HFC-32, R-454B

Indoor Condition during usage of target product		Value
Room size (m^2)	max	25
	min	16
Height of installation(m)		2.1
Ceiling height(m)		2.8
Ventilation	yes/no	YES
	Ventilation amount (m^3/hr.)	80
The area of the gap under the door (m^2)		0.02
other openings, if any (m^2)		0

Outdoor Condition during usage of target product		Value
Size of the place enclosed with walls , or fences etc.(m^2)	max	8
	min	4

Condition during repair of target product	value
Average size of outdoor spaces for repairs (m^3)	20
Percentage of single outdoor unit installations(A%)	50
Percentage of the installations of multiple outdoor units (B%)	50
Average working hours per repair (outdoor unit) (hr.)	1
Average working hours per repair (indoor unit)(hr.)	0.5
Wind condition (wind velocity) (m/s)	1 TO 3
Windless condition percentage (%)	10

(Windless condition; 0.1m/s or less. the windless rate in one year.)

Notes:

- No alternative refrigerant is available from the manufacturer for this product;
- Ventilation amount was calculated based on 1.5 air changes per hour;
- Gap under door was based on the door width is 1.00 m, gap with floor is 2 cm;
- The outdoor unit was assumed to be installed on a roof open area.

The methodology is to calculate the probability of ignition due to a space factor and a time factor.

Space Factor

The space factor takes into consideration the space volume, the volume of the flammable cloud, and the volume of the source of ignition. The volume of the flammable cloud depends on the leakage rate and other considerations such as pressure. The volume of the source of ignition can be very small as in the case of a spark, or sizeable as in the case of an open flame.

Time Factor

The time factor takes into consideration the number of occurrences of the ignition source and the duration of each occurrence.

Terminology

The following terminology will be used in the calculation example:

T_{Ref} = Time of application: 24 hours for usage or duration of maintenance for service
T_S = Time of Ignition Source
T_F = Time of Flammable Cloud
V_{FT} = Flammable Volume Time Integration
V_{SOI} = Volume of source of Ignition
$V_{F\ Cloud}$ = Volume of Flammable Cloud
V_{Ref} = Volume of space or room
$P_{A,\ B\ or\ C}$ = Probability of ignition for the different sources of ignition (A), (B), or (C)
P_R = Refrigerant Leak Probability

Equations

The Volume of Flammable Cloud is the Flammable Volume Time Integration divided by the Time of Flammable Cloud

$\mathbf{V_{F\ Cloud} = V_{FT} / T_F}$

The probability of ignition is the sum of the space and time factors for each source of ignition.

The probability of time is calculated as the sum of the time of the flammable cloud plus time of source of ignition divided by the time of reference (usage or service time).

$\mathbf{P_T = (T_F + T_S) / T_{Ref}}$

The Probability for Space is similarly calculated as the sum of the volume of source of ignition plus the volume of the flammable cloud divided by the reference volume which is the volume of the room or space where service is done.

$\mathbf{P_S = (V_{F\ Cloud} + V_{SOI}) / V_{Ref}}$

The probability for one source of ignition (A), referred to as "Event" is the multiple of the Time probability and the Space probability:

$\mathbf{P_A = P_T \times P_S}$

The probability for all events is sum of the probabilities for all sources of ignition. The three sources identified in the example i.e. charcoal, cigarette and candle are herein called A, B, and C

$\mathbf{P_{Events} = P_A + P_B + P_C}$

$\mathbf{P_R}$ = Leak Frequency x Number of Occurrence in a 24 hour period

The Total probability is the multiple of the probability of each event by the Refrigerant Leak probability

$\mathbf{P_{Total} = P_{Events} \times P_R}$

4.3.1. Simulation of Time Factor and Space factor During Usage Stage

The data in Table 19 was provided by the PRAHA-II team for the workshop.

TABLE 19: DATA FOR THE CALCULATION OF RISK FOR USAGE STAGE

Event	Ignition source	No. of Occurrence	Duration per day	T_S = Time of Source
A	Charcoal + lighter	2	1 hour	I hr/2
B	Cigarette+ lighter	2	0.2 hour	0.2 hr/2
C	Aroma candle	4	3 hours	3 hr/4

Flammable volume Time Integration:

- T_F = 18 minutes/60 minutes = 18/60 hour Time of the flammable cloud. The time is derived from lab data for the type of refrigerant
- Ts is show in table 19
- $V_{F\ Cloud}$ = 6.4 x 10^{-2} m^3 min/ 18 minutes: Volume of the flammable cloud for indoor unit is derived from simulation data for the class of refrigerant and type of application.
- V_{SOI} is negligible.
- T_{Ref} = 24 hours: Time of application is 24 hours since usage is throughout the day
- V_{Ref} = 25 m^2 floor area x 2.1 m height of the indoor unit.
- $1x10^{-3}$ = Leak frequency per year taken from a study for Japan as data is not available from the countries under study.

Figure 21 shows the FTA calculation for the usage stage.

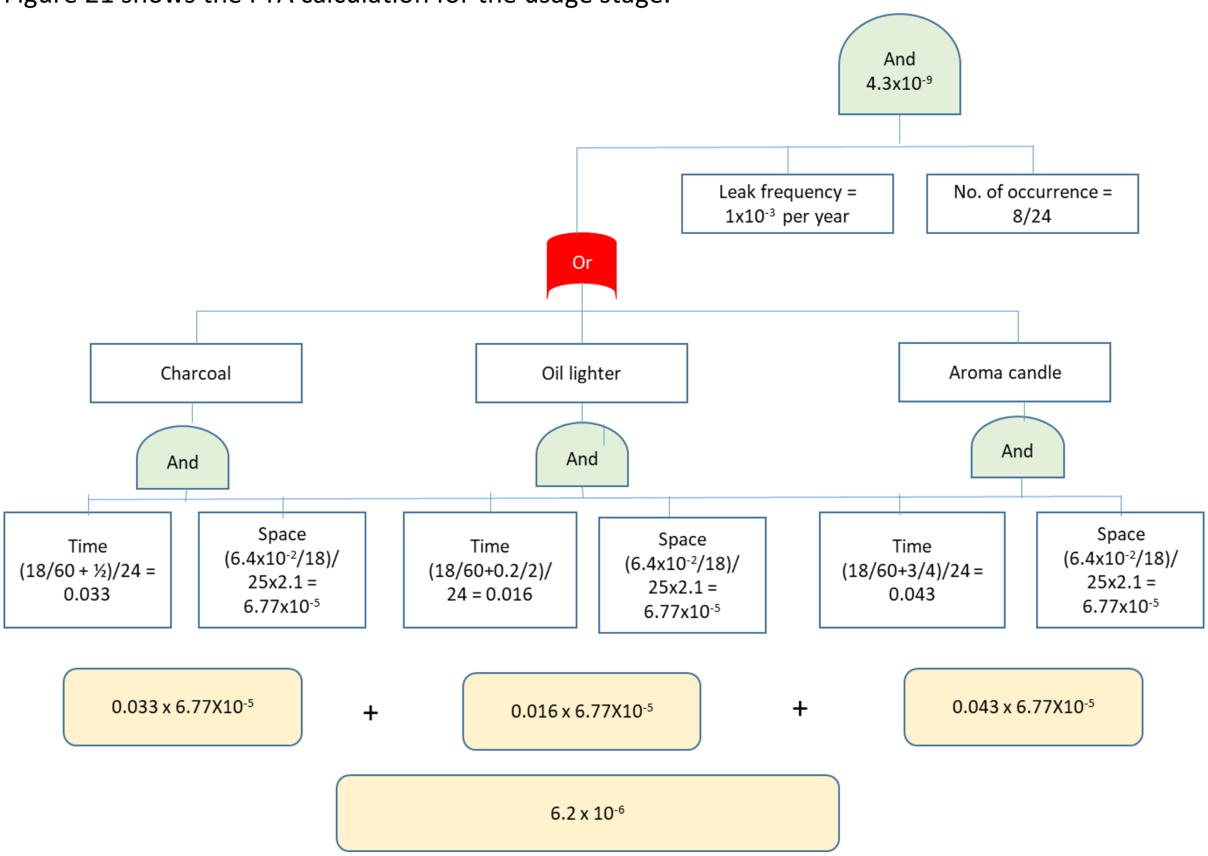


FIGURE 21: FTA FOR USAGE STAGE

For each event, i.e. charcoal, oil lighter, and aroma candle the probability of time and space are calculated according to the equations given above, for example:

- For charcoal the time factor is the sum of the time of the flammable cloud and the time of the ignition source divided by the usage time which is 24 hours. The probability equation is $(T_F + T_S)/T_{Ref.}$ T_F is 18/60 derived from data, Ts = ½ from table 19 and T_{Ref} is 24 hours.
- The space factor for charcoal is $(V_{F\ Cloud} + V_{SOI})/V_{Ref.}$ $V_{F\ Cloud}$ is $6.4x10^{-2}$ /18 while V_{SOI} is negligible. $V_{Ref.}$ is the volume up to the height of the unit = 25x2.1

- The addition of the three ignition sources gives a probability of $6.2x10^{-6}$ which is P_{Events}
- $P_{R} = 1x10^{-3} \times (8/24) = 7x10^{-4}$
- The Total probability is P Events x PR = $(6.2x10^{-6}) \times (7x10^{-4}) = 4.3x10^{-9}$ shown in the top "And". This puts the probability in the "Extremely Difficult" area of Figure 17: Risk Map.

4.3.2. Simulation of Time Factor and Space factor During Servicing Stage

TABLE 20: DATA FOR CALCULATION OF RISK FOR SERVICE STAGE

Event	Ignition source	No. of Occurrence	Duration per day	T_S = Time of Source
A	Burner	2	2 minutes	4/2
B	Cigarette	2	3 minutes	6/2
C	Lighter	2	10 seconds	0.167/2

Flammable Volume Time Integration

$V_{F\ Cloud} = 6.3 \times 10^4\ m^3$ sec/3600 sec Volume of the flammable cloud for outdoor unit is derived from simulation data for the class of refrigerant and type of application.

V_{SOI} is negligible

T_{Ref} = 60 minutes (I hour)

$V_{Ref} = 20\ m^2$ space x 3.5 m height. This is the volume of service space for the outdoor unit.

Ts is shown in table 20

T_F is 60 minutes is the time of the flammable cloud

T_{Ref} is the time of service which is 60 minutes

The FTA for servicing stage is shown in Figure 22.

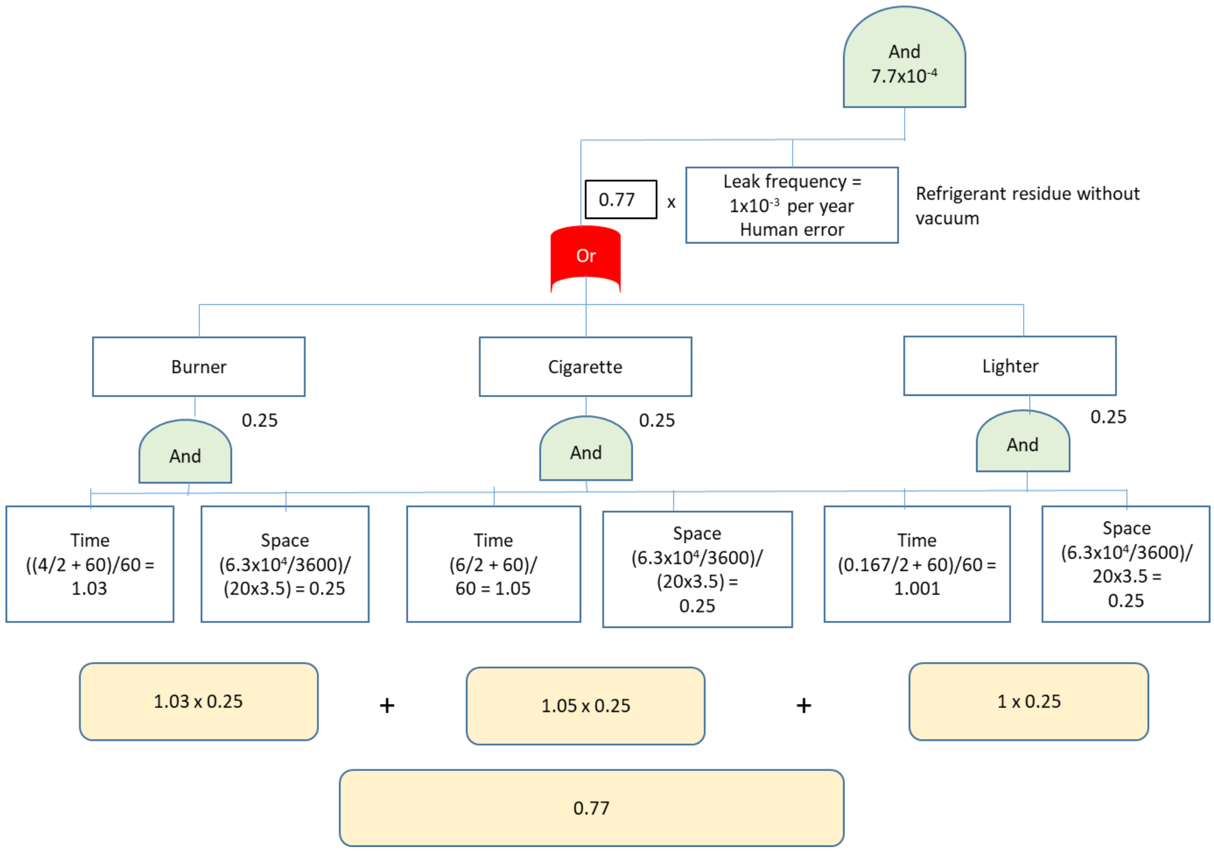


FIGURE 22: FTA FOR SERVICING STAGE

The calculations are similar to the usage stage example given above.
The Total probability is 0.77x $1x10^{-3}$ = $7.7x10^{-4}$ which is shown in the top "And". This puts it in the "Frequent" from the Risk Map of table 17 and mitigation measures should be taken. One evident measure is to ban smoking in the service area!

4.4. Conclusions and Recommendations from the Risk Assessment Element

The above two FTA were created in collaboration with HAT countries and Japan. The purpose of this FTA was to simulate a risk scenario in HAT region with unique climate, product-usage, lifestyle and culture which differs from Japan's case. The exercise has shown the need for a reliable data for the HAT region on leaks, practices etc.

Building a risk assessment model for the HAT countries that suits the climate and the service practices of the local technicians helps the HAT countries, as well as setting the foot for all A5 countries, in understanding the risk associated with flammable refrigerants and adopting the needed regulations and training programs especially in relation to the logistics of lower-GWP based technologies i.e. installation, transportation, storage, servicing and decommissioning.

The recommendation is for HAT countries to continue the risk assessment based on actual situations, and reduce the risk by implementing various measures that are verified by FTA. It is also important to minimize ignition probability by implementing various measures that are verified by FTA. In addition, the risk assessments of other stages matching cultural and lifestyle aspects should be studied.

References for chapter 4

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US Nuclear Regulatory Commission (US NRC). 1981. "Fault Tree Handbook." NUREG-0492. 209p. January.

5. Overall Conclusions and Recommendations

The outcomes of PRAHA-II components can draw several concluding remarks in relation to the main objectives of the project which can be summarized as follows:

In relation to support the process advancing the promotion and deployment of lower-GWP alternatives:

I. A tailored Risk Assessment is essential, not only for HAT countries, in better understanding safety implications associated with deploying alternative refrigerants, either A2L or A3, considering the specifics of different types of equipment and life stages.
II. Efforts in building risk assessment models should be exerted towards analyzing risks in the logistics side of the supply-chain i.e. Installation, In-door use, outdoor use, servicing and end of life (decommission); understanding the design and manufacturing risk assessment are covered by relevant international standards which should more or less apply to most countries.
III. The concept of risk assessment is quite similar worldwide, including methodologies in calculating and analyzing severity and frequency of risks. However, criteria for acceptable tolerance levels may differ depending on local considerations. Measures to mitigate risks would depend on type of exiting/operational standards and/or codes in each country noting
IV. Few Article 5 countries and some of the non-Article-5 countries have built similar models. Learning from the pioneers in risk assessment models through partnership and cooperation will leapfrog the technical difficulties and provide a quick access to building the model.
V. PRAHA-II was the first step in is providing the impetus for this leapfrogging. Similarly, Building the risk assessment model with the involvement of local research institutes and organizations will add depth and reach for those institutes and involve the HAT countries in the global research efforts on new alternatives as well as build countries' ownership.
VI. Building a risk assessment model for the HAT countries that suits the climate and the service practices of the local technicians will help the HAT countries will set the foot of A5 countries, not only HAT, in understanding and establishing local risk assessment models hence adopting the needed regulations and training programs especially in relation to the logistics of lower-GWP based technologies i.e. installation, transportation, storage, servicing and decommissioning.

In relation to building capacities of local industry to better design with lower-GWP alternatives:

VII. The optimization work on the prototypes of PRAHA-I is helping the OEMs who built the original prototypes get the best support in their R&D efforts. The activities of that element substantiated by result of testing of the optimized units confirm that enhanced design and the use of the proper components can lead to better performance and energy efficiency.

VIII. The optimization element of PRAHA-II also pointed out that components, especially compressors for the new refrigerants, are still not widely available. These and other limitations have to be dealt in order to help manufacturers make an informed decision on the way forward.

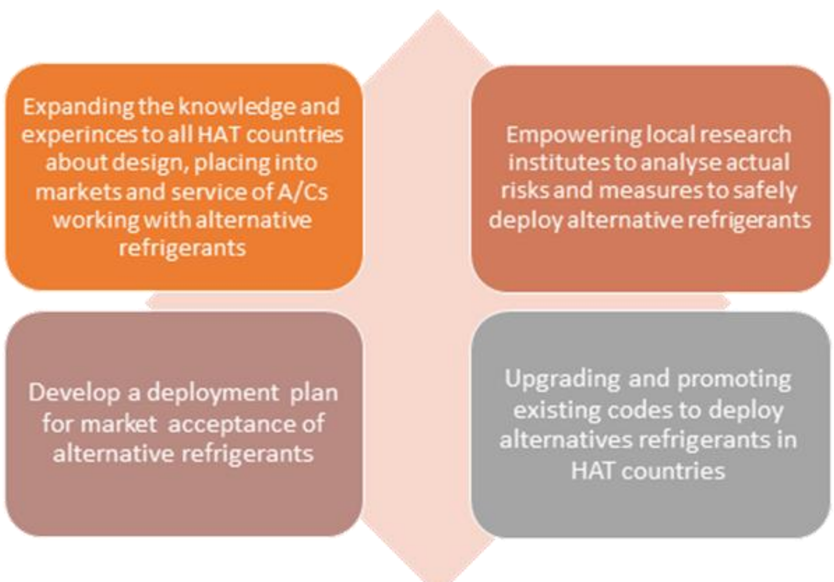
IX. PRAHA capacity building activities have helped the PRAHA stakeholders in acquiring added knowledge about working with alternative lower-GWP refrigerants that are flammable. The study tours have exposed stakeholders to the latest in technology for both A2L and A3 refrigerants at global technology centers. The capacity building activities helped many manufacturers in HAT countries in building or engaging in other research projects.

In relation to maintaining sustainable technical platform to support PRAHA process and sharing knowledge about up-to-date technological developments amongst HAT countries:

X. The capacity building efforts have turned PRAHA into a global process that can be extended to all 35 HAT counties and not only the Gulf and Middle East countries that were engaged with PRAHA-I.

XI. PRAHA-II events continued to attract global and regional participation in terms of government authorities, technology providers, manufacturers, and international/regional institutes. PRAHA presentations and knowledge sharing at networks of ozone officers and international conferences have become a fixture for exchanging experiences and knowledge about HAT technological related aspects. PRAHA-II has helped to spread the awareness of HAT challenges in optimization and risk assessment as well as opportunities.

Key take-home messages from PRAHA-II conclusions and recommendations can be illustrated as below.



Annex I – Examples of Risk Assessment Model Data Tables filled

A. Target Product

	value
model number	
type(cooling / HP)	
capacity(kW)	
refrigerant type	
refrigerant amount(kg)	
alternative refrigerant type	R32?

B-1. Indoor condition during usage of target product

		value
room size (m^2)	max	
	min	
height of installation(m)		
ceiling height(m)		
ventilation	yes/no	
	Ventilation (m^3/hr.)	
gap under door area (m^2)		
other openings, if any (m^2)		

B-2. Outdoor condition during usage of target product

		value
the size of the place enclosed with walls ,or fences etc.(m2)	max	
	min	

(ex. the internal area of a balcony)

C. Condition during repair of target product

	value
the average size of outdoor spaces for repairs (m3)	
the percentage of single outdoor unit installations (A%)	
the percentage of the installations of multiple outdoor units (B%)	
the average working hours per repair (outdoor unit) (hr.)	
the average working hours per repair (indoor unit) (hr.)	
wind condition (wind velocity) (m/s)	
windless condition (percentage %)	
(Windless condition; 0.1m/s or less. the windless rate in one year.)	

(note1)A+B=100% (note 2) multiple outdoor units installed with a considerable amount of spaces between them is included in the single installation category.

Praha-II List of Possible Ignition Source and estimation of ignition occurrence in Kuwait's case

(during usage - indoor)

Type of Ignition source	Ignition Source		Estimate of ignition occurrence / day: Occurrence (times/day)	Duration (hours/day)
Ignition source caused within flammable region (triggered by the ignition source)	open flame	cigarette		
	Electric spark (human conduct)	oil lighter		
		ignition switch of heater		
		connect / disconnect of electric plug		
		on/off relay within electrical equipment		
		relay operation of electrical equipment		
		brush motor		
	Electric spark (excluding human conduct)	malfunction of equipment		
		slip on / off the clothes		
	Human conduct	slip on / off the clothes		
open flame (triggered by flammable region)	open flame	candle		
		heater		
		stove burner		
		catch fire		
	High temperature surface	Electric heater		

Annex II - Safety

Overview of RACHP safety standards (Source: *TEAP report Volume 4: Decision XXX/5 on Cost and Availability of Low-GWP Technologies/Equipment that Maintain/Enhance Energy Efficiency*)

The requirements and implications of various international and regional safety standards covering RACHP sectors are detailed in report TEAP TF XXVIII/4.[3] This includes a table of relevant standards and the applicable various sub-sectors (Table 2-1). An extract of that table is provided below (Table I). Throughout the report there are discussions on what the upper charge limits are.

Table I: Scope of selected RACHP safety standards that include flammable refrigerants

Sector	Vertical (Product Standards)		Horizontal (Group Standards)
	IEC 60335-2-40	IEC 60335-2-89	ISO 5149-1,-2,-3,-4
Commercial refrigeration		×	×
Air-to-air air conditioners & heat pumps	×		×

Table II attempts to summarise the upper charge limits, where values have been separated into two categories.

- "with limited measures" means only with elimination of potential ignition sources
- "With additional measures" refers to situations where additional protective measures have to be applied, such as imposing a minimum room size, additional ventilation, etc.

It is not straight-forwards to summarise the "with additional measures" charge limits as they often depend upon the choice of several measures, installation conditions and so on. The exercise should be carried out on a case-by-case basis.

Table II: Maximum charge size limits for flammable refrigerants according to RACHP safety standards

	With limited measures			With additional measures		
	A3	A2	A2L	A3	A2	A2L
IEC 60335-2-89	0.15 kg	0.15 kg	0.15 kg	n/a	n/a	n/a
IEC 60335-2-40	0.15 kg	0.5 kg	1.8 kg	0.3 kg/1.0 kg	3.4 kg	8.0 kg/78 kg
ISO 5149	0.15 kg	0.5 kg	1.8 kg	1.5 kg/2.5 kg/ unlimited	3.4 kg/ unlimited	60 kg/ unlimited

All of these standards are in various stages of revision including with special attention to application of flammable refrigerants. Again, a summary of these may be found in the TEAP TF XXVIII/4 report.

Overview of safe refrigerant handling

In terms of refrigerant safe handling training, the situation differs widely amongst countries, due to the variety of national legislation. The IIR has published an information note on qualification and competence of technicians,[4] which offers an overview of schemes available in many countries.

[3] TEAP TASK FORCE Decision XXVIII/4 Report: on safety standards relevant for low-GWP alternatives

[4] http://www.iifiir.org/userfiles/file/publications/notes/NoteTech_28_EN.pdf

Some international and regional standard touch on the topic. An international standard is under preparation, ISO 22712 - Refrigerating systems and heat pumps — Competence of personnel (currently in the form EN 13113), which addresses the required competence of technicians for all refrigerant types and tasks. More specifically, IEC 60335-2-40 includes an Annex (DD) covering requirements for operation, service and installation manuals of appliances using flammable refrigerants, which is essentially a compilation of procedures. Another annex (HH) addresses "Competence of service personnel". Whilst neither IEC 60335-2-89 nor ISO 5149 contains any such material, EN 378-4 does have a short annex on competence of persons working with flammables.

Most countries tend to operate training programmes that are either national or private schemes. There are also a number of regional training programmes in existence, such as the "Real Alternatives" scheme, which covers most of the European countries.[5] In North America there are two such schemes: North America Training Excellence (NATE) for HVAC[6] and AHAM-Home Appliance[7]. China operates a national training scheme for flammables as does JRAIA in Japan.

The entire topic is rather disparate, but it is expected that the global approach will become more harmonised as introduction of flammable refrigerants become more prevalent.

[5] https://www.realalternatives.eu/learning-platform
[6] https://www.natex.org/site/1/Homehttp://
[7] www.aham.org/AHAM/Safety/Safe_Servicing_of_Cold_Appliances/AHAM/Safety/Safe_Servicing_of_Cold_Appliances.aspx?hkey=23d1344d-f8b0-410a-9e21-8181048b2b82

ANNEX-III: AHRTI Final Report

Promoting Alternative Refrigerants in High-Ambient Countries Phase (PRAHA-II):

Optimization Study on PRAHA I Equipment

September 2019

Final Report

AHRTI Report No. 9011

Promoting Alternative Refrigerants in High-Ambient Countries Phase II (PRAHA-II): Optimization Study on PRAHA I Equipment

Final Report

September 2019

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DISCLAIMER

TABLE OF CONTENTS

1. Executive Summary 5
2. Introduction 9
3. Activity 1 - Analyzing the Design of PRAHA-I Prototypes 9
 3.1. Physical Units 9
 3.2. PRAHA-I Performance Reports Assessment 10
 3.3. Hardware Improvement Assessment 11
  3.3.1. Heat Exchanger (HX) First Order Analysis (FOA) 11
  3.3.2. Compressors 12
  3.3.3. Expansion Devices 12
  3.3.4. Fan and Blower 12
  3.3.5. Units Component Modification Potential 12
 3.4. Conclusions and Recommendations 12
4. Activity 2 - Design Improvements 13
 4.1. Hardware 13
 4.2. Refrigerant 13
 4.3. System Design Modification Framework 14
  4.3.1. Compressors 16
  4.3.2. HX Design and Selection 17
  4.3.3. System Design 19
 4.4. Modified Systems Results Summary 19
 4.5. Conclusions and Recommendations 19
5. Activities 3, 4 & 5 - Prototype Units Fabrication, Evaluation of the Optimized Prototypes and Analyzing Leaks of Alternatives 21
 5.1. Unit 6 21
 5.2. Unit 10 22
 5.3. Leak Tests 23
 5.4. Conclusions and Recommendations 24
6. Conclusions 25
 APPENDIX A - Activity 2 Design Framework Results 27
 APPENDIX B – Unit 6 Initial Tests, Scope Change and Test Setup 36
 APPENDIX C - Unit 6 Raw and Processed Tested Data 39
 APPENDIX D - Unit 10 Baseline Re-Test 42
 APPENDIX E - Unit 10 Raw and Processed Tested Data 42
 APPENDIX F - Model Verification and Validation 47

LIST OF FIGURES

Figure 1. Heat Exchangers FOA. 11
Figure 2. Refrigerants Investigated for Units 1 and 4. 14
Figure 3. Refrigerants Investigated for Units 6 and 10. 14
Figure 4. Refrigerant Temperature Glides. 14
Figure 5. Illustrative T-s diagram for baseline and improved cycles. 15
Figure 6. P-h Diagrams Illustrating COP Improvement: a) Real Cycle; b) Ideal Cycle (Carnot). 16
Figure 7. System Re-Design Framework, ... 16
Figure 8. Condenser Forms: Unit 1 (left), Unit 10 (center), Unit 6 Cabinet (right). 18
Figure 9. HX Form Examples: L-shape (left), Flat (right). 18
Figure 10. Unit 1 – Modified Systems P-h Diagrams. 28
Figure 11. Unit 4 – Modified Systems P-h Diagrams. 29
Figure 12. Unit 6 – System Level Analysis: Performance Results for R32. 32
Figure 13. Unit 6 – System Level Analysis: Performance Results for R454B. 32
Figure 14. Unit 6 - Comparative System Performance Summary for R452B. 32
Figure 15. Unit 10 – System Level Analysis: Performance Results for R452B. 35
Figure 16. Unit 10 – System Level Analysis: Performance Results for R447B. 35
Figure 17. Unit 10 – System Level Analysis: Performance Results for R454B. 35
Figure 18. Hot Water Thermal Imaging. 36
Figure 19. Test Diagram. 37
Figure 20. OTS Setup: outdoor chamber (left), Unit 10 and frequency converter inside chamber (right). 37
Figure 21. OTS Setup: indoor closed loop left side view (left), refrigerant mass flow meter (right). 38
Figure 22. OTS Setup: indoor closed loop right side view (left), vapor / liquid lines, sight glasses and TXV (right). 38
Figure 23. Unit 6 – R32 Performance Test Summary P-h and T-s Diagrams. 47
Figure 24. Unit 6 – R454B Performance Test Summary P-h and T-s Diagrams. 48
Figure 25. Unit 10 – R447B P-h and T-s Diagrams. 48
Figure 26. Unit 10 – R452B P-h and T-s Diagrams. 49

LIST OF TABLES

Table 1: Activity 2 Summary Modeling Results. 6
Table 2: Tests Summary Results. 7
Table 3: Unit Specifications Summary. 10
Table 4: Units 1, 4, 6 and 10 Components. 12
Table 5: Compressor Models. 17
Table 6: Activity 2 Results. 20
Table 7: Unit 6 Modifications for Testing. 21
Table 8: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 35°C. 22
Table 9: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 46°C. 22
Table 10: Unit 10 Modifications for Testing. 23
Table 8: Unit 10 - Performance Test Summary for R32 Baseline @ 35°C. 23
Table 9: Unit 10 - Performance Test Summary for R32 Baseline @ 46°C. 23
Table 12: Unit 10 – R447B Leak Test Summary Results. 24
Table 13: Unit 1 – Theoretical Cycle Re-Design Summary. 27
Table 14: Unit 1 – HX Analysis Summary 27
Table 15: Unit 1 – Compressor Performance Summary. 27

Table 16: Unit 1 – Expected Modified System Performances. 27
Table 17: Unit 4 – Theoretical Cycle Re-Design Summary. 28
Table 18: Unit 4 – HX Analysis Summary. 28
Table 19: Unit 4 – Compressor Performance Summary. 29
Table 20: Unit 4 – Expected Modified System Performances. 29
Table 21: Unit 6 – Theoretical Cycle Re-Design Summary. 30
Table 22: Unit 6 – HX Analysis for R32 30
Table 23: Unit 6 – HX Analysis for R452B. 30
Table 24: Unit 6 – HX Analysis for R447B. 31
Table 25: Unit 6 – HX Analysis for R454B. 31
Table 26: Unit 6 – Compressor Performance Summary. 31
Table 27: Unit 10 – Theoretical Cycle Re-Design Summary. 33
Table 28: Unit 10 – HX Analysis for R32 33
Table 29: Unit 10 – HX Analysis for R452B. 33
Table 30: Unit 10 – HX Analysis for R447B. 34
Table 31: Unit 10 – HX Analysis for R454B. 34
Table 32. Unit 10 - Compressor Performance Summary. 34
Table 33: Test Scope Change. 37
Table 34: List of Measurements. 39
Table 35: Unit 6 – Performance Tests 39
Table 36: Unit 10 – Performance Tests. 42
Table 37: Unit 10 – R447B Leak Tests 44
Table 38: Unit 6 – Model Verification and Validation for Alternative 1 – R32 @ 46°C. 47
Table 39: Unit 6 – Model Verification and Validation for Alternative 2 – R454B @ 46°C. 47
Table 40: Unit 10 – Model Verification and Validation for Alternative 1 – R447B @ 46°C. 48
Table 41: Unit 10 – Model Verification and Validation for Alternative 2 – R452B @ 46°C. 48

1. Executive Summary

Over the past several years through the Promoting low- Global Warming Potential (GWP) Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA-I) project, 18 different prototypes have been developed and compared to respective baselines to support the assessment of alternative lower-GWP refrigerants for air-conditioning applications. Since the work originally started in 2012, researchers have identified gaps in the performance and operation of the PRAHA-1 prototypes. These gaps include the need to redesign and optimize prototype air-conditioning units, evaluate new alternative refrigerants, and improve component selection. As such, a new project, *Advancing the Designs of PRAHA-I for Meeting or Exceeding the Baseline Designs Performance*, conducted by Optimized Thermal Systems, Inc. (OTS) is herein presented.

The objectives of this project include the following:

1) Evaluate the design limitation of the PRAHA-I prototypes;
2) Optimize and physically evaluate selected prototypes with new refrigerants not evaluated during PRAHA-I; and,
3) Assess potential refrigerant fractionation impact due to leakage.

The project was organized into six activities for which a summary of the results, conclusions and recommendations are presented below:

1) Activity 1: Analyzing the Design of PRAHA-I Prototypes
    a. Certification laboratories, such as the one used for testing the units in PRAHA I, provide limited information for the purposes of product design and development. For future reference it is recommended that for research-oriented efforts such as this one, the units undergo a more rigorous testing process along with full characterization of the system and its individual components operating conditions and performance.
    b. In applications of high ambient temperatures, it is expected that performance will degrade as compared to operating under more temperate conditions and the resultant impact on performance must be considered. The key components for performance improvement identified herein were the compressor, condenser and expansion device.
        i. At higher temperatures, the saturation temperatures and refrigerant density at compressor's suction port can be very different than that from the rated conditions. Larger displacement volumes and efficiency curves optimized for higher pressure lifts might be required. Therefore, the proper selection of the compressor is paramount.
        ii. A better performance condenser will reduce the approach temperature between refrigerant and air, helping the compressor not to discharge refrigerant at very high pressure and temperatures, which degrade performance.
    c. At high ambient conditions, the system is forced to operate in higher pressure lift than at rated conditions, but still requires a certain refrigerant mass flow rate. Passive devices such as capillary tubes and orifices may not be able to provide enough expansion to allow the system to operate in higher temperature conditions. An active expansion device such as EXV's can adequately control operating conditions and maintain stable superheat.

2) Activity 2: Design Improvements (Summary results in Table 1)
    a. R290 and R32 have wider saturation regions allowing the system to operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer. Their cycles

may get closer to that of the ideal Carnot cycle compared to refrigerants with narrower saturation.

b. Refrigerants with high temperature glide may require new heat exchanger (HX) designs, namely condensers. The original designs proved to be sufficiently effective to allow for most systems to operate with the different refrigerants, however, better designs would allow for higher system efficiency and potentially less charge. HX designs are severely constrained by allowed envelope dimensions. A complete system re-design would provide an opportunity for designing HX's with even higher efficiency.

c. The results of this analysis suggest that for an effective refrigerant replacement, a proper compressor selection must be accompanied with it. Higher isentropic efficiencies are desired for higher temperatures, but most importantly, the displacement volume requirements can vary considerably from one refrigerant to another.

d. It is also imperative that having an active expansion device (preferably an Electronic Expansion valve (EXV)) to not only allow for more controlled superheat, but also to enable the unit to run with different refrigerants with very different thermophysical properties.

Table 1: Activity 2 Summary Modeling Results.

General Information			Hardware				Performance	
System	**Rated Capacity (@35°C)**	**System Configuration**	**Compressor**	**Condenser**	**Expansion Device**	**Ref.**	**Cooling Capacity (@46°C)**	**EER (@46°C)**
-	**BTU/hr**	**-**	**Efficiency (-)**	**Type**	**Type**	**-**	**BTU/hr**	**BTU/hr.W**
Unit 1	18000	**Baseline**	**0.66**	**Tube-Fin (5mm Tube)**	**Passive**	**R444B**	**17403**	**7.4**
		Alternate 1	0.7	Same as Baseline	Active (EXV)	R290	17639	8.01
		Alternate 2	0.69			R454C	18104	7.31
		Alternate 3	0.7			R444B	18140	8.14
		Alternate 4	0.68	MCHX		R457A	17749	7.63
Unit 4	24000	**Baseline**	**0.61**	**Tube-Fin (9.5mm Tube)**	**Passive**	**R290**	**17940**	**7.52**
		Alternate 1	0.7	Tube-Fin (5mm Tube)	Active (EXV)	R290	18147	9.12
		Alternate 2	0.7			R290	24120	6.72
Unit 6	24000	**Baseline**	**0.6**	**Tube-Fin (7mm Tube)**	**Passive**	**R32**	**23115**	**8.46**
		Alternate 1	0.65	Tube-Fin (5mm Tube)	Active (EXV)	R32	23798	9.41
		Alternate 2	0.67			R454B	22894	9.71
		Alternate 3	0.7			R452B	23702	9.6
Unit 10	36000	**Baseline**	**0.44**	**Tube-Fin (9.5mm Tube)**	**Passive**	**R32**	**29005**	**6.39**
		Alternate 1	0.65	Tube-Fin (5mm Tube)	Active (EXV)	R447B	30478	9.43
		Alternate 2	0.67			R452B	30796	10.27
		Alternate 3	0.67			R454B	30809	10

3) Activities 3-5: Prototype Modification and Testing (Summary results in Table 2)

a. Unit 6 re-tested baseline exhibited similar performance to that found in PRAHA I testing. It should be stressed that the baseline unit by design had its capillary tube located in the outdoor unit. This would cause liquid refrigerant leaving the outdoor unit to flash. The refrigerant enthalpy at the condenser outlet state was used to calculate the refrigerant-side capacity assuming an isentalphic expansion without heat loss in connecting pipe. This is different from the modified systems of which the capillary tube was removed, and a manual expansion valve was placed at the inlet of the indoor unit. For modified systems, the enthalpy at the expansion valve inlet was used to calculate the refrigerant-side capacity.

b. Unit 10 exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This supports the hypothesis of low compressor efficiency during PRAHA I tests, which also indicates the importance of proper compressor selection.
c. The above is also evidenced by the fact that even with R447B and R452B (zeotropic mixtures), Unit 10 had higher cooling capacity and efficiency than the baseline for the 46°C test condition, as projected in activity 2. The tests at 35°C, however, did not have the same trend.
d. The impact of refrigerant replacement was not very clear, in part due to the hardware change along with it. But because of the differences in saturation curves from the Activity 2 analysis, R32 tends to result in systems with higher efficiency and less charge. The zeotropic mixtures consistently required compressors with larger displacement volumes and even higher mass flow rates for cooling capacities of the same magnitude.
e. Refrigerant fractionation as evidenced by the leak tests, does not appear to a great concern since less than 2% in cooling capacity was observed after the system's re-charge.
f. The Unit 6 modified systems had lower performance than expected from the Activity 2 models. The R32 system configuration exhibited around 10% less flow rate than anticipated, which corresponded to 10% lower capacity. The R454B configuration exhibited a deviation of 5% between model and test due also in part to a 3% flow rate over prediction in the model. Unit 10, on the other hand, exhibited an excellent agreement to the models with less than 2% deviation in cooling capacity.
g. The model's validation adds confidence in the numerical simulation findings and recommendations provided in activity 2.

Table 2: Tests Summary Results.

Syst.	Test	Refrigerant	Charge	35°C			46°C		
				Cooling Capacity	Total Power	EER	Cooling Capacity	Total Power	EER
			lb	BTU/hr	kW	BTU/hr. W	BTU/hr	kW	BTU/hr. W
Unit 6	Perfor mance	R32 (Baseline)	3.83	25192	2.43	10.4	23390	3.10	7.54
		R32 (Alternate 1)	4.27	23585	2.12	11.1	21450	2.74	7.84
		R454B (Alternate 2)	5.02	21966	2.10	10.4	21821	2.67	8.17
Unit 10	Perfor mance	R32 (Baseline)*	5.63	34517	3.76	9.18	29005	3.84	7.55
		R447B (Alternate 1)	6.63	32195	3.62	8.88	31073	3.90	7.96
		R452B (Alternate 2)	6.63	28128	3.38	8.33	30292	3.90	7.76
	Liquid Line	Low Charge	4.23	N/A			14216	3.90	3.64
		Re-Charged	6.63				30865	4.16	7.42
	Vapor Line	Low Charge	4.27				15171	3.92	3.87
		Re-Charged	6.77				30587	-	-

**Original baseline values from PRAHA*

4) Conclusions: This report presented a comprehensive set of activities with the objectives of advancing the PRAHA program. The original scope and schedule were modified during the project as new findings and challenges surfaced. The tests that were carried out for PRAHA-I, while sufficient for the purpose of measuring capacity and energy efficiency for the purposes of PRAHA-I, did not have enough essential data to enable a complete cycle evaluation for optimization purposes. This is primarily due to using standard test rig on systems with critical hardware configuration differences. The analyses presented in Activity 2 (design assessment through modeling) provided good insights on adequate component design and/or selection for proper system functioning when using novel refrigerants. The tests in activities 3-5 partially served as validation for the models developed, and as check for previous test data from PRAHA I. The final recommendations for future development are listed as follows:
    a. Establish a baseline system by conducting comprehensive testing including measurements and metrics not typically performed in energy certification tests. Furthermore, testing systems with different configurations require custom test rigs as such to adequately measure working fluid's states to avoid mischaracterization of the operating conditions and performance. Such approach is considerably more labor-intensive which should be factored in the scope in future developments.
    b. Using alternate low-GWP refrigerants is viable and can be competitive to commonly used pure refrigerants but doing so requires proper component design and selection; compressor and expansion device particularly. Drop-in replacement without hardware change is never recommended as evidenced by the change requirements in Activity 2 and performance tests in the subsequent activities.
    c. It is recommended to always perform numerical simulations, and to conduct at least some level of "soft" optimization analyses that will provide information for an educated system re-design / retrofit at much lower costs than gradual trial-and-error changes.
    d. Always test the modified systems with the same instrumentation as the baseline, however mindful of the modifications as such to properly place sensors to obtain adequate readings as suggested in item a above.

2. Introduction

Over the past several years through the Promoting low- Global Warming Potential (GWP) Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA-I) project, 18 different prototypes have been developed and compared to respective baselines to support the assessment of alternative lower-GWP refrigerants for air-conditioning applications. Since the work originally started in 2012, researchers have identified gaps in the performance and operation of the PRAHA-1 prototypes. These gaps include the need to redesign and optimize prototype air-conditioning units, evaluate new alternative refrigerants, and improve component selection. As such, a new project, *Advancing the Designs of PRAHA-I for Meeting or Exceeding the Baseline Designs Performance*, is desired.

The objectives of this project include the following:

4) Evaluate the design limitation of the PRAHA-I prototypes;
5) Optimize and physically evaluate selected prototypes with new refrigerants not evaluated during PRAHA-I; and,
6) Characterize leaks.

The project is divided into six activities namely:

- **Activity 1 – Analyzing the Design of PRAHA-I Prototypes**: evaluate systems performance from selected units tested in PRAHA-I, and assess potential design improvements
- **Activity 2 – Design Improvement**: improve design of specific units targeting higher efficiencies while using alternate low-GWP refrigerants
- **Activity 3 - Prototype Units Fabrication**: modify the a sub-set of the units according to modifications proposed in Activity 2
- **Activity 4 - Evaluation of the Optimized Prototypes**: conduct performance tests on modified units at standard and high ambient temperature conditions (35°C and 46°C)
- **Activity 5 - Analyzing Leaks of Alternatives**: simulate refrigerant leakage and evaluate possible impact of zeotropic mixtures fractionation on performance
- **Activity 6 - Reporting and Data Management**: simulation and test data processing, preparing progress and final reports

3. Activity 1 - Analyzing the Design of PRAHA-I Prototypes

Activity 1 was comprised of three major tasks including: reception of 12 physical units at the OTS facility followed by visual inspection and parts identification; review of performance test reports from PRAHA I tests; and lastly, analyze data and identify, for units of interest, opportunity for improvement targeting higher performance and minimal charge. OTS has completed this activity and an executive summary of the findings are presented herein.

3.1. Physical Units

All 12 units of interest to this project (Table 3) were received on November 8th, 2018. Visual inspection indicated no evident signs of damage. Relevant information to the project such as compressor model, heat exchanger (HX) geometry and circuiting, as well as expansion device were also received.

Table 3: Unit Specifications Summary.

Category	Unit #	Ref.	Designed Capacity Btu/h	Measured Cap. Btu/h	Voltage	Ref. (New designs)	Ref. (Tests)
Window	**1**	**L-20 (R-444B)**	**18,000**	**19,104**	**208-230/60/1**	**L-20, R454C, R290, R457A**	**R290**
	2	L-20 (R-444B)	18,000	16,924	208-230/60/1		
	3	DR-3 (R-454C)	18,000	18,063	208-230/60/1		
Decorative splits	**4**	**R-290**	**24000 (18,000)**	**19,000**	**208-230/60/1**	**R-290**	**R-290**
	5	R-32	24000 (18,000)	19,328	208-230/60/1		
	6	**R-32**	**24,000**	**25,456**	**208-230/60/1**	**R32, R459A**	**R32, R459A**
	7	L-41 (R-447A)	24,000	24,830	208-230/60/1		
	8	L-20 (R-444B)	24,000	22,740	208-230/60/1		
	9	DR-3	24,000	14,638	208-230/60/1		
Ducted splits	**10**	**R-32**	**36,000**	**35,500**	**220-240/50/1**	**R447B, R452B**	**R447B, R452B**
	11	L-20	36,000	36,553	220-240/50/1		
	12	DR-3 (R-454C)	36,000	33,032	220-240/50/1		

3.2. PRAHA-I Performance Reports Assessment

OTS received a complete package of files containing the performance reports for all units tested in PRAHA I. The tests conducted in PRAHA I were meant to assess high-level performance of these units focusing on a large control volume where only total energy in and out was evaluated. As such, these tests were not comprehensive in terms of measurements for cycle analysis required in PRAHA II. Refrigerant side measurements, in most cases, were very limited (few pressure and temperature measurements and no flow rates); thus, it is not possible to fully characterize the cycle and perform energy balances between air and refrigerant sides of the system. Common issues found in the reports include:

- Tag mislabeling and / or mismatching sensor location and tag
- No independent outdoor capacity reported – typically reported the same as indoor capacity
- Missing energy balance checks
- Missing measurement on either airside pressure drop and temperature or fan power
- Inconsistent reported measurements with thermophysical properties for units tested with L-20
- Systematic inconsistency in reported superheat and subcooling
- Missing measurements on refrigerant side at evaporator inlet
- Missing temperature and/or pressure measurements on refrigerant side
- Missing refrigerant mass flow measurements

A summary of the original PRAHA-1 data and results of the data reduction are provided under separate documentation.

3.3. Hardware Improvement Assessment

3.3.1. Heat Exchanger (HX) First Order Analysis (FOA)

This section outlines a FOA for the HXs of Units 1, 4, 6 and 10 to identify improvement potential. The project's objective, as stated above, is to improve performance while minimizing charge. One way of addressing both objectives is by reducing tube / channel diameter. Heat transfer coefficients are inversely proportional to surface hydraulic diameters, however, so is pressure drop. Smaller tubes result in more compact (C = surface area / footprint volume), with reduced internal volume, HXs.

A qualitative analysis using values from literature was carried out to demonstrate the relative impact of diameter over abovementioned metrics, specifically: heat transfer coefficient, compactness and overall thermal conductance (UA). The left-hand side plot in Figure 1 show three curves inversely proportional to the diameter; a 5mm tube can achieve, in this example, 70% greater UA than a conventional 9.5mm, within the same footprint volume (or cabinet).

These are further explored to illustrate the impact on a system level. Systems respond to UA of both condenser and evaporators, but for the purposes of this analysis, condenser only is considered. The UA represents the overall thermal conductance, which will impact the approach temperatures in the system (ΔT_{app}). If the heat rejection is kept constant, the higher the UA, the smaller are the ΔT_{app}'s, thus allowing the condenser to operate in lower pressure levels, which will consequently increase the system performance. An example using a hypothetical R32 cycle with an EER of 12 as base is shown in the right-hand side plot in Figure 1. Performance improvement is limited by the Second Law, when the approach temperatures near zero; however, in this illustration, the EER has potential to increase in over 20% with better condenser design alone.

It is imperative to note that the results presented in this section are for **illustration purposes only**. Further in this report it is presented in more detail a re-design framework, applied to the units of interest in this project, using the metrics outlined in this section.
Unit 1 already had a 5mm condenser, which limits the options for HX re-design. Unit 6 had a 7mm HX on both the indoor and outdoor units, which allows some room for improvement if reducing to 5mm. Lastly, both coils for Unit 10 had 9.5mm tubes, thus there is greater potential for charge reduction and performance improvement for that unit in particular.

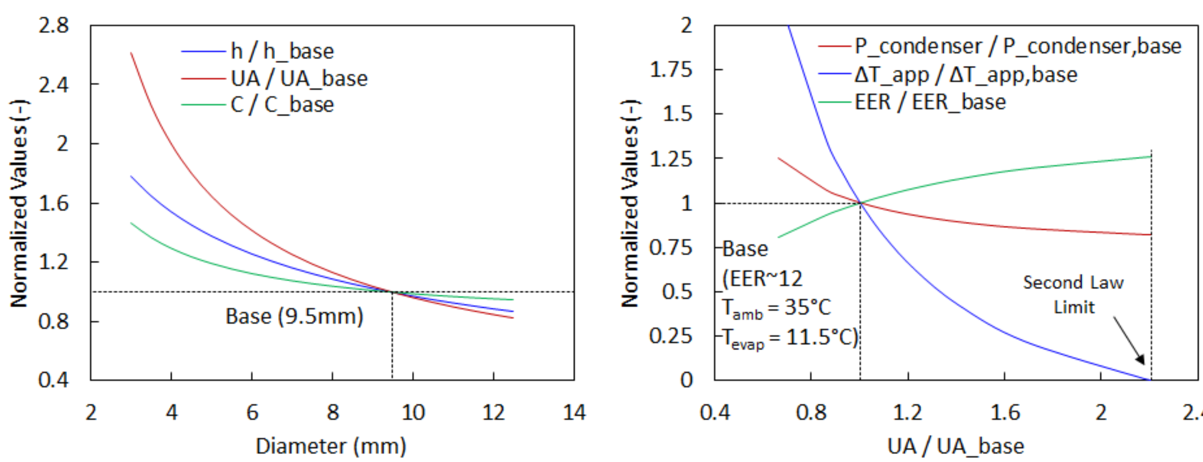

Figure 1. Heat Exchangers FOA.

3.3.2. Compressors

The existing units mostly use compressors sized specifically for R410A or R22 and in some cases custom made for this effort. There is, however, opportunity for a better compressor selection when migrating from R32 to R454B or R447B on Units 6 and 10, respectively.

3.3.3. Expansion Devices

Expansion devices such as TXV's and EXV's may allow for better control and reduced losses in connecting pipes if located near the evaporator. Some units, such as 6 and 10, have a capillary tube in the outdoor unit, which forces the refrigerant to travel in two-phase along the connecting pipes, and at lower temperatures, thus increasing pressure drop and heat gain.

3.3.4. Fan and Blower

Replacing the fan and blower may be necessary if newly designed HXs offer considerable change in pressure drop over the baseline since the flow rates are kept constant. The lack of test data on pressure drop forces us to rely on predicted values only. These will be considered for replacement as a last priority.

3.3.5. Units Component Modification Potential

Table 4 shows the detailed existing components for the units of interest for modification.

Table 4: Units 1, 4, 6 and 10 Components.

System	Unit 1	Unit 4	Unit 6	Unit 10
Refrigerant	**R444B**	**R290**	**R32**	**R32**
Compressor	HIGHLY SL260DG-C8EU	HIGHLY PSH356DG-C8DU3	GMCC KSG226N1UMT	Copeland ZP42K5E-PFJ-XXX
Condenser	5mm Louver TFHX	9.5mm Wavy TFHX	7mm Louver TFHX	9.5mm Louver TFHX
Expansion Device	Capillary Tube	Capillary Tube	Capillary Tube	Capillary Tube
Evaporator	9.5mm Louver TFHX	7mm Louver TFHX	7mm Slit TFHX	9.5mm Louver TFHX

3.4. Conclusions and Recommendations

The first part of this activity regarded data analysis and processing from the original tests conducted in the original PRAHA-I project, which was designed to conduct testing and comparison of cooling capacity vs. EER for the prototypes against the baseline units from same manufacturers. Since limited certification tests were required then, more testing parameters would have been needed to support the optimization and/or redesign process within the scope of PRAHA-II. The second part pertained assessing potential hardware modifications that could result in higher performance and less charge, with the intent of replacing the original refrigerants with alternative, low-GWP ones. The key conclusions and recommendations are:

1- Certification laboratories, such as the one used for testing the units in PRAHA I, provide limited information for the purposes of product design and development. For future reference it is recommended that for research-oriented efforts such as this one, the units undergo a more rigorous testing process along with full characterization of the system and its individual components operating conditions and performance.

2- In applications of high ambient temperatures, it is expected that performance will degrade as compared to operating under more temperate conditions and the resultant impact on performance must be considered. The key components for performance improvement identified herein were the compressor, condenser and expansion device.

a. At higher temperatures, the saturation temperatures and refrigerant density at compressor's suction port can be very different than that from the rated conditions. Larger displacement volumes and efficiency curves optimized for higher pressure lifts might be required. Therefore, the proper selection of the compressor is paramount.
b. A better performance condenser will reduce the approach temperature between refrigerant and air, helping the compressor not to discharge refrigerant at very high pressure and temperatures, which degrade performance.

3- At high ambient conditions, the system is forced to operate in higher pressure lift than at rated conditions, but still requires a certain refrigerant mass flow rate. Passive devices such as capillary tubes and orifices may not be able to provide enough expansion to allow the system to operate in higher temperature conditions. An active expansion device such as EXV's can adequately control operating conditions and maintain stable superheat.

4. Activity 2 - Design Improvements

The details of modeling and simulation results are provided in a separate document submitted in conjunction with this one, while in this section only the summarized performance results are presented.

4.1. Hardware

A general design improvement assessment was presented in the report for Activity 1, focusing on the units of interest to this study. A first order analysis on the HX's showed that moving towards smaller hydraulic diameter tubes can be beneficial from a material savings and charge reduction standpoint. Units 4 and 10 use conventional 9.5mm diameter tube condensers (Table 4), making them good candidates for condenser replacement with either a smaller tube diameter or a microchannel heat exchanger (MCHX). The compressors used on Units 1, 4 and 6 do not have available performance maps making it difficult to assess their fitness for the system. The focus of this study is on proper compressor selection and condenser re-design.

4.2. Refrigerant

R32 and R290 have wide saturation regions (Figure 2 and Figure 3) putting them at an advantage since they may operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer. Their cycles may get closer to that of the ideal Carnot cycle compared to refrigerants with narrower saturation.

Amongst the blends investigated for Unit 1, R444B has the widest saturation region while also having the highest temperature glide (Figure 4). The latter is typically not beneficial, in particular for evaporators, but it may help the condenser. The glide enables the refrigerant temperature profile to get closer to the air temperature profile without crossing (Figure 4). From a thermodynamic perspective, this means R444B can have its condensing pressure reduced further, resulting in higher theoretical COP.

For Units 6 and 10, the investigated blends, although having narrower saturation than the baseline R32, have similar thermophysical characteristics (Figure 3) with lower temperature glides (Figure 4) making them more competitive from a capacity and performance perspective.

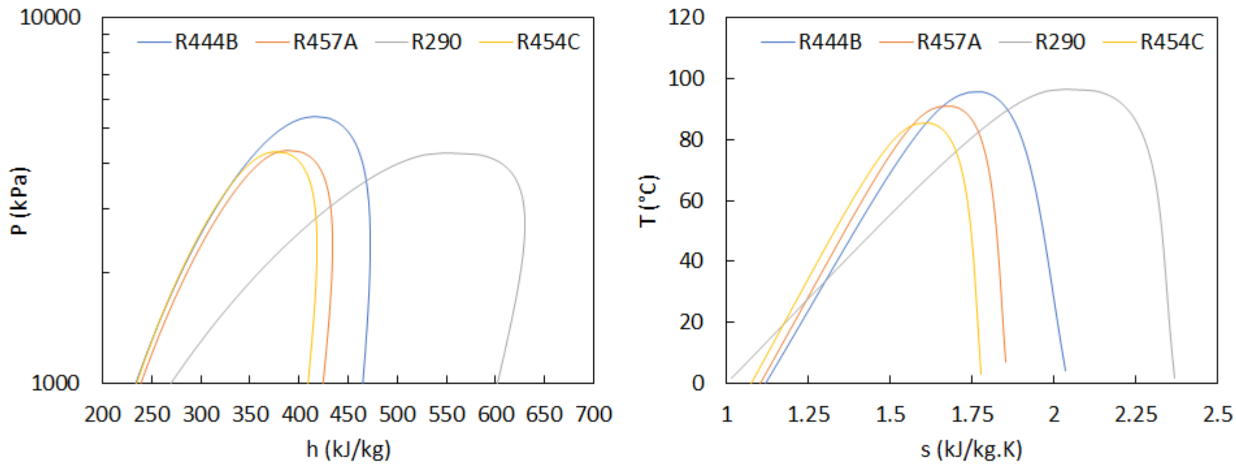

Figure 2. Refrigerants Investigated for Units 1 and 4.

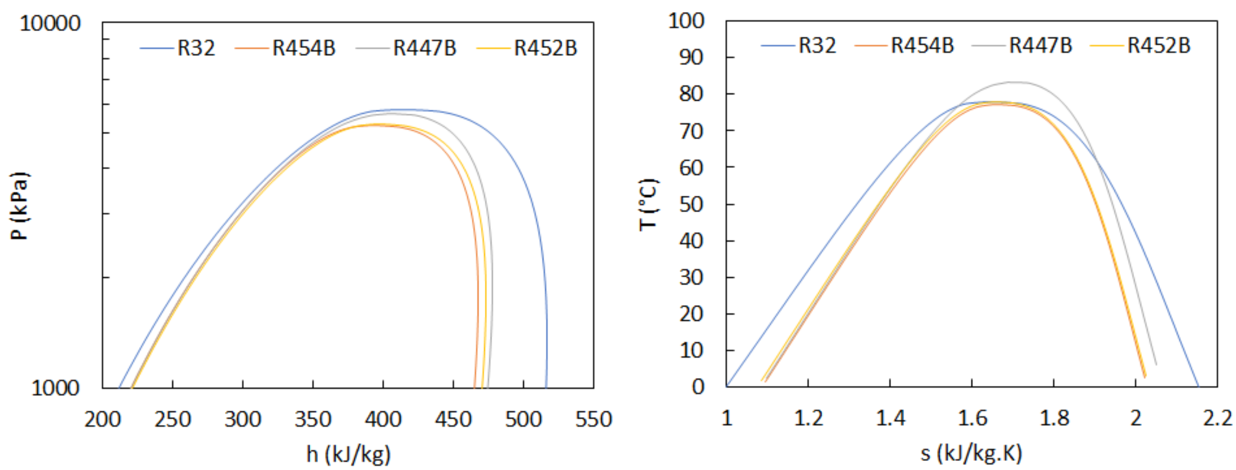

Figure 3. Refrigerants Investigated for Units 6 and 10.

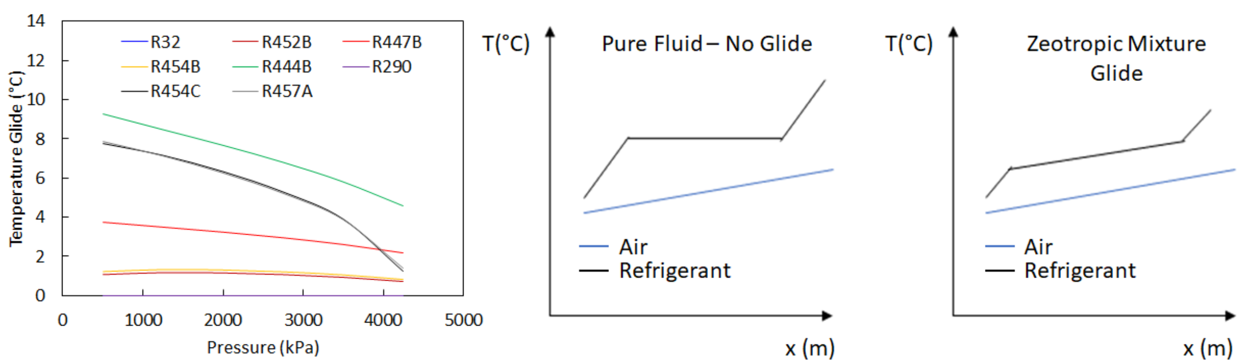

Figure 4. Refrigerant Temperature Glides.

4.3. System Design Modification Framework

The systems' re-design herein presented ultimately consists of a retrofit of the existing units by properly designing and selecting components that can be replaced as drop-ins, with minimal or no modification of the packaging (cabinets). In other words, any component replaced must occupy the same envelope as the baseline component. The focus of the re-design is on:

- Compressor
- Condenser, and
- Expansion valve

The evaporator designs were not changed for two main reasons: a) some are custom-made wrap-around the blower units, such as in Unit 6, making it harder to quickly find an off-the-shelf option; and, b) the goal is to deliver the same cooling capacity while improving efficiency. For the latter, there's more room for improvement in the condenser by reducing condensing pressure, assuming the evaporator can already deliver the expected capacity.

The fans and blowers were also not considered for change, in part due to the lack of information on the performance curves from the baseline models, but also due to potential high cost and lead time for replacement with secondary impact on performance since 80-90% of the power consumed comes from the compressor.

The first step to assess the level of performance required for each component is to investigate an improved theoretical cycle, which will indicate how much COP improvement can be expected, as well as refrigerant flow rate needs and HX size (UA). To improve the performance of a vapor compression cycle, the pressure lift between evaporating and condensing pressures must be reduced. Consequently, the approach temperatures between air and refrigerant will be reduced as well (Figure 5), thus the thermal capacitance of the heat exchangers must increase. Furthermore, the closer to the saturation region, the closer the cycle reaches the ideal Carnot efficiency (Figure 6).

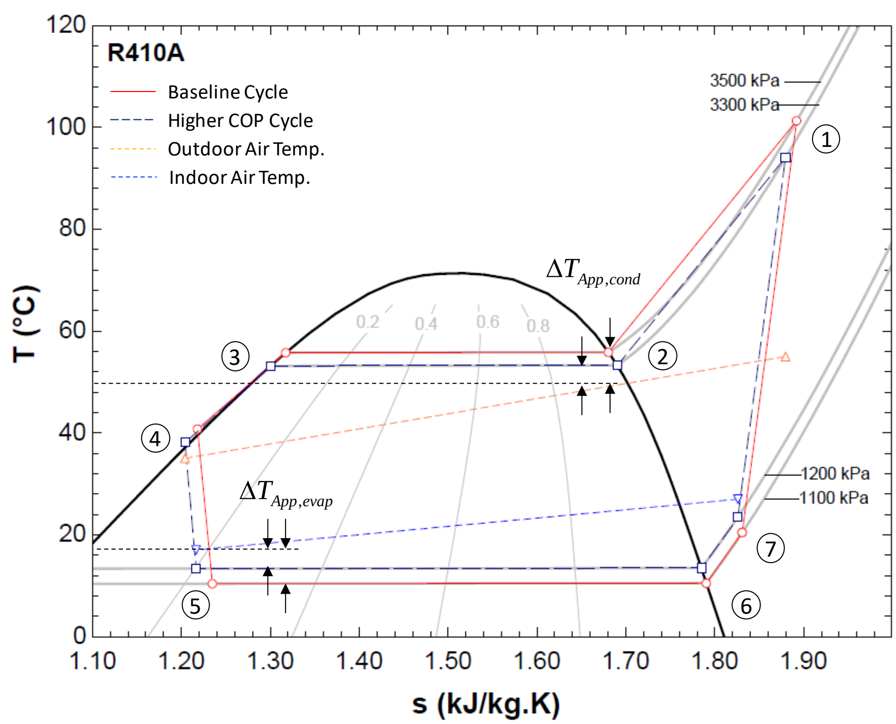

Figure 5. Illustrative T-s diagram for baseline and improved cycles.

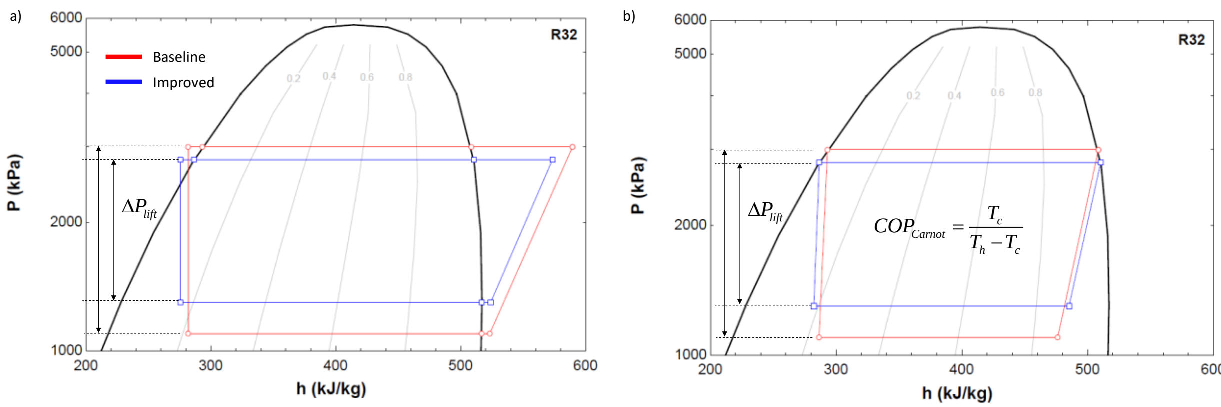

Figure 6. P-h Diagrams Illustrating COP Improvement: a) Real Cycle; b) Ideal Cycle (Carnot).

The system design framework is performed according to Figure 7.

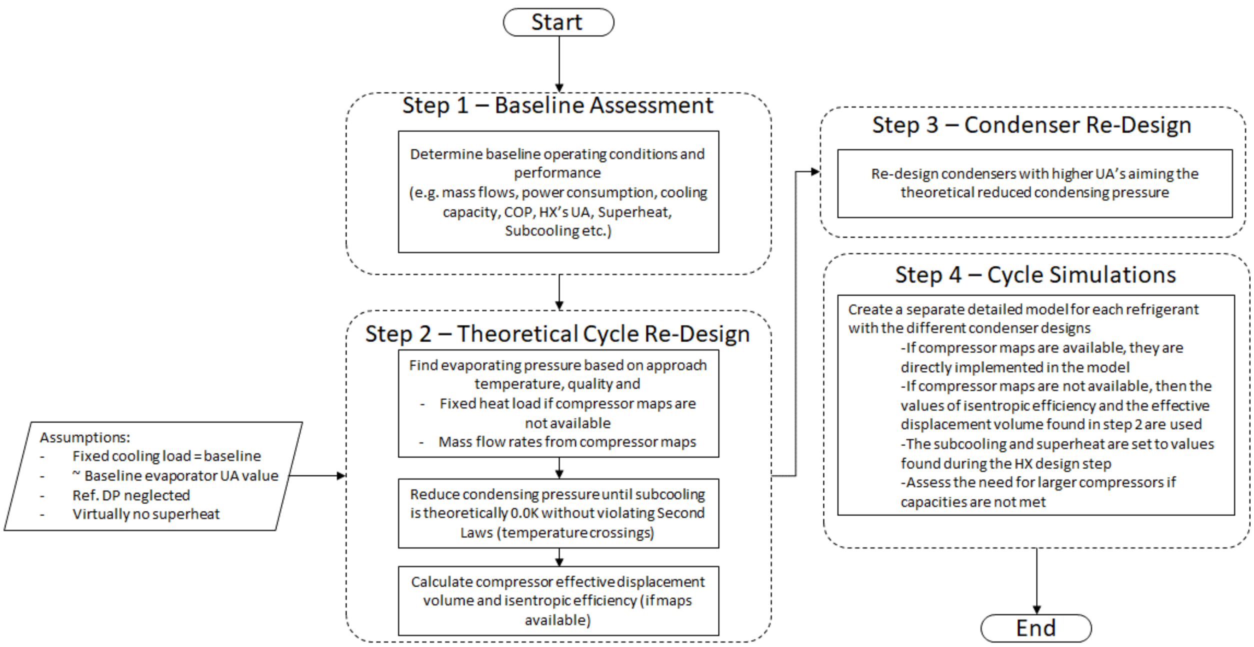

Figure 7. System Re-Design Framework,

4.3.1. Compressors

Modeling compressors are handled in two possible ways, as suggested previously: using performance maps when available or using fixed isentropic efficiency and effective displacement volume. For the larger capacity units (6 and 10), performance maps were provided. Although these compressors were originally designed for R410A refrigerant they may operate – not necessarily optimally – with other refrigerants. Compressor manufacturers supporting this project used proprietary simulation tools, with aid from available empirical data (tests with other refrigerants), to develop theoretical maps for the various refrigerants of interest (Table 5) and made them available to OTS for modeling purposes. It is understood that the predictions are for reference only, and the compressor manufacturer does not guarantee performance for any refrigerants for which the compressors haven't been fully tested.

Table 5: Compressor Models.

Model	Capacity (BTU/hr)	Frequency (Hz)	Refrigerants
ZP20K5E-PFV	24,000	60	R32, R459A, R454B, R410A
ZP21K5E-PFV	24,000		
ZP31K6E-PFV	36,000	50/60	R447B, R452B, R454B, R410A
ZP34K6E-PFV	36,000		

For the smaller units (1 and 4), which were re-designed using R290 (Propane), compressor performance maps were not available. The approach for these units then was to set a target isentropic efficiency of 0.7 (baseline data suggests that the compressor efficiencies ranged from 0.55 to 0.65). The required mass flow rate is calculated based on capacity in the theoretical cycle model described above. From there, the effective displacement volume can be determined (eq. (1))[1]. The latter serves to determine whether a system can use the same compressors for different refrigerants.

$$V_{eff} = \eta_{vol} \cdot V_{disp} = \frac{\dot{m}_{required}}{f \cdot \rho_{suction}} \qquad (1)$$

4.3.2. HX Design and Selection

The condensers design procedure takes into consideration the following:

- **Face area**: baseline face area must be preserved or at most reduced. Furthermore, the aspect ratio must also match that of the baseline so the HX can be drop-in replaced in the same cabinet.
    - Find the number of tube rows and tube length to match as closely as possible to tube face area and aspect ratio

- **Airside pressure drop and flow rate**: the test data from reports contain only air flow rate measurements, while no information on pressure drop is provided. Additionally, the fan performance curves are also not available, which limits the ability to find the exact operating condition. The baseline models provide an estimate prediction for the pressure drop, which is used as reference.

- **Thermal performance**: this step must be iteratively conducted with the previous step, as such for each design change the air flow rate and capacity are evaluated under the new conditions found in the theoretical cycle re-design.
    - Gradually increment the condensing pressure until attainable performance is achieved. This process is done iteratively using the theoretical cycle model, to find new expected operating conditions for evaporating pressure, superheat, subcooling and refrigerant flow rate.

- **HX Form**: as indicated previously, the HX design is constrained by cabinet dimensions as well as form. In the case of units 1 and 4, the condensers are flat coils placed 90° inside the cabinet (Figure 8), which makes it simpler for drop-in replacement as long as new designs have the same overall dimensions. For units 6 and 10, however, the condensers are L-shaped inside the cabinet (Figure 8). Forming coils is widely done, however, for custom coils it may be a challenge, in particular for MCHX. For this reason, the MCHX designs for units 6 and 10 are sized for a full-face area, assuming the coil can be formed, and a second design that is a single flat slab placed in longer side of the "L" shape(Figure 9).

[1] Variable definitions in the Nomenclature list after final conclusions section in this document.

- **HX Name Tag Convention:** for practical purposes, the HX's will be tagged according to the following W XX YY Z
    - **W:** B = Baseline or N = New Design
    - **XX:** TF **=** Tube-Fin or MC = Microchannel
    - **YY:** D# = Tube Diameter or Height
    - **Z:** R = Reduced Face Area
    - **Example:** New Tube Fin Design with 5.0mm diameter with same face area as the baseline → NTFD5

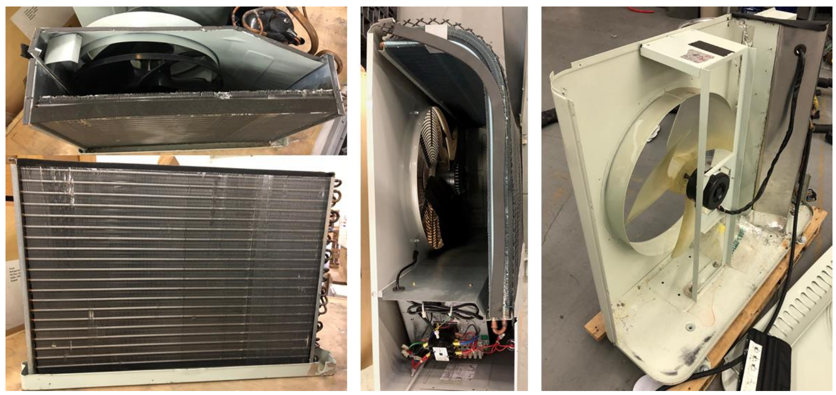

Figure 8. Condenser Forms: Unit 1 (left), Unit 10 (center), Unit 6 Cabinet (right).

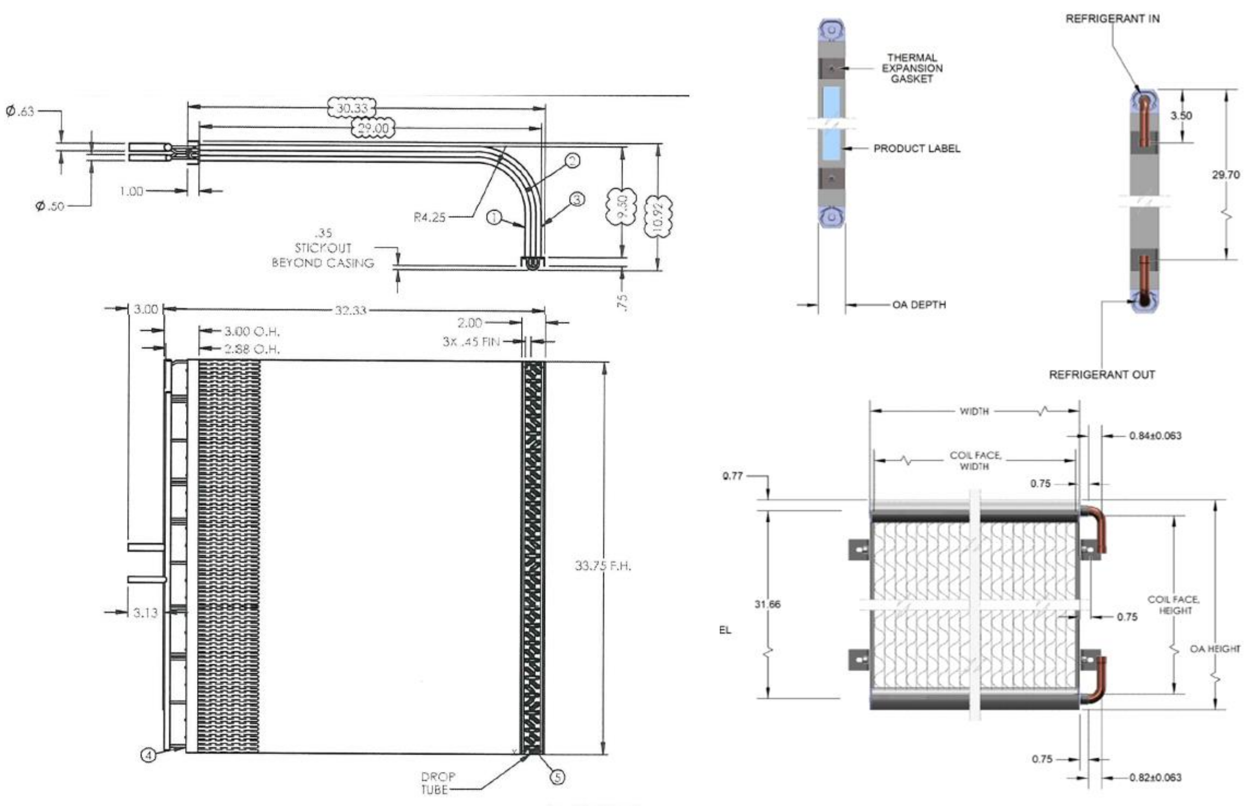

Figure 9. HX Form Examples: L-shape (left), Flat (right).

4.3.3. System Design

In the final step, the modified systems are evaluated holistically through system level modeling and simulation using an in-house Steady-State vapor compression cycle software that has the capability to integrate with the HX and compressor models (performance maps, generic etc.). For each modified system and each refrigerant, a system model was created.

4.4. Modified Systems Results Summary

The final results of Activity 2 are summarized in Table 6. For more detailed results in the framework steps refer to APPENDIX A .

4.5. Conclusions and Recommendations

This section presents a systematic approach based on first order analysis providing educated guidance towards the direction of more efficient systems with fewer simulations and minimal changes to the systems. The study includes a wide variety of refrigerants as well as condenser designs and compressor model options. Given the challenges with original test data the baseline models serve as a numerical reference only. The findings are strictly valid to comparisons against the baseline models and OTS does not guarantee that results would be reflected in actual systems as herein reported. The key conclusions and recommendations are:

1- R290 and R32 have wider saturation regions allowing the system to operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer.

2- Refrigerants with high temperature glide may require new heat exchanger (HX) designs, namely condensers. The original designs proved to be sufficiently effective to allow for most systems to operate with the different refrigerants, however, better designs would allow for higher system efficiency and potentially less charge. HX designs are severely constrained by allowed envelope dimensions. A complete system re-design would provide an opportunity for designing HX's with even higher efficiency.

3- The results of this analysis suggest that for an effective use of alternate low-GWP refrigerant, a proper compressor selection must be accompanied with it. Higher isentropic efficiencies are desired for higher temperatures, but most importantly, the displacement volume requirements can vary considerably from one refrigerant to another.

4- It is also imperative that having an active expansion device (preferably an EXV) to not only allow for more controlled superheat, but also to enable the unit to run with different refrigerants with very different thermophysical properties.

Table 6: Activity 2 Results.

General Information			Hardware					Ref.	Performance			
System	Rated Capacity (@35°C)	System Configuration	Compressor		Condenser		Expansion Device		Cooling Capacity (@46°C)		EER (@46°C)	
-	BTU/hr	-	Effective Disp. Vol. (cm³)*	Efficiency (-)	Type	Effectiveness (-)	Type	-	BTU/hr	%	BTU/hr. W	%
Unit 1	18000	**Baseline**	**19.8**	**0.66**	**Tube-Fin (5mm Tube)**	**0.20**	**Passive**	**R444B**	**17403**	**0.00%**	**7.4**	**0.00%**
		Alternate 1	25.9	0.70	Same as Baseline	0.35	Active (EXV)	R290	17639	1.40%	8.01	8.20%
		Alternate 2	24.8	0.69		0.26		R454C	18104	4.00%	7.31	-1.30%
		Alternate 3	19.6	0.70		0.23		R444B	18140	4.20%	8.14	9.90%
		Alternate 4	25.3	0.68	MCHX	0.24		R457A	17749	2.00%	7.63	3.10%
Unit 4	24000	**Baseline**	**26.4**	**0.61**	**Tube-Fin (9.5mm Tube)**	**0.24**	**Passive**	**R290**	**17940**	**0.00%**	**7.52**	**0.00%**
		Alternate 1	26.3	0.70	Tube-Fin (5mm Tube)	0.26	Active (EXV)	R290	18147	1.20%	9.12	21.40%
		Alternate 2	37.9	0.70		0.20		R290	24120	34.40%	6.72	-10.60%
Unit 6	24000	**Baseline**	**16.0**	**0.60**	**Tube-Fin (7mm Tube)**	**0.12**	**Passive**	**R32**	**23115**	**0.00%**	**8.46**	**0.00%**
		Alternate 1	16.9	0.65	Tube-Fin (5mm Tube)	0.15	Active (EXV)	R32	23798	3.00%	9.41	11.20%
		Alternate 2	18.4	0.67		0.19		R454B	22894	-1.00%	9.71	14.80%
		Alternate 3	19.0	0.70		0.17		R452B	23702	2.50%	9.6	13.50%
Unit 10	36000	**Baseline**	**19.6**	**0.44**	**Tube-Fin (9.5mm Tube)**	**0.13**	**Passive**	**R32**	**29005**	**0.00%**	**6.39**	**0.00%**
		Alternate 1	22.3	0.65	Tube-Fin (5mm Tube)	0.25	Active (EXV)	R447B	30478	5.10%	9.43	47.50%
		Alternate 2	23.0	0.67		0.25		R452B	30796	6.20%	10.27	60.70%
		Alternate 3	23.3	0.67		0.25		R454B	30809	6.20%	10	56.50%

** Product of displacement volume and volumetric efficiency*

5. Activities 3, 4 & 5 - Prototype Units Fabrication, Evaluation of the Optimized Prototypes and Analyzing Leaks of Alternatives

Activities 3-5 officially began in April 2019 when the first round of tests on modified Unit 6 were carried out. Initial tests resulting in unsuccessful outcomes leading OTS to change the system modifications and the scope. Additional information found in APPENDIX B . The detailed test data and charge optimization for Units 6 and 10 are presented in APPENDIX C through APPENDIX E . Comparisons between Activity 2 model validations and experimental data are presented in APPENDIX F .

5.1. Unit 6

Some modifications were made to Unit 6 to improve its efficiency. The baseline compressor was replaced with alternate models to account for the change in refrigerant and to improve efficiency. The compressor used with R454B had a higher displacement volume than the one used with R32. Furthermore, the capillary tubes were replaced with a manual TXV that was installed directly at the evaporator inlet to increase the cooling capacity of the evaporator. A summary of the design modifications evaluated for Unit 6 is listed in Table 7, while Table 8 and Table 9 show the performance of Unit 6 for baseline and modifications at 35°C and 46°C ambient, respectively. The baseline system performed similar, within 2%, to reported performance in PRAHA I. There is a discrepancy in the measurements from condenser outlet to expansion inlet in the baseline case, since the capillary tube (removed in the modified systems) was located in the outdoor unit The expansion causes the refrigerant to flash in the liquid line thus compromising the readings at the expansion device. For calculation purposes, the condenser outlet enthalpy was used instead of the expansion inlet.

Table 7: Unit 6 Modifications for Testing.

System	Unit 6		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R32	R454B
Compressor	GMCC KSG226N1UMT	Copeland ZP20K5E	Copeland ZP21K5E
Expansion Device	Capillary Tube (Outdoor unit)	Manual Valve (Indoor Unit)[2]	Manual Valve (Indoor Unit)[2]

Cooling capacity for the modified unit with either refrigerant was consistently lower by 6-12% than the baseline. The modified R32 system reportedly showed lower mass flow rate than expected, likely the main cause for the lower-than-expected thermal performance. The R454B system resulted in a poorer performance but was less sensitive to ambient temperature than its R32 counterpart - i.e. cooling capacity was near the same at both 35°C and 46°C, while for R32 there was a ~2,000BTU/hr reduction with the temperature increase. It is also possible that there is a mismatch between thermophysical property library and actual refrigerant properties for R454B which can happen with newer fluids. The libraries need periodic update as more test data become available.

[2] A manual valve was used to mimic a TXV or EXV recommended as component modification in these systems configurations.

Table 8: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	-	**R32**	**R32**	**R454B**	-	-
Charge	lb	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	25192	23585	21966	-6.4%	-12.8%
Energy Balance	%	-2.28%	-4.66%	-3.06%	-	-
Compressor Power	kW	2.11	1.79	1.77	-15.1%	-16.2%
Fan Power	kW	0.32	0.33	0.33	2.2%	4.2%
Total Power	kW	2.43	2.12	2.10	-12.8%	-13.5%
EER	BTU/hr.W	10.37	11.12	10.44	7.2%	0.68%

Table 9: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 46°C.

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	-	**R32**	**R32**	**R454B**	-	-
Charge	lb	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	23390	21450	21821	-8.3%	-6.7%
Energy Balance	%	-1.78%	-4.42%	-7.61%	-	-
Compressor Power	kW	2.71	2.32	2.25	-14.2%	-16.6%
Fan Power	kW	0.40	0.42	0.42	5.3%	5.3%
Total Power	kW	3.10	2.74	2.67	-11.7%	-13.8%
EER	BTU/hr.W	7.55	7.84	8.17	3.8%	8.2%

5.2. Unit 10

Applying what was learned in the initial modifications to Unit 6, modifications to Unit 10 were limited to include the compressor and expansion device only. Unlike Unit 6, however, the re-test of the baseline system was not successful; refer to APPENDIX D for additional information. However since Unit 6 baseline re-test showed good reproducibility from original data, it is assumed that the Unit 10 original baseline is appropriate for comparison against the modified system configurations. A summary of the design modifications evaluated for Unit 10 is listed in Table 10. The detailed test data is presented in APPENDIX E .

At 35°C the modified units exhibited almost 20% less cooling capacity with 10% less power consumption, resulting in up to 11% less EER (Table 11). These results were not unexpected since the modified units were re-designed using the 46°C temperature, when the baseline system's performance showed a great degradation of performance. At 46°C condition, the tests exhibited 2-5% greater cooling capacity with up to 12% less power consumption compared to the baseline, which was equivalent to 13-17% greater system performance.

In Activity 2 the compressor power consumptions were underestimated, as well as the total fan power consumption, leaving the impression the overall performance improvement would considerably be greater than the observed. The cooling capacity, on the other hand, was predicted with less than 2% deviation from test data, validating at least the models created.

Table 10: Unit 10 Modifications for Testing.

System	Unit 10		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R447B	R452B
Compressor	Copeland ZP42K6E	Copeland ZP34K5E	Copeland ZP31K5E
Expansion Device	Orifice	Manual Valve	Manual Valve

Table 11: Unit 10 - Performance Test Summary for R32 Baseline @ 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**447B**	**452B**	**-**	**-**
Charge	lb	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	35543	32195	28128	-9.42%	-20.86%
Energy Balance	%	---	7.52%	-3.29%	-	-
Compressor Power	kW	-	2.67	2.4	-	-
Fan Power	kW	-	0.95	0.98	-	-
Total Power	kW	3.761	3.62	3.38	-3.75%	-10.13%
EER	BTU/hr.W	9.451	8.894	8.322	-5.89%	-11.94%

Table 12: Unit 10 - Performance Test Summary for R32 Baseline @ 46°C.

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**447B**	**452B**	**-**	**-**
Charge	lb	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	29633	31073	30292	4.86%	2.22%
Energy Balance	%	---	4.21%	1.21%	-	-
Compressor Power	kW	---	3.18	2.93	-	-
Fan Power	kW	---	0.95	0.97	-	-
Total Power	kW	4.466	4.13	3.9	-7.52%	-12.67%
EER	BTU/hr.W	6.64	7.52	7.76	13.33%	16.95%

5.3. Leak Tests

In the interest of time the leak tests were conducted only on Unit 10 for R447B. The choice of refrigerant was based on temperature glide, where R447B exhibits the highest glide amongst the refrigerants evaluated between Unit 6 and Unit 10 (refer to Figure 4). The leak tests were conducted to closely represent field operation. The procedure applied includes the following steps:

1- Run unit until steady-state is achieved (repeat 46°C performance test), monitoring capacity and subcooling
2- Gradually remove refrigerant from vapor line until capacity is reduced to approximately 50%, if possible
3- Store and weigh removed refrigerant
4- Re-charge with new refrigerant until same subcooling is achieved
5- Compare cooling capacities; if more than 5% deviation is observed, repeat steps 1-4, however in step 2, reduce capacity to 25% only
6- Repeat steps 1-5 for the liquid line

The comparison herein presented refers to a leakage of approximately 30% of charge, while reducing capacity by approximately 50% based on airside only. The leak tests showed less than 2% deviation in cooling capacity after re-charge from both vapor and liquid lines (Table 13). Since the capacity deviation was less than 5%, no further testing for 25% capacity reduction was conducted. The results suggest little impact due to fractionation.

Table 13: Unit 10 – R447B Leak Test Summary Results.

System			*Liquid Line Leak*		*Vapor Line Leak*	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	**-**	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Charge	lb	6.625	4.27	6.625	4.23	6.77
Cooling Capacity	BTU/hr	31073	14216	30865	15171	30587
Energy Balance	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Compressor Power	kW	3.18	2.93	3.18	2.94	-[3]
Fan Power	kW	0.95	0.98	0.98	0.98	0.98
Total Power	kW	4.13	3.90	4.16	3.92	-[3]
EER	BTU/hr.W	7.52	3.64	7.42	3.87	-[3]

5.4. Conclusions and Recommendations

This section presented the performance tests conducted on units 6 and 10. The key conclusions and recommendations are:

1- Unit 6 re-tested baseline exhibited similar performance to that found in PRAHA I testing. It should be stressed that the baseline unit by design had its capillary tube located in the outdoor unit. This would cause liquid refrigerant leaving the outdoor unit to flash. The refrigerant enthalpy at the condenser outlet state was used to calculate the refrigerant-side capacity assuming an isentalphic expansion without heat loss in connecting pipe. This is different from the modified systems of which the capillary tube was removed, and a manual expansion valve was placed at the inlet of the indoor unit. For modified systems, the enthalpy at the expansion valve inlet was used to calculate the refrigerant-side capacity.

2- Unit 10 exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This also indicates the importance of proper compressor selection.

3- The higher-than-expected power consumption in the Unit 10 baseline tests is also evidenced by the fact that even with zeotropic mixtures (R447B and R452B), Unit 10 had higher cooling capacity and efficiency than the baseline for the 46°C test condition, as projected in activity 2.

4- Because of the differences in saturation curves from the Activity 2 analysis, R32 tends to result in systems with higher efficiency and less charge when no modifications to the hardware are made. The results showed however, that making appropriate component selection, such as compressors with larger displacement volumes and higher mass flow rates for the zeotropic mixtures, cooling capacities and overall performance were of the same order of magnitude.

5- Refrigerant fractionation as evidenced by the leak tests, does not appear to be a great concern since less than 2% deviation in cooling capacity was observed after the system's re-charge.

6- The Unit 6 modified systems had lower performance than expected from the Activity 2 models. The R32 system configuration exhibited more than 10% less flow rate than anticipated due to performance

[3] Compressor power consumption was not properly recorded for this test; the error was identified after the fact and the team was unable to retrieve that information. While that compromises the assessment of the overall system performance, the deviations are expected to be marginal. The leak test on liquid line suggest minimal impact on power consumption after re-charge, while cooling capacity was reportedly fully recovered after recharge on both leak tests.

maps overprediction, which corresponded to 10% lower capacity. The R454B configuration exhibited a deviation of 5% between model and test due also in part to a 3% flow rate over prediction in the model. Unit 10, on the other hand, exhibited an excellent agreement to the models with less than 2% deviation in cooling capacity.

7- The model's validation adds confidence in the numerical simulation findings and recommendations provided in activity 2.

6. Conclusions

This report presents a comprehensive set of activities with the objectives of advancing the PRAHA program. The original scope and schedule were modified during the project as new findings and challenges surfaced. The tests that were carried out for PRAHA-I, while sufficient for the purpose of measuring capacity and energy efficiency for the purposes of PRAHA-I, did not have enough essential data to enable a complete cycle evaluation for optimization purposes. This is primarily due to using standard test rig on systems with critical hardware configuration differences. The analyses presented in Activity 2 (design assessment through modeling) provided good insights on adequate component design and/or selection for proper system functioning, when using novel refrigerants.

The final recommendations for future development are listed as follows:

1- Establish a baseline system by conducting comprehensive testing including measurements and metrics not typically performed in energy certification tests. Furthermore, testing systems with different configurations require custom test rigs as such to adequately measure working fluid's states to avoid mischaracterization of the operating conditions and performance. Such approach is considerably more labor-intensive which should be factored in the scope in future developments.

2- Using alternate low-GWP refrigerants is viable and can be competitive to presently used refrigerants but doing so requires proper component design and selection; compressor and expansion device particularly. Drop-in replacement without hardware change is never recommended as evidenced by the change requirements in Activity 2 and performance tests in the subsequent activities.

3- It is recommended to always perform numerical simulations, and to conduct at least some level of "soft" optimization analyses that will provide information for an educated system re-design / retrofit at much lower costs than gradual trial-and-error changes.

4- Always test the modified systems with the same instrumentation as the baseline, however mindful of the modifications as such to properly place sensors to obtain adequate readings as suggested in item 1 above.

Nomenclature

COP	Coefficient of Performance	-
D_o	Tube Outer Diameter	mm
f	Frequency	Hz
FPI	Fins per Inch	1/in
h	Enthalpy	kJ/kg
h_t	Tube Height	mm
HX	Heat Exchanger	-
$\dot{m}$	Mass Flow Rate	kg/s
MCHX	Microchannel Heat Exchanger	-
P	Pressure	kPa
P_l	Tube Longitudinal Pitch	mm
P_t	Tube Transverse Pitch	mm
s	Entropy	kJ/kg.K
T	Temperature	°C
TFHX	Tube-Fin Heat Exchanger	-
UA	Thermal Conductance	kW/K
V	Volume	m^3
w_t	Tube Width	mm
η_{vol}	Volumetric Efficiency	-
ρ	Density	kg/m^3

APPENDIX A - Activity 2 Design Framework Results

Table 14: Unit 1 – Theoretical Cycle Re-Design Summary.

System		*Baseline*	*Alternative 1*	*Alternative 2*	*Alternative 3*	*Alternative 4*
Case	-	Simulation	Target			
Refrigerant	**-**	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Condenser	-	BTFD5	-	-	-	-
Compressor	-	SL260DG-C8EU	-	-	-	-
Cooling Capacity	BTU/hr	17403	17477	17477	17477	17477
Compressor Power	kW	1.92	1.49	1.49	1.33	1.43
Fan Power	kW	0.43	0.43	0.43	0.43	0.43
Total Power	kW	2.35	1.92	1.93	1.76	1.86
COP	-	2.17	2.66	2.66	2.91	2.75
COP Gain	-	1.00	1.23	1.23	1.34	1.27

Table 15: Unit 1 – HX Analysis Summary

Condenser		**R444B**		**R290**		**R454C**		**R457A**	
Inputs		***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56
Refrigerant Pressure	kPa	2875.0	2875.0	2170.7	2170.7	2436.4	2436.4	2183.9	2183.9
Saturation Temperature at Inlet	°C	61	61	61	61	61	61	61	61
Refrigerant Temperature	°C	110.00	110.00	110.00	110.00	110.00	110.00	110.00	110.00
Mass Flow Rate	kg/s	0.03	0.03	0.02	0.02	0.03	0.03	0.03	0.03
Outputs									
Heat Load	W	7512.9	7441.2	8232.4	8016.6	6168.0	6040.0	6592.0	6429.0
Air Dry-Bulb Temperature	°C	58.6	58.2	59.7	59.6	56.3	56.3	57.0	56.9
Refrigerant Temperature	°C	46.7	48.1	50.3	53.8	47.2	49.5	48.0	51.1
LMTD	°C	12	15	19	23	14	18	16	21
UA	W/K	635.57	482.84	439.36	350.35	451.67	327.93	424.35	313.48
NTU	-	1.04	0.79	0.72	0.57	0.74	0.53	0.69	0.51
Effectiveness	-	0.1915	0.1896	0.2098	0.2043	0.1572	0.1539	0.1680	0.1638
Refrigerant Pressure Drop	kPa	78.2	1.4	85.0	1.7	79.3	1.4	87.2	1.7
Airside DP	Pa	75.1	75.5	75.1	75.5	75.1	75.5	75.1	75.5
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	130.0	148.3	130.0	148.3	130.0	148.3	130.0	148.3
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	3341.0	1721.0	4113.0	2033.0	3040.0	1382.0	3423.0	1601.0
Subcooling	°C	13.20	13.14	8.96	7.35	6.77	5.93	5.34	4.05
Charge	kg	0.3822	0.1143	0.1079	0.0352	0.3097	0.094	0.2522	0.0764

Table 16: Unit 1 – Compressor Performance Summary.

Compressor		Baseline				
Refrigerant	-	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Isentropic efficiency	-	0.66	0.70	0.69	0.70	0.68
Power	kW	1.9175	1.7682	2.0449	1.7966	1.8932
Pressure Lift	kPa	2284.8	1556.0	2087.7	1902.2	1904.9
Effective Displacement Volume	cm^3	19.80	25.87	24.80	19.64	25.35
Rotation Speed	RPM	3600	3600	3600	3600	3600

Table 17: Unit 1 – Expected Modified System Performances.

System		*Baseline*				
Case	-	Simulation	Expected			
Refrigerant	**-**	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Condenser	-	BTFD5	BTFD5	BTFD5	BTFD5	NMCD2
Compressor	-	SL260DG-C8EU	-	-	-	-
Cooling Capacity	BTU/hr	17403	17639	18104	18140	17749
Compressor Power	kW	1.92	1.77	2.04	1.80	1.89
Fan Power	kW	0.43	0.43	0.43	0.43	0.43
Total Power	kW	2.35	2.20	2.48	2.23	2.33
COP	-	2.17	2.35	2.14	2.38	2.24
COP Gain	-	1.00	1.08	0.99	1.10	1.03

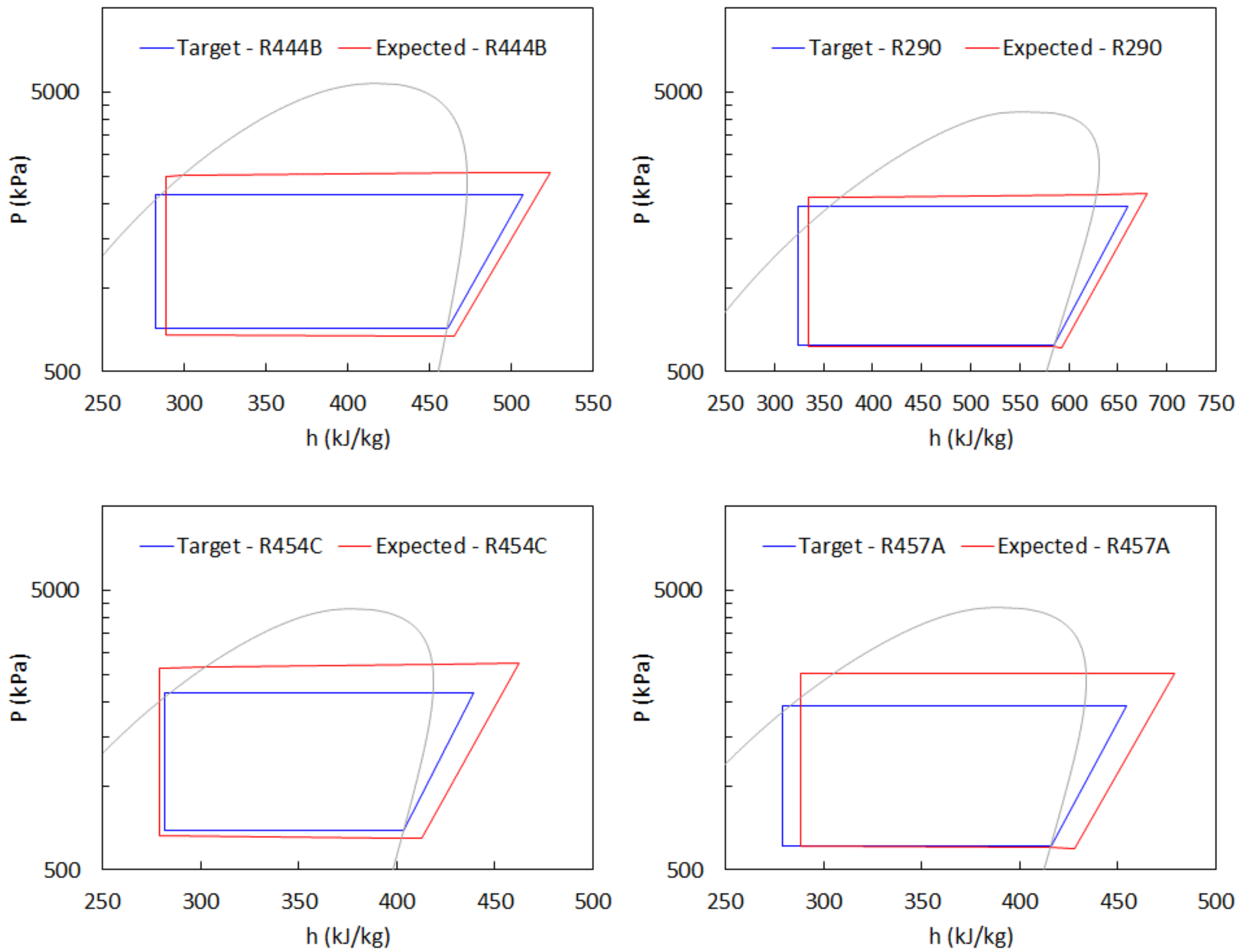

Figure 10. Unit 1 – Modified Systems P-h Diagrams.

Table 18: Unit 4 – Theoretical Cycle Re-Design Summary.

System		*Baseline*	*Alternative 1*	*Alternative 2*
			Target	
Refrigerant	-	**R290**	**R290**	**R290**
Condenser	-	BTFD9	-	-
Compressor	-	PSH356DG-C8DU4	-	-
Cooling Capacity	BTU/hr	17940	17940	23920
Compressor Power	kW	2.11	1.40	3.23
Fan Power	kW	0.28	0.28	0.28
Total Power	kW	2.39	1.68	3.51
COP	-	2.20	3.14	2.00
COP Gain	-	1.00	1.42	0.91

Table 19: Unit 4 – HX Analysis Summary.

Condenser		**R290 - 18kBTU**		**R290 - 24kBTU**	
Inputs		***BTFD9***	***NTFD5***	***BTFD9***	***NTFD5***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	0.81	0.76	0.81	0.76
Refrigerant Pressure	kPa	2875	2875	2875	2875
Saturation Temperature at Inlet	°C	75.5	75.5	75.5	75.5

Condenser		R290 - 18kBTU		R290 - 24kBTU	
Inputs		***BTFD9***	***NTFD5***	***BTFD9***	***NTFD5***
Refrigerant Temperature	°C	110	110	110	110
Mass Flow Rate	kg/s	0.02	0.02	0.03	0.03
Outputs					
Heat Load	W	8139	8148	12080	12190
Air Dry-Bulb Temperature	°C	55.0	56.1	59.5	61.2
Refrigerant Temperature	°C	46.2	46.0	47.7	46.4
LMTD	°C	9.6	7.4	14.3	10.0
UA	W/K	848	1097	846	1216
NTU	-	0.97	1.34	0.97	1.48
Effectiveness	-	0.15	0.16	0.22	0.23
Refrigerant Pressure Drop	kPa	4.2	13.4	11.0	35.2
Airside DP	Pa	16.0	15.9	16.0	15.9
Air Heat Transfer Coefficient (Average)	W/m².K	82.9	100.7	82.9	100.7
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	1535.2	1493.7	2382.4	2505.6
Subcooling	°C	29.2	29.2	27.6	28.4
Charge in Tubes	kg	0.90	0.46	0.76	0.39

Table 20: Unit 4 – Compressor Performance Summary.

Compressor		**Baseline**	**18kBTU/Hr**		**24kBTU/Hr**	
Refrigerant	**-**	**R290**	**R290**	**R290**	**R290**	**R290**
Isentropic efficiency	-	0.61	0.70	0.70	0.70	0.70
Power	kW	2.1067	1.7364	1.7093	3.3152	3.31
Pressure Lift	kPa	1457.6	1556.3	1513.7	2947.1	2937.4
Effective Displacement Volume	cm³	26.394	26.309	26.309	37.866	37.866
Rotation Speed	RPM	3600	3600	3600	3600	3600

Table 21: Unit 4 – Expected Modified System Performances.

System		***Baseline***	***Alternative 1***		***Alternative 2***	
			Expected			
Refrigerant	**-**	**R290**	**R290**	**R290**	**R290**	**R290**
Condenser	-	BTFD9	BTFD9	NTFD5	BTFD9	NTFD5
Compressor	-	PSH356DG-C8DU4	-	-	-	-
Cooling Capacity	BTU/hr	17940	17991	18147	24045	24120
Compressor Power	kW	2.11	1.74	1.71	3.32	3.31
Fan Power	kW	0.28	0.28	0.28	0.28	0.28
Total Power	kW	2.39	2.02	1.99	3.60	3.59
COP	-	2.20	2.61	2.67	1.96	1.97
COP Gain	-	1.00	1.19	1.21	0.89	0.89

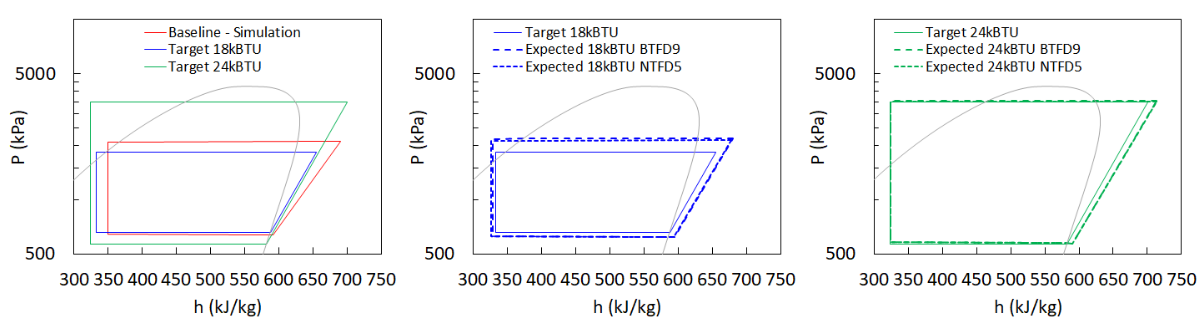

Figure 11. Unit 4 – Modified Systems P-h Diagrams.

Table 22: Unit 6 – Theoretical Cycle Re-Design Summary.

System			*Alternate 1*	*Alternate 2*	*Alternate 3*
		Simulation	**Target**		
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R452B**
Condenser	-	BTFD9	-	-	-
Compressor	-	GMCC KSG226N1UMT	ZP20K5E	ZP21K5E	-
Cooling Capacity	BTU/hr	23115	23114	23114	23115
Compressor Power	kW	2.73	2.37	2.29	2.04
Fan Power	kW	8.46	9.75	10.10	11.31
Total Power	kW	2.73	2.37	2.29	2.04
COP	-	2.48	2.86	2.96	3.32
COP Gain	-	1.00	1.15	1.19	1.34

Table 23: Unit 6 – HX Analysis for R32

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3562	3562	3562	3562
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	9159	9416	9332	9113
Air Dry-Bulb Temperature	°C	53.63	55.35	54.27	55.24
Refrigerant Temperature	°C	49.78	46.15	47.40	50.47
LMTD	°C	19.94	9.46	15.13	20.57
UA	W/K	459.40	995.12	616.75	443.09
NTU	-	0.39	0.97	0.52	0.43
Refrigerant Pressure Drop	kPa	100.98	26.10	3.06	4.70
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	5543.00	2624.00	2353.00	2978.00
Subcooling	°C	4.48	9.04	8.10	5.07
Charge	kg	0.39	0.71	0.17	0.11

Table 24: Unit 6 – HX Analysis for R452B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3247	3247	3247	3247
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7876	7964	7936	7866
Air Dry-Bulb Temperature	°C	52.52	53.94	53.06	53.99
Refrigerant Temperature	°C	47.41	46.05	46.53	47.61
LMTD	°C	15.49	8.09	12.37	15.72
UA	W/K	508.37	984.95	641.46	500.33
NTU	-	0.43	0.96	0.55	0.49
Refrigerant Pressure Drop	kPa	71.90	21.03	2.60	3.70
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	4252.00	2077.00	2103.00	2112.00
Subcooling	°C	6.14	8.20	7.99	6.89
Charge	kg	0.55	0.90	0.21	0.15

Table 25: Unit 6 – HX Analysis for R447B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3025	3025	3025	3025
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7607	8241	8157	7914
Air Dry-Bulb Temperature	°C	52.41	54.19	53.25	54.04
Refrigerant Temperature	°C	50.00	46.24	47.63	51.40
LMTD	°C	20.58	10.45	15.92	22.14
UA	W/K	369.65	788.34	512.32	357.47
NTU	-	0.31	0.77	0.44	0.35
Refrigerant Pressure Drop	kPa	185.90	27.30	3.18	4.90
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	5396.00	2439.00	2397.00	3281.00
Subcooling	°C	0.00	6.05	5.17	1.22
Charge	kg	0.33	0.70	0.16	0.11

Table 26: Unit 6 – HX Analysis for R454B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3204	3204	3204	3204
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7993	8094	8060	7976
Air Dry-Bulb Temperature	°C	52.61	54.06	53.16	54.10
Refrigerant Temperature	°C	47.59	46.06	46.61	47.91
LMTD	°C	15.95	8.28	12.72	16.40
UA	W/K	501.09	977.17	633.67	486.37
NTU	-	0.43	0.96	0.54	0.48
Refrigerant Pressure Drop	kPa	74.70	22.02	2.70	4.10
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	4445.93	2140.00	2008.00	2201.00
Subcooling	°C	5.75	8.03	7.75	6.43
Charge	kg	0.51	0.87	0.20	0.14

Table 27: Unit 6 – Compressor Performance Summary.

		Baseline	***Alternate 1***	***Alternate 2***	***Alternate 3***
Refrigerant		***R32***	***R32***	***R454B***	***R452B***
Isentropic Efficiency	-	0.60	0.64	0.66	0.70
Volumetric Efficiency	-	-	0.87	0.90	-
Displacement Volume	cm³	-	19.34	20.31	-
Frequency	Hz	60	60	60	60
Effective Displacement	cm³	16.0	16.8	18.3	19.0
Compressor Power	kW	2.4	2.3	2.3	2.1

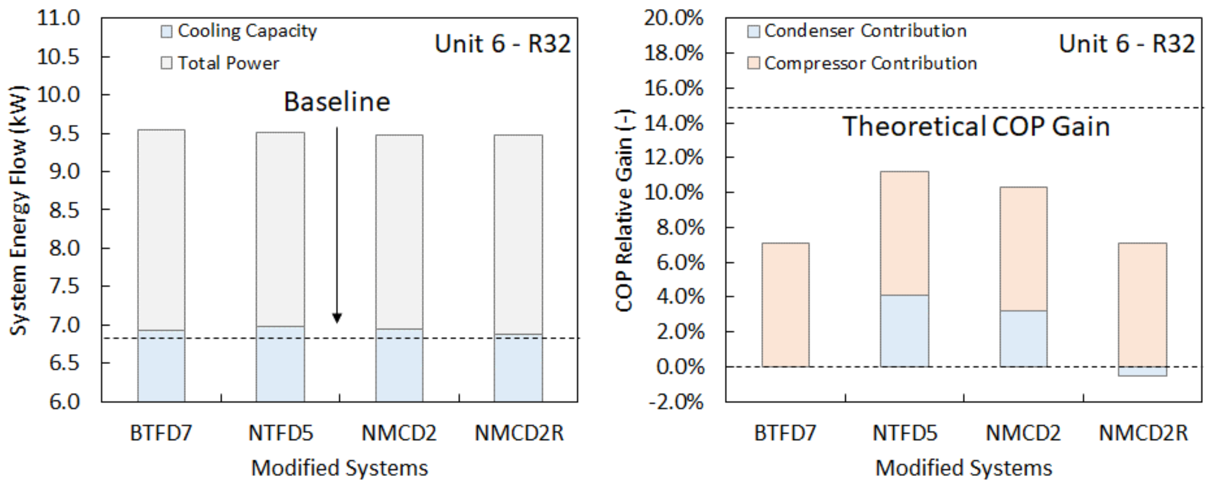

Figure 12. Unit 6 – System Level Analysis: Performance Results for R32.

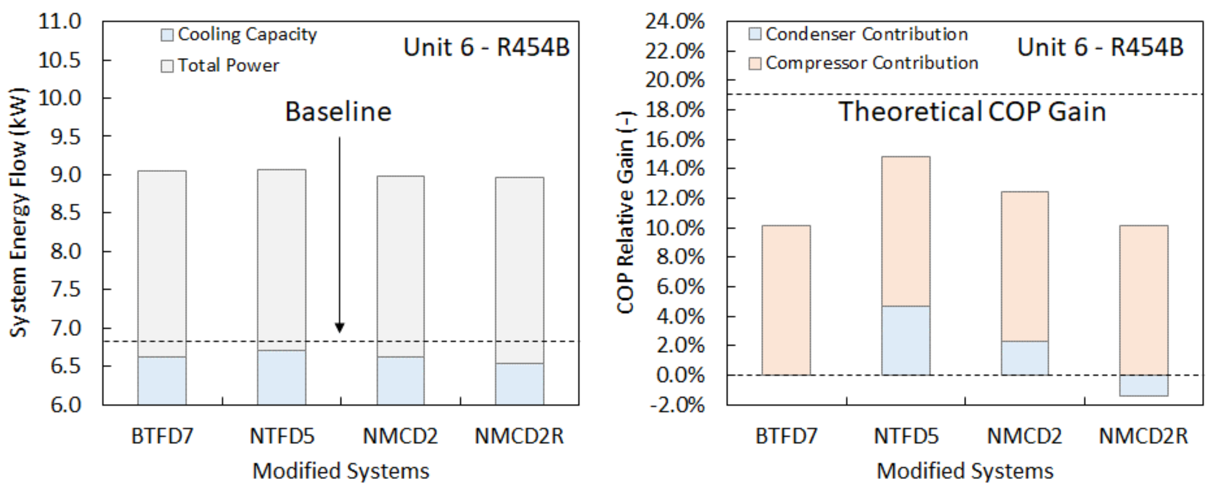

Figure 13. Unit 6 – System Level Analysis: Performance Results for R454B.

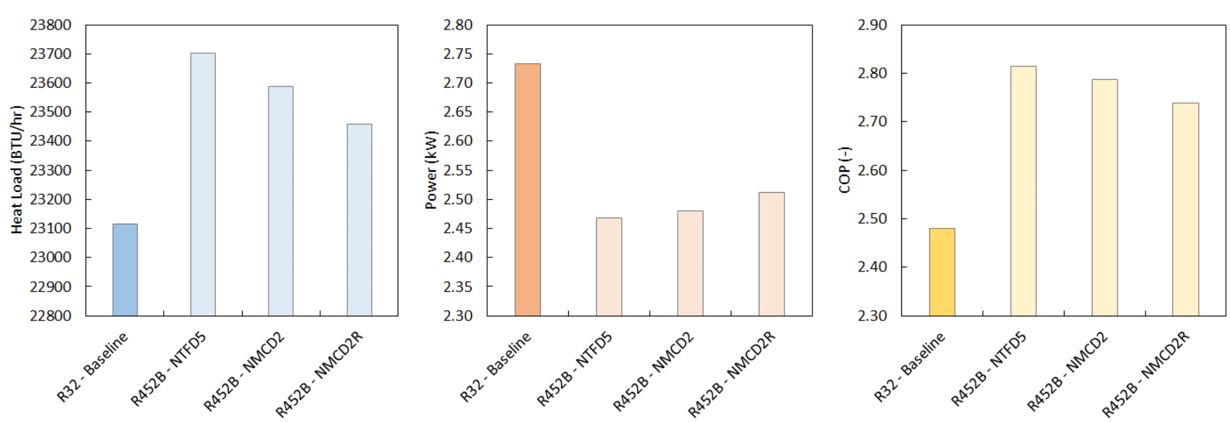

Figure 14. Unit 6 - Comparative System Performance Summary for R452B.

Table 28: Unit 10 – Theoretical Cycle Re-Design Summary.

System		*Baseline*	*Alternate 1*	*Alternate 2*	*Alternate 3*
		Simulation		**Target**	
Refrigerant	**-**	**R32**	**R452B**	**R447B**	**R454B**
Condenser	-	BTFD9	-	-	-
Compressor	-	ZP42K5E	ZP31K5E	ZP34K5E	ZP31K5E
Cooling Capacity	BTU/hr	29005	34311	31611	34608
Compressor Power	kW	3.84	2.81	2.31	2.65
Fan Power	kW	0.70	0.70	0.70	0.70
Total Power	kW	4.54	3.51	3.01	3.35
COP	-	1.87	2.87	3.08	3.03
COP Gain	-	1.00	1.53	1.64	1.62

Table 29: Unit 10 – HX Analysis for R32

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3562	3562	3562	3562
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	10693	11074	11435	10669
Air Dry-Bulb Temperature	°C	54.1	57.0	54.9	55.8
Refrigerant Temperature	°C	55.2	52.9	49.3	55.4
LMTD	°C	22.8	19.8	15.9	22.5
UA	W/K	468	560	717	475
NTU	-	0.35	0.55	0.54	0.42
Refrigerant Pressure Drop	kPa	26.7	67.1	6.8	10.1
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	3823	4239	3050	3991
Subcooling	°C	0.00	1.75	6.17	0.00
Charge	kg	0.61	0.43	0.17	0.11

Table 30: Unit 10 – HX Analysis for R452B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3247	3247	3247	3247
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9549	9812	9751	9500
Air Dry-Bulb Temperature	°C	53.2	55.8	53.6	54.8
Refrigerant Temperature	°C	49.5	46.4	47.1	50.1
LMTD	°C	16.7	9.2	12.2	17.1
UA	W/K	573	1067	802	557
NTU	-	0.43	1.04	0.60	0.49
Refrigerant Pressure Drop	kPa	17.2	47.1	5.6	8.2
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	2974	3038	2537	2812
Subcooling	°C	4.82	7.51	7.34	4.38
Charge	kg	0.83	0.79	0.23	0.15

Table 31: Unit 10 – HX Analysis for R447B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m^3/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3025	3025	3025	3025
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9016	9632	9923	9085
Air Dry-Bulb Temperature	°C	52.9	55.6	53.8	54.4
Refrigerant Temperature	°C	52.4	51.7	49.9	52.7
LMTD	°C	20.4	18.9	17.1	20.3
UA	W/K	441	510	579	448
NTU	-	0.33	0.50	0.43	0.40
Refrigerant Pressure Drop	kPa	29.2	67.3	7.2	10.8
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	3528	3833	2999	3458
Subcooling	°C	0.00	0.00	2.67	0.00
Charge	kg	0.56	0.45	0.17	0.10

Table 32: Unit 10 – HX Analysis for R454B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m^3/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3204	3204	3204	3204
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9634	9953	9901	9597
Air Dry-Bulb Temperature	°C	53.3	55.9	53.8	54.9
Refrigerant Temperature	°C	50.4	46.7	47.3	50.8
LMTD	°C	17.9	10.5	12.7	18.0
UA	W/K	537	952	782	532
NTU	-	0.40	0.93	0.59	0.47
Refrigerant Pressure Drop	kPa	18.8	51.1	5.9	8.7
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	3095	3211	2633	2942
Subcooling	°C	3.71	6.98	6.98	3.40
Charge	kg	0.78	0.71	0.22	0.14

Table 33. Unit 10 - Compressor Performance Summary.

Compressor			Copeland ZP31K5E-PFV	Copeland ZP34K5E-PFV	Copeland ZP31K5E-PFV
Refrigerant		***R32***	***R452B***	***R447B***	***R454B***
Isentropic Efficiency	-	0.439	0.638	0.662	0.662
Volumetric Efficiency	-		0.760	0.803	0.790
Displacement Volume	cm^3		29.350	29.350	29.350
Frequency	Hz	50	50	50	50
Effective Displacement Volume	cm^3	19.646	22.301	23.581	23.183

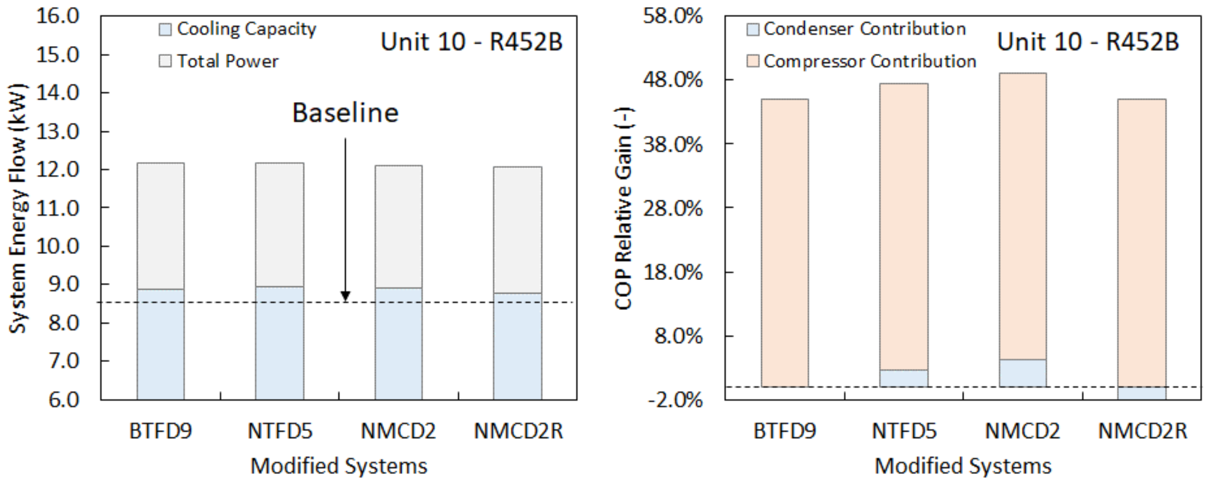

Figure 15. Unit 10 – System Level Analysis: Performance Results for R452B.

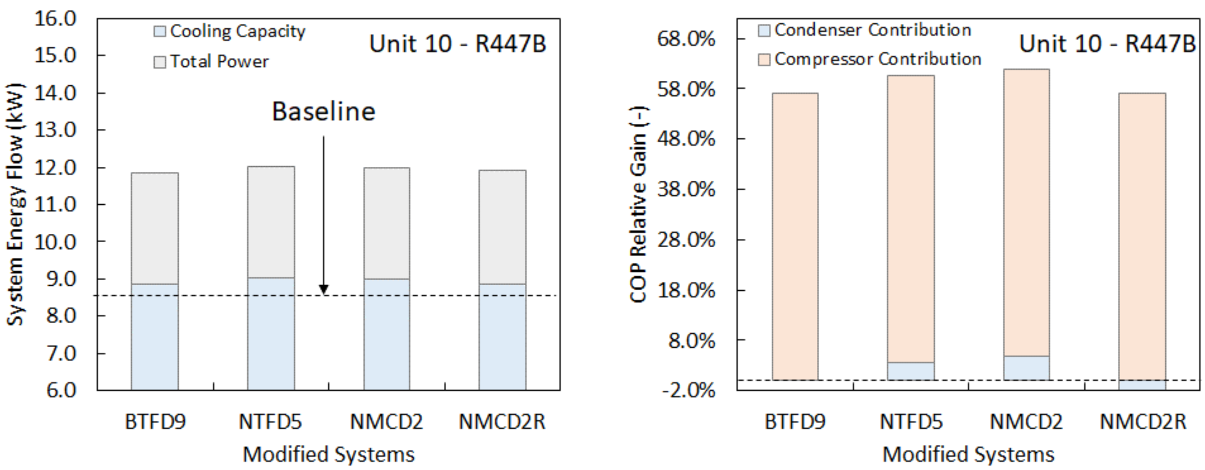

Figure 16. Unit 10 – System Level Analysis: Performance Results for R447B.

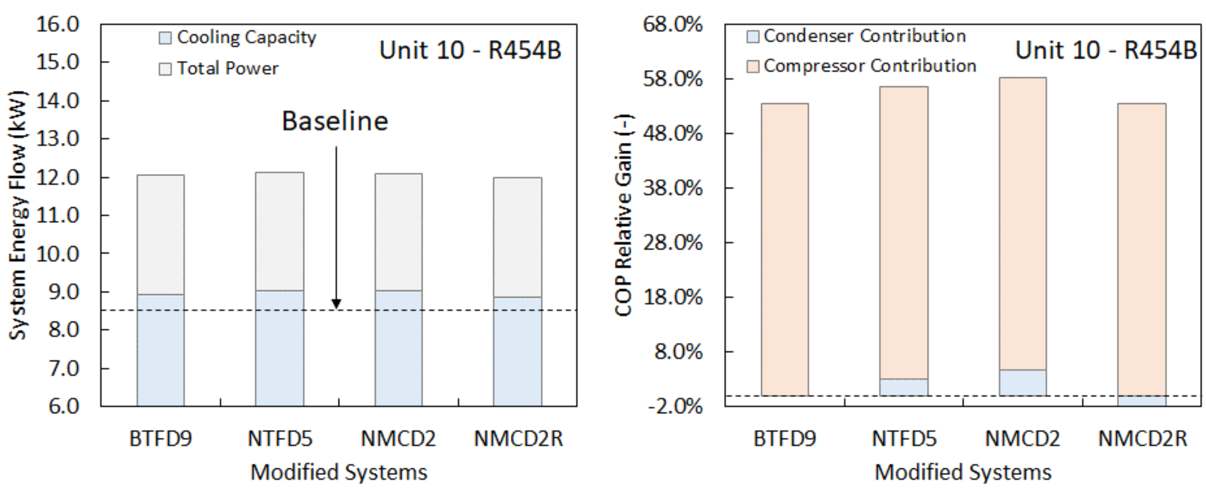

Figure 17. Unit 10 – System Level Analysis: Performance Results for R454B.

APPENDIX B – Unit 6 Initial Tests, Scope Change and Test Setup

Unit 6 was initially modified and tested at a separate facility and the test results exhibited a considerably lower cooling capacity than expected (~20%). Power consumption was also greater than designed. The condensing pressures were 20-30% above expectations, and the refrigerant pressure drop across the condenser was at least twice as high as expected. The outlet conditions of the condenser for R32 were possibly in two-phase. The condenser airflow rate was 10%-15% lower than expected. Superheat hardly met the setpoint values.

OTS formulated a hypothesis that the degraded performance was due to the condenser not being fully active; i.e. some regions were not transferring heat. One way for this to happen is by having severe maldistribution thus impeding heat transfer, increasing pressure drop – thus the condensing pressure – and possibly reducing the flow rate as well; all of which were observed in the test data. OTS tested the hypothesis by running hot water through the HX and observing with a thermal camera (Figure 18), which revealed the "dead zones". Upon inspection by the manufacturer, it was confirmed there were blockages in some of the tubes. A new HX was built, but the same pattern was observed, forcing OTS to remove the condenser replacement from the scope given the project schedule.

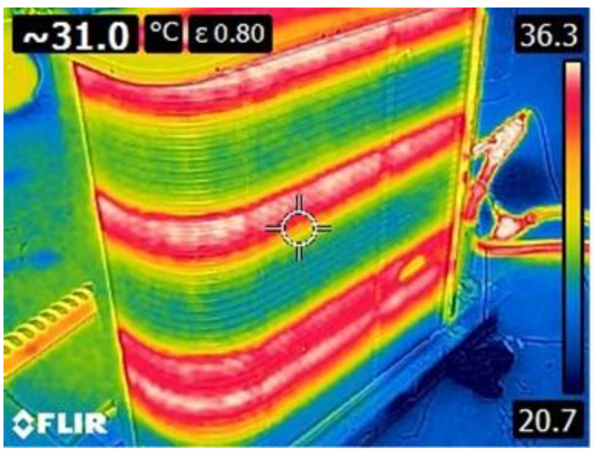

Figure 18. Hot Water Thermal Imaging.

Given the challenges with the initial tests and unit modification, the scope was re-defined. The original test plan was changed to accommodate time and resources as appropriate. Table 34 outlines the major changes to the scope. The tests were conducted at the OTS laboratory (Figure 19 to Figure 22). A summary of the key differences between the test setups (original and at OTS) is presented in Table 35.

Table 34: Test Scope Change.

Unit	Refrigerant	Test	Original Scope		New Scope	
			Planned	Actual	Planned	Actual
Unit 1	R290	Charge Optimization	Yes	No	No	No
		Performance Tests	Yes	No	No	No
Unit 6	R32 (Baseline)	Charge Optimization	No	No	Yes	Yes
		Performance Tests	No	No	Yes	Yes
	R32 (Modified)	Charge Optimization	Yes	Yes	Yes	Yes
		Performance Tests	Yes	Yes	Yes	Yes
	R454B	Charge Optimization	Yes	Yes	Yes	Yes
		Performance Tests	Yes	Yes	Yes	Yes
		Leak Tests	Yes	Yes	No	No
Unit 10	R32 (Baseline)	Charge Optimization	No	No	Yes	Yes*
		Performance Tests	No	No	Yes	Yes*
	R447B	Charge Optimization	Yes	No	Yes	Yes
		Performance Tests	Yes	No	Yes	Yes
		Leak Tests	Yes	No	Yes	Yes
	R452B	Charge Optimization	Yes	No	Yes	Yes
		Performance Tests	Yes	No	Yes	Yes
		Leak Tests	Yes	No	No	No

** Tests were conducted; however, no useful data was obtained (see section 5.2)*

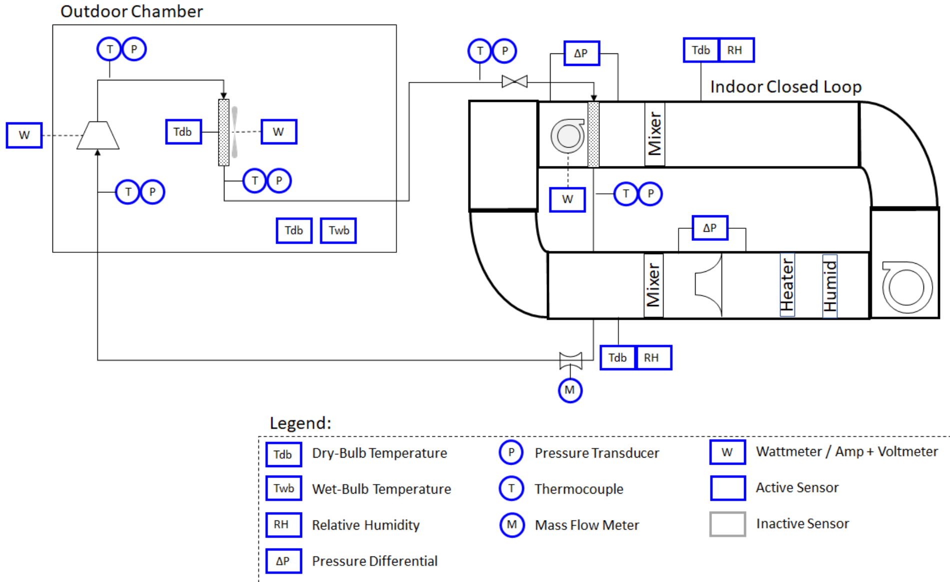

Figure 19. Test Diagram.

Figure 20. OTS Setup: outdoor chamber (left), Unit 10 and frequency converter inside chamber (right).

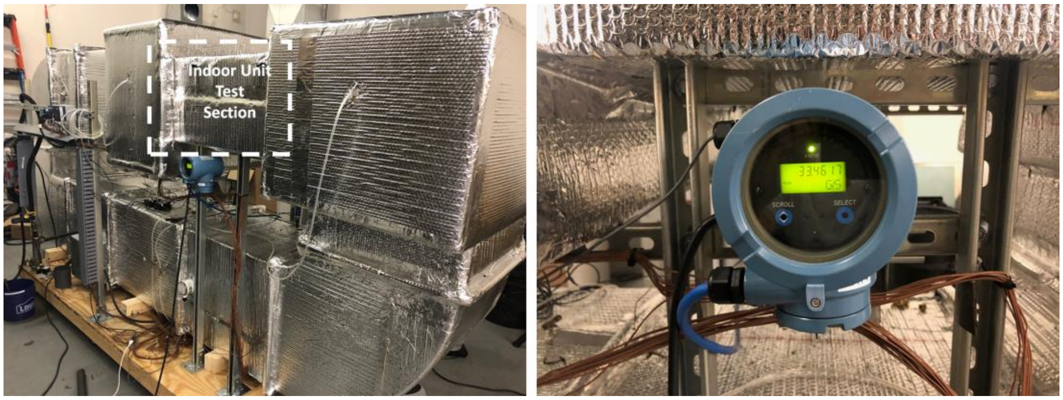

Figure 21. OTS Setup: indoor closed loop left side view (left), refrigerant mass flow meter (right).

Figure 22. OTS Setup: indoor closed loop right side view (left), vapor / liquid lines, sight glasses and TXV (right).

Table 35: List of Measurements.

Component	Refrigerant Side			Air Side		
	Measurement	**Original Scope**	**New Scope**	**Measurement**	**Original Scope**	**New Scope**
Condenser	Inlet Temperature	Yes	Yes	Air Flow Rate	Yes	**No**
	Inlet Pressure	Yes	Yes	Air Pressure Drop	No	No
	Outlet Temperature	Yes	Yes	Fan Power	No	**Yes**
	Outlet Pressure	Yes	Yes	Inlet Dry-bulb	Yes	Yes
	Subcooling	Yes*	**Yes**	Inlet Wet-Bulb / RH	Yes	Yes
				Outlet Dry-bulb	Yes	Yes
				Outlet Wet-Bulb / RH	Yes	Yes
Evaporator	Inlet Temperature	No	No	Air Flow Rate	Yes	Yes
	Inlet Pressure	No	No	Air Pressure Drop	No	**Yes****
	Outlet Temperature	Yes	Yes	Blower Power	No	**Yes**
	Outlet Pressure	Yes	Yes	Inlet Dry-bulb	Yes	Yes
	Superheat	Yes*	**Yes**	Inlet Wet-Bulb / RH	Yes	Yes
	Refrigerant Mass Flow Rate	No	**Yes**	Outlet Dry-bulb	Yes	Yes
				Outlet Wet-Bulb / RH	Yes	Yes
Compressor	Suction Temperature	Yes	Yes			
	Suction Pressure	Yes	Yes			
	Discharge Temperature	Yes	Yes			
	Discharge Pressure	Yes	Yes			
	Compressor Power	No	**Yes**			
Expansion Device	Suction Temperature	Yes	Yes			
	Suction Pressure	Yes	Yes			
	Discharge Temperature	No	No			
	Discharge Pressure	No	No			

Charge Optimization

The charge optimization procedure as originally scoped was not implemented due to the following:

a. The systems responded less sensitively to charge on subcooling and superheat, which were difficult to control with charging alone. A manual valve was added (Unit 10 exhibited little expansion) such that superheat could be better controlled. The valve also allowed for better control over the pressure levels compared to charge levels alone.

b. For the modified systems, the charge was gradually increased, departing from the original charge from PRAHA I tests, until it was observed that the superheat and subcooling better matched design conditions for validation purposes.

c. For the refrigerant blends, removing charge could result in fractionation (evaluated as a separate task), so it was decided to only incrementally increase charge, without removing it. For this procedure, a small gradual increment is necessary to avoid overcharging.

APPENDIX C - Unit 6 Raw and Processed Tested Data

Table 36: Unit 6 – Performance Tests

		Baseline (35°C)	**Alternate 1 (35°C)**	**Alternate 2 (35°C)**	**Baseline (46°C)**	**Alternate 1 (46°C)**	**Alternate 2 (46°C)**
Refrigerant	-	**R32**	**R32**	**R454B**	**R32**	**R32**	**R454B**
Charge	lb	3.83	4.27	5.02	3.83	4.27	5.02
Cooling Capacity	BTU/hr	25193	23585	21966	23390	21450	21821
Energy Balance	%	-2.28%	-4.66%	-3.06%	-1.78%	-4.42%	-7.61%
Compressor Power	kW	2.11	1.79	1.77	2.71	2.32	2.25
Fan Power	kW	0.32	0.33	0.33	0.40	0.42	0.42
Total Power	kW	2.43	2.12	2.10	3.10	2.74	2.67
EER	BTU/hr.W	10.36	11.12	10.44	7.54	7.84	8.17
Evaporator							
Airside							
Inlet							
Air Flow Rate	m^3/s	0.31	0.31	0.31	0.31	0.31	0.30

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R32**	**R32**	**R454B**
Temperature	°C	27.0	27.0	27.0	29.0	29.0	29.0
Wet Bulb	°C	19.68	19.68	19.68	21.33	21.33	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.011	0.011	0.011	0.013	0.013	0.013
Density	kg/m³	1.15	1.15	1.15	1.14	1.14	1.14
Enthalpy	kJ/kg	56.3	56.2	56.2	61.9	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0	1.0
Outlet							
Air Flow Rate	m³/s	0.29	0.29	0.29	0.29	0.29	0.29
Temperature	°C	14.3	15.1	15.8	16.9	17.7	18.1
Wet Bulb	°C	14.35	14.35	14.35	14.35	14.35	14.35
Relative Humidity	%	83.6	82.4	80.0	84.5	83.3	81.3
Humidity Ratio	kg/kg	0.008	0.009	0.009	0.010	0.011	0.011
Density	kg/m³	1.21	1.20	1.20	1.19	1.19	1.19
Enthalpy	kJ/kg	35.8	37.5	38.5	42.7	44.7	45.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0	1.0
Refrigerant Side							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	4.58	6.19	4.76	7.49	8.33	8.47
Pressure	kPa	939.13	986.90	876.76	1026.70	1053.10	979.34
Quality	-	0.16	0.19	0.20	0.20	0.25	0.27
Enthalpy	kJ/kg	273.64	269.78	268.60	301.30	291.37	289.89
Entropy	kJ/kg.K	1.20	1.25	1.30	1.27	1.32	1.37
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	8.08	9.26	9.46	9.08	13.54	11.80
Pressure	kPa	939	987	877	1027	1053	979
Superheat	K	3.50	3.07	4.89	1.59	5.20	3.58
Enthalpy	kJ/kg	520.49	520.22	473.43	518.52	523.27	472.93
Entropy	kJ/kg.K	2.15	2.15	2.03	2.13	2.15	2.02
HX Level							
Average Cooling Capacity	kW	7.384	6.912	6.438	6.855	6.287	6.395
Energy Balance (Qair - Qref)/Qref	%	-2.28%	-4.66%	-3.06%	-1.78%	-4.42%	-7.61%
Sensible Heat Ratio	-	0.64	0.66	0.65	0.64	0.67	0.66
Superheat	K	3.500	3.066	4.885	1.593	5.205	3.582
LMTD	K	13.783	12.822	14.015	13.985	12.184	13.041
UA	kW/K	0.573	0.539	0.459	0.550	0.516	0.490
Air Pressure Drop	Pa	N/A	N/A	N/A	N/A	N/A	N/A
Refrigerant Pressure Drop	kPa	N/A	N/A	N/A	N/A	N/A	N/A
Fan Power	kW	0.120	0.127	0.134	0.196	0.217	0.217
Condenser							
Airside							
Inlet							
Air Flow Rate	m³/s	0.9516	0.9838	1.0091	0.9580	0.9735	1.0613
Temperature	°C	35.01	34.76	35.12	46.06	45.93	46.05
Wet Bulb	°C	20.0	19.8	20.0	27.4	27.3	27.4
Humidity Ratio	kg/kg	0.008	0.008	0.009	0.015	0.015	0.015
Density	kg/m³	1.13	1.13	1.13	1.08	1.08	1.08
Enthalpy	kJ/kg	57.0	56.4	57.2	86.2	85.8	86.2
Specific Heat	kJ/kg.K	1.01	1.01	1.01	1.02	1.02	1.02
Outlet							
Air Flow Rate	m³/s	0.98	1.01	1.03	0.98	1.00	1.09
Temperature	°C	43.40	42.29	42.08	54.74	53.60	53.19
Wet Bulb	°C	22.4	22.0	22.1	29.3	29.0	29.0
Humidity Ratio	kg/kg	0.008	0.008	0.009	0.015	0.015	0.015
Density	kg/m³	1.10	1.10	1.10	1.05	1.05	1.05
Enthalpy	kJ/kg	65.6	64.1	64.3	95.2	93.7	93.6
Specific Heat	kJ/kg.K	1.01	1.01	1.01	1.02	1.02	1.02

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	-	R32	R32	R454B	R32	R32	R454B
Refrigerant Side							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	89.78	82.73	78.33	109.00	107.24	90.75
Pressure	kPa	2724.15	2643.18	2360.90	3464.77	3365.88	3010.13
Superheat	K	45.9	40.1	35.9	54.7	54.2	38.0
Enthalpy	kJ/kg	580.73	573.07	523.39	594.42	593.52	528.90
Entropy	kJ/kg.K	2.20	2.18	2.08	2.21	2.21	2.07
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	39.17	34.52	34.68	51.79	45.63	45.79
Pressure	kPa	2675.81	2598.75	2310.89	3416.39	3324.50	2958.91
Subcooling	K	4.00	7.44	5.59	1.89	6.84	5.07
Enthalpy	kJ/kg	273.6	264.0	266.4	301.3	287.0	287.8
Entropy	kJ/kg.K	1.24	1.21	1.28	1.33	1.28	1.34
HX Level							
Heat Rejection	kW	9.19	8.53	8.08	9.25	8.31	8.42
Subcooling	K	4.00	7.44	5.59	1.89	6.84	5.07
Refrigerant Pressure Drop	kPa	48.34	44.43	50.01	48.38	41.38	51.22
Fan Power	kW	0.20	0.20	0.20	0.20	0.20	0.20
TXV							
Refrigerant							
Inlet		[4]			[4]		
Temperature	°C	30.64	37.31	35.83	39.70	47.55	46.78
Pressure	kPa	1991.01	2587.20	2301.38	2528.52	3317.42	2945.62
Subcooling	°C	*(Two-Phase)	4.47	4.27	*(Two-Phase)	4.83	3.88
Enthalpy	kJ/kg	*(Two-Phase)	269.8	268.6	*(Two-Phase)	291.4	289.9
Entropy	kJ/kg.K	*(Two-Phase)	1.233	1.284	*(Two-Phase)	1.299	1.349
Compressor							
Refrigerant							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	11.57	12.55	12.76	13.81	17.63	13.07
Pressure	kPa	936.06	984.95	874.98	1024.91	1052.17	969.56
Superheat	K	7.09	6.43	8.26	6.38	9.32	5.18
Enthalpy	kJ/kg	524.9	524.4	477.3	524.6	528.3	474.8
Entropy	kJ/kg.K	2.170	2.161	2.048	2.156	2.166	2.028
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	89.8	82.7	78.3	109.0	107.2	90.8
Pressure	kPa	2724.2	2643.2	2360.9	3464.8	3365.9	3010.1
Superheat	K	45.9	40.1	35.9	54.7	54.2	38.0
Enthalpy	kJ/kg	580.7	573.1	523.4	594.4	593.5	528.9
Entropy	kJ/kg.K	2.200	2.183	2.084	2.205	2.207	2.074
Compressor Level							
Power Consumption	kW	2.11	1.79	1.77	2.71	2.32	2.25
Isentropic Efficiency	-	0.80	0.84	0.73	0.74	0.76	0.69
Frequency	Hz	60	60	60	60	60	60

[4] The baseline configuration does not have an expansion valve, the state point herein presented refers to measurement readings at indoor unit inlet.

APPENDIX D - Unit 10 Baseline Re-Test

Prior to modifying Unit 10, it was tested in its received, baseline condition with the components used to test during PRAHA I. Given the results of the data review in Activity 1, and the challenges experienced in the initial testing of Unit 6, the project team agreed that testing the units in their baseline configuration would be important for more accurate comparison.

The electrical components for Unit 10 have phase mismatch, i.e. the fan and blower are three-phase while the compressor is single-phase, but all operate in 50Hz. OTS does not have a Variable Frequency Drive (VFD) for single-phase motors, requiring the use of a frequency converter to reduce the compressor speed. According to the baseline data from PRAHA 1, the total power consumption of Unit 10 varied between 3.5-4.5kW; OTS has a 5.0kW converter, which should be sufficiently large to meet testing needs.

Initial tests suggested that the compressor peak start current exceeds the converter threshold, causing the latter to trip and shut off. Although the blower and the fan run normally with the converter, the compressor alone does not. The compressor motor was tested at 60Hz direct from the grid and it works, thus confirming that the issue is indeed the peak current. A soft starter was acquired with the objective to mitigate the issue. The soft starter capacitors weren't fast enough to smooth the peak current, however, thus requiring manual charging, which eventually lead to component failure.

The last tentative to run the baseline was connecting the compressor to 60Hz and the fans to 50Hz. The refrigerant mass flow rate was too high impeding full condensation and full evaporation. A manual TXV was added along with two sight glasses in the liquid and vapor lines and reasonable data was obtained for the 35°C ambient temperature condition. While attempting to test the system under the 46°C ambient temperature, the compressor overheats and shuts down. Heavier gauge wire, new contactors and switch bypass were unsuccessfully employed. In the interest of time, the baseline re-tests were discontinued. The analysis will be carried out using the original baseline performance for comparison purposes.

APPENDIX E - Unit 10 Raw and Processed Tested Data

Table 37: Unit 10 – Performance Tests.

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	-	**R447B**	**R452B**	**R447B**	**R452B**
Charge	lb	6.625	6.625	6.625	6.625
Cooling Capacity	BTU/hr	32195	28128	31073	30292
Energy Balance	%	7.52%	-3.29%	4.21%	1.21%
Compressor Power	kW	2.67	2.40	3.16	2.93
Fan Power	kW	0.95	0.98	0.95	0.97
Total Power	kW	3.62	3.38	4.11	3.90
EER	BTU/hr.W	8.88	8.33	7.55	7.76
Evaporator					
Airside					
Inlet					
Air Flow Rate	m^3/s	0.74	0.73	0.74	0.73
Temperature	°C	27.0	27.0	29.0	29.0
Wet Bulb	°C	19.68	19.69	21.33	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.011	0.011	0.013	0.013
Density	kg/m^3	1.15	1.15	1.14	1.14
Enthalpy	kJ/kg	56.2	56.3	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R447B**	**R452B**	**R447B**	**R452B**
Outlet					
Air Flow Rate	m³/s	0.72	0.71	0.71	0.70
Temperature	°C	17.4	19.1	19.7	19.8
Wet Bulb	°C	15.80	16.64	17.91	18.06
Relative Humidity	%	85.1	78.5	84.7	84.5
Humidity Ratio	kg/kg	0.011	0.011	0.012	0.012
Density	kg/m³	1.19	1.18	1.18	1.18
Enthalpy	kJ/kg	44.3	46.8	50.7	51.1
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0
Refrigerant Side					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	9.81	5.53	12.90	13.09
Pressure	kPa	996.41	907.20	1085.49	1133.86
Quality	-	0.19	0.19	0.27	0.25
Enthalpy	kJ/kg	272.43	264.74	296.09	288.71
Entropy	kJ/kg.K	1.32	1.30	1.40	1.38
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	15.22	25.20	16.76	23.36
Pressure	kPa	996	907	1085	1134
Superheat	K	5.79	19.82	4.42	10.47
Enthalpy	kJ/kg	477.29	485.20	476.43	477.36
Entropy	kJ/kg.K	2.04	2.09	2.03	2.03
HX Level					
Average Cooling Capacity	kW	9.436	8.244	9.107	8.878
Energy Balance (Qair - Qref)/Qref	%	7.52%	-3.29%	4.21%	1.21%
Sensible Heat Ratio	-	0.81	0.85	0.83	0.87
Superheat	K	5.794	19.818	4.422	10.474
LMTD	K	9.534	5.829	9.222	6.171
UA	kW/K	0.990	1.414	0.988	1.439
Air Pressure Drop	Pa	N/A	N/A	N/A	N/A
Refrigerant Pressure Drop	kPa	N/A	N/A	N/A	N/A
Fan Power	kW	0.502	0.523	0.501	0.519
Condenser					
Airside					
Inlet					
Air Flow Rate	m³/s	1.44	1.50	1.44	1.42
Temperature	°C	35.03	35.08	46.14	46.22
Wet Bulb	°C	20.0	20.0	27.4	27.5
Humidity Ratio	kg/kg	0.008	0.009	0.016	0.016
Density	kg/m³	1.13	1.13	1.08	1.07
Enthalpy	kJ/kg	57.0	57.2	86.5	86.7
Specific Heat	kJ/kg.K	1.01	1.01	1.02	1.02
Outlet					
Air Flow Rate	m³/s	1.47	1.53	1.48	1.45
Temperature	°C	41.90	40.83	53.36	53.26
Wet Bulb	°C	22.0	21.7	29.0	29.1
Humidity Ratio	kg/kg	0.008	0.009	0.016	0.016
Density	kg/m³	1.10	1.11	1.05	1.05
Enthalpy	kJ/kg	64.0	63.0	94.0	94.0
Specific Heat	kJ/kg.K	1.01	1.01	1.02	1.02
		0.00010	0.00038	0.00011	-0.00001
Refrigerant Side					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	-	**R447B**	**R452B**	**R447B**	**R452B**
Temperature	°C	78.84	92.46	93.29	97.45
Pressure	kPa	2493.84	2600.61	3199.13	3357.43
Superheat	K	31.5	46.5	35.3	40.4
Enthalpy	kJ/kg	522.20	532.28	529.64	527.68
Entropy	kJ/kg.K	2.09	2.11	2.08	2.07
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	40.68	35.54	53.44	48.65
Pressure	kPa	2481.63	2599.27	3187.26	3351.92
Subcooling	K	3.37	9.26	1.62	7.33
Enthalpy	kJ/kg	274.8	266.6	300.2	291.9
Entropy	kJ/kg.K	1.32	1.29	1.39	1.37
HX Level					
Heat Rejection	kW	11.39	9.94	11.59	11.10
Energy Balance (Qair - Qref)	kW	N/A	N/A	N/A	N/A
Subcooling	K	3.37	9.26	1.62	7.33
Air Pressure Drop	Pa	-	-	-	-
Refrigerant Pressure Drop	kPa	12.21	1.34	11.87	5.51
Fan Power	kW	0.45	0.45	0.45	0.45
TXV					
Refrigerant					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	39.42	34.55	51.55	47.11
Pressure	kPa	2462.98	2583.59	3166.49	3331.97
Subcooling	°C	4.31	9.99	3.21	8.59
Enthalpy	kJ/kg	272.4	264.7	296.1	288.7
Entropy	kJ/kg.K	1.310	1.284	1.382	1.358
Compressor					
Refrigerant					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	16.84	26.01	17.17	24.96
Pressure	kPa	993.13	902.34	1082.17	1128.72
Superheat	K	7.52	20.81	4.94	12.23
Enthalpy	kJ/kg	479.3	486.2	477.0	479.4
Entropy	kJ/kg.K	2.052	2.090	2.035	2.042
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	78.8	92.5	93.3	97.5
Pressure	kPa	2493.8	2600.6	3199.1	3357.4
Superheat	K	31.5	46.5	35.3	40.4
Enthalpy	kJ/kg	522.2	532.3	529.6	527.7
Entropy	kJ/kg.K	2.087	2.112	2.082	2.073
Compressor Level					
Power Consumption	kW	2.67	2.40	3.16	2.93
Isentropic Efficiency	-	0.72	0.83	0.68	0.77
Frequency	Hz	60	60	60	60

Table 38: Unit 10 – R447B Leak Tests

System			***Liquid Line Leak***		***Vapor Line Leak***	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	-	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Charge	lb	6.625	4.27	6.625	4.23	6.77

System			***Liquid Line Leak***		***Vapor Line Leak***	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	**-**	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Cooling Capacity	BTU/hr	31073	14216	30865	15171	30587
Energy Balance	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Compressor Power	kW	3.18	2.93	3.18	2.94	-
Fan Power	kW	0.95	0.98	0.98	0.98	0.98
Total Power	kW	4.13	3.90	4.16	3.92	-
EER	BTU/hr.W	7.52	3.64	7.42	3.87	-
Evaporator						
Airside						
Inlet						
Air Flow Rate	m³/s	0.74	0.73	0.74	0.73	0.74
Temperature	°C	29.0	29.0	29.0	29.0	29.0
Wet Bulb	°C	21.33	21.34	21.34	21.34	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.013	0.013	0.013	0.013	0.013
Density	kg/m³	1.14	1.14	1.14	1.14	1.14
Enthalpy	kJ/kg	62.0	62.0	62.0	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0
Outlet						
Air Flow Rate	m³/s	0.71	0.72	0.71	0.72	0.71
Temperature	°C	19.7	23.3	19.6	23.2	19.7
Wet Bulb	°C	17.91	19.87	18.08	19.77	18.05
Relative Humidity	%	84.7	73.1	86.3	73.6	86.0
Humidity Ratio	kg/kg	0.012	0.013	0.012	0.013	0.012
Density	kg/m³	1.18	1.16	1.18	1.16	1.18
Enthalpy	kJ/kg	50.7	57.0	51.2	56.7	51.1
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0
Refrigerant Side						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	12.90	2.61	12.94	2.81	12.75
Pressure	kPa	1085.49	794.22	1086.62	799.23	1080.50
Quality	-	0.27	0.30	0.27	0.30	0.27
Enthalpy	kJ/kg	296.09	291.52	296.48	290.79	296.24
Entropy	kJ/kg.K	1.40	1.40	1.41	1.40	1.41
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	16.76	28.23	17.07	27.95	17.01
Pressure	kPa	1085	794	1087	799	1080
Superheat	K	4.42	26.24	4.70	25.76	4.82
Enthalpy	kJ/kg	476.43	496.65	476.77	496.25	476.88
Entropy	kJ/kg.K	2.03	2.14	2.03	2.13	2.03
HX Level						
Average Cooling Capacity	kW	9.107	4.167	9.046	4.446	8.965
Energy Balance (Qair – Qref)/Qref	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Sensible Heat Ratio	-	0.83	1.18	0.90	1.12	0.89
Superheat	K	4.422	26.235	4.695	25.756	4.823
LMTD	K	9.222	6.051	9.065	6.501	9.217
UA	kW/K	0.988	0.689	0.998	0.684	0.973
Fan Power	kW	0.501	0.524	0.524	0.524	0.524
Condenser						
Airside						
Inlet						
Air Flow Rate	m³/s	1.44	1.49	1.42	1.48	1.42
Temperature	°C	46.14	46.08	46.21	45.77	46.02
Wet Bulb	°C	27.4	27.4	27.5	27.2	27.4
Humidity Ratio	kg/kg	0.016	0.015	0.016	0.015	0.015
Density	kg/m³	1.08	1.08	1.07	1.08	1.08
Enthalpy	kJ/kg	86.5	86.3	86.7	85.3	86.1
Specific Heat	kJ/kg.K	1.02	1.02	1.02	1.02	1.02

System			Liquid Line Leak		Vapor Line Leak	
		Full Charge	Low Charge	Re-Charged	Low Charge	Re-Charged
Refrigerant	-	R447B	R447B	R447B	R447B	R447B
Outlet						
Air Flow Rate	m³/s	1.48	1.52	1.46	1.50	1.46
Temperature	°C	53.36	51.27	53.52	51.05	53.28
Wet Bulb	°C	29.0	28.6	29.1	28.4	29.0
Humidity Ratio	kg/kg	0.016	0.015	0.016	0.015	0.015
Density	kg/m³	1.05	1.06	1.05	1.06	1.05
Enthalpy	kJ/kg	94.0	91.7	94.3	90.8	93.6
Specific Heat	kJ/kg.K	1.02	1.02	1.02	1.02	1.02
Refrigerant Side						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	93.29	121.77	94.07	120.31	94.34
Pressure	kPa	3199.13	2846.79	3200.02	2847.47	3175.47
Superheat	K	35.3	68.9	36.1	67.4	36.7
Enthalpy	kJ/kg	529.64	569.70	530.67	567.95	531.39
Entropy	kJ/kg.K	2.08	2.20	2.08	2.20	2.09
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	53.44	50.27	53.37	50.13	53.28
Pressure	kPa	3187.26	2843.00	3188.61	2843.11	3164.31
Subcooling	K	1.62	-0.33	1.71	-0.19	1.45
Enthalpy	kJ/kg	300.2	293.2	300.0	293.2	299.9
Entropy	kJ/kg.K	1.39	1.37	1.39	1.37	1.39
HX Level						
Heat Rejection	kW	11.59	8.60	11.57	8.69	11.49
Energy Balance (Qair – Qref)	kW	N/A	N/A	N/A	N/A	N/A
Subcooling	K	1.62	-0.33	1.71	-0.19	1.45
Refrigerant Pressure Drop	kPa	11.87	3.79	11.40	4.36	11.16
Fan Power	kW	0.45	0.45	0.45	0.45	0.45
TXV						
Refrigerant						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	51.55	49.15	51.74	48.80	51.60
Pressure	kPa	3166.49	2827.45	3168.66	2827.31	3144.31
Subcooling	°C	3.21	0.54	3.06	0.89	2.84
Enthalpy	kJ/kg	296.1	291.5	296.5	290.8	296.2
Entropy	kJ/kg.K	1.382	1.369	1.383	1.366	1.382
Compressor						
Refrigerant						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	17.17	29.26	18.00	28.98	18.47
Pressure	kPa	1082.17	793.15	1082.65	797.99	1076.58
Superheat	K	4.94	27.30	5.75	26.83	6.41
Enthalpy	kJ/kg	477.0	497.7	478.0	497.3	478.8
Entropy	kJ/kg.K	2.035	2.140	2.038	2.138	2.041
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	93.3	121.8	94.1	120.3	94.3
Pressure	kPa	3199.1	2846.8	3200.0	2847.5	3175.5
Superheat	K	35.3	68.9	36.1	67.4	36.7
Enthalpy	kJ/kg	529.6	569.7	530.7	568.0	531.4
Entropy	kJ/kg.K	2.082	2.200	2.085	2.195	2.087
Compressor Level						
Power Consumption	kW	3.18	2.93	3.18	2.94	0.00
Isentropic Efficiency	-	0.68	0.68	0.68	0.69	0.68
Frequency	Hz	60	60	60	60	60

System			Liquid Line Leak		Vapor Line Leak	
		Full Charge	Low Charge	Re-Charged	Low Charge	Re-Charged
Refrigerant	-	R447B	R447B	R447B	R447B	R447B

APPENDIX F - Model Verification and Validation

Table 39: Unit 6 – Model Verification and Validation for Alternative 1 – R32 @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	27	31	14%
Cooling Capacity	BTU/hr	21450	23653	10%
Total Power	kW	2.74	2.67	-2%
EER	BTU/hr.W	7.84	8.86	13%

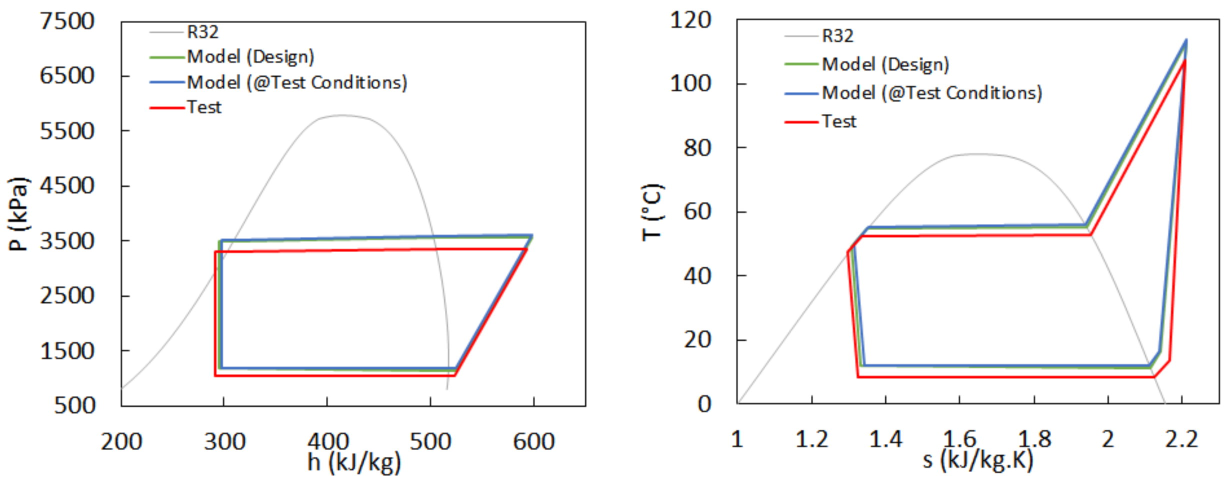

Figure 23. Unit 6 – R32 Performance Test Summary P-h and T-s Diagrams.

Table 40: Unit 6 – Model Verification and Validation for Alternative 2 – R454B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	35	36	3%
Cooling Capacity	BTU/hr	21821	22969	5%
Total Power	kW	2.67	2.49	-7%
EER	BTU/hr.W	8.17	9.24	13%

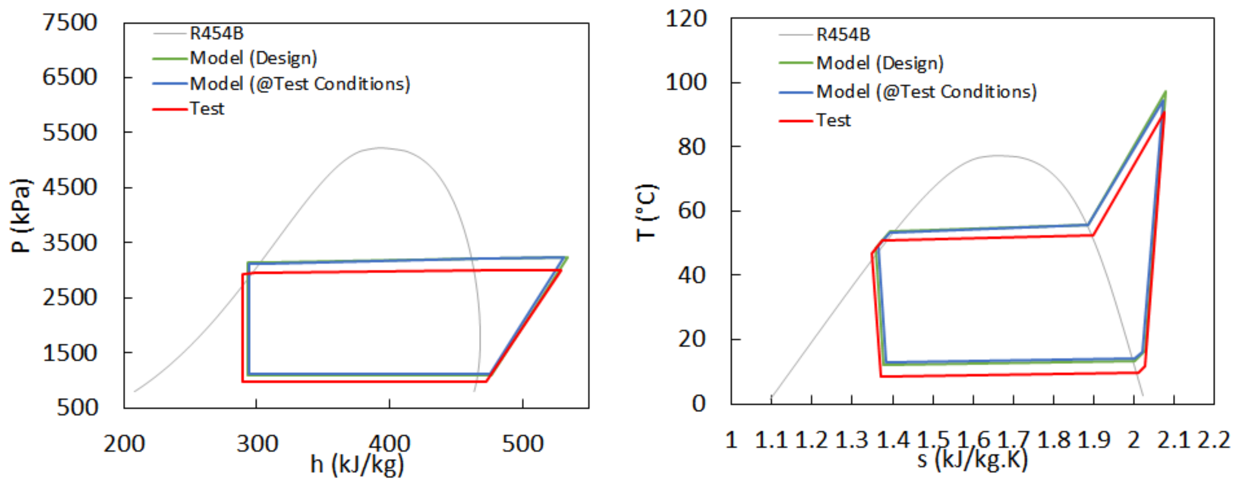

Figure 24. Unit 6 – R454B Performance Test Summary P-h and T-s Diagrams.

Table 41: Unit 10 – Model Verification and Validation for Alternative 1 – R447B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	51	49	-3%
Cooling Capacity	BTU/hr	31169	31026	-0.5%
Total Power	kW	2.70	3.00	11%
EER	BTU/hr.W	11.54	10.34	-10%

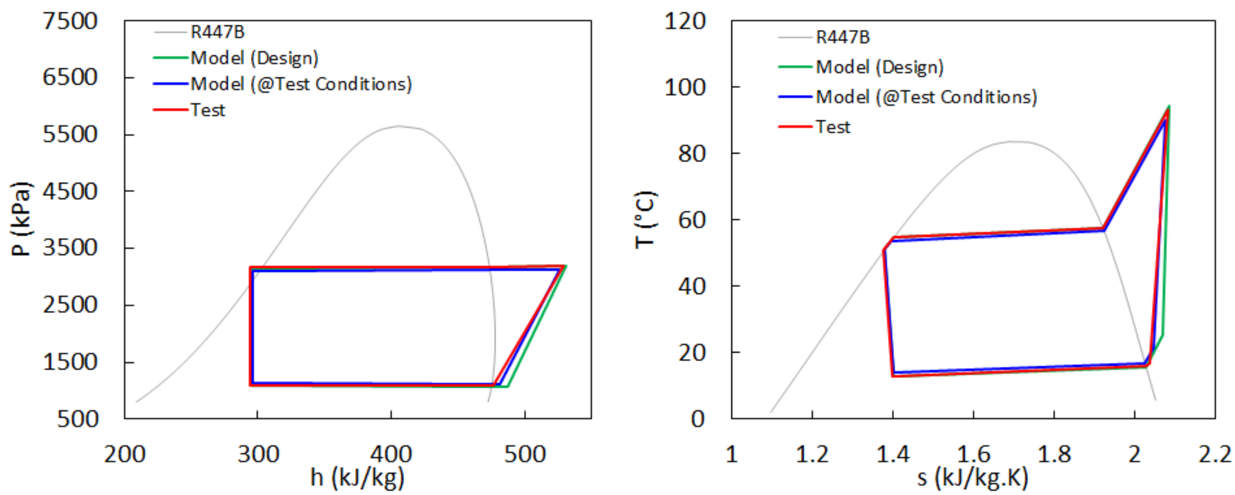

Figure 25. Unit 10 – R447B P-h and T-s Diagrams.

Table 42: Unit 10 – Model Verification and Validation for Alternative 2 – R452B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	47	48	2%
Cooling Capacity	BTU/hr	30292	30704	1.4%
Total Power	kW	3.90	3.34	-14%
EER	BTU/hr.W	7.76	9.19	18%

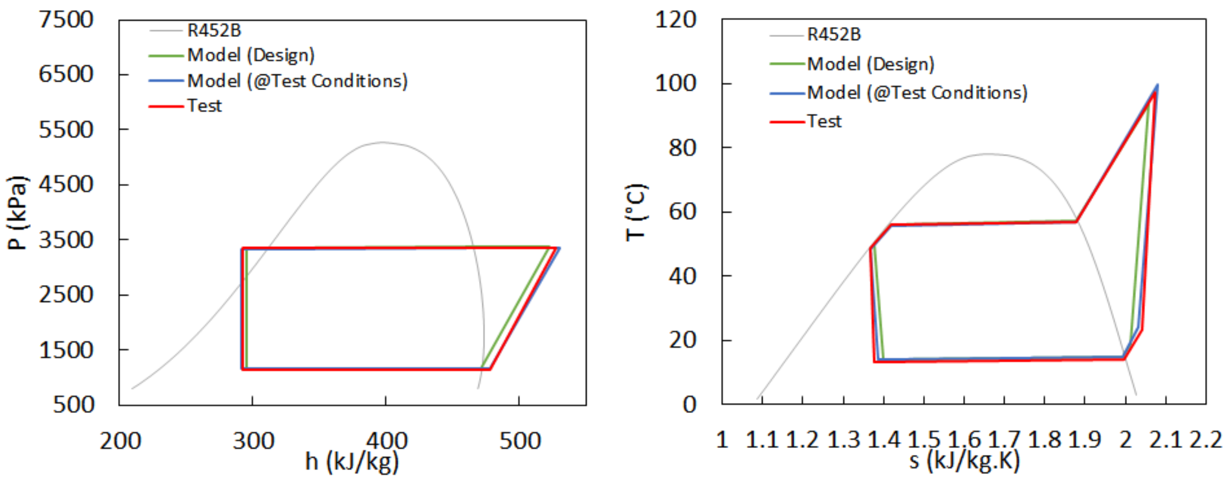

Figure 26. Unit 10 – R452B P-h and T-s Diagrams.

Air-Conditioning, Heating and Refrigeration Technology Institute

Final Report

AHRTI Report No. 9011

Promoting Alternative Refrigerants in High-Ambient Countries Phase II (PRAHA-II): Optimization Study on PRAHA I Equipment

Final Report

September 2019

By

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TABLE OF CONTENTS

1. Executive Summary 5
2. Introduction 9
3. Activity 1 - Analyzing the Design of PRAHA-I Prototypes 9
3.1. Physical Units 9
3.2. PRAHA-I Performance Reports Assessment 10
3.3. Hardware Improvement Assessment 11
3.3.1. Heat Exchanger (HX) First Order Analysis (FOA) 11
3.3.2. Compressors 12
3.3.3. Expansion Devices 12
3.3.4. Fan and Blower 12
3.3.5. Units Component Modification Potential 12
3.4. Conclusions and Recommendations 12
4. Activity 2 - Design Improvements 13
4.1. Hardware 13
4.2. Refrigerant 13
4.3. System Design Modification Framework 14
4.3.1. Compressors 16
4.3.2. HX Design and Selection 17
4.3.3. System Design 19
4.4. Modified Systems Results Summary 19
4.5. Conclusions and Recommendations 19
5. Activities 3, 4 & 5 - Prototype Units Fabrication, Evaluation of the Optimized Prototypes and Analyzing Leaks of Alternatives 21
5.1. Unit 6 21
5.2. Unit 10 22
5.3. Leak Tests 23
5.4. Conclusions and Recommendations 24
6. Conclusions 25
APPENDIX A - Activity 2 Design Framework Results 27
APPENDIX B – Unit 6 Initial Tests, Scope Change and Test Setup 36
APPENDIX C - Unit 6 Raw and Processed Tested Data 39
APPENDIX D - Unit 10 Baseline Re-Test 42
APPENDIX E - Unit 10 Raw and Processed Tested Data 42
APPENDIX F - Model Verification and Validation 47

LIST OF FIGURES

Figure 1. Heat Exchangers FOA. 11
Figure 2. Refrigerants Investigated for Units 1 and 4. 14
Figure 3. Refrigerants Investigated for Units 6 and 10. 14
Figure 4. Refrigerant Temperature Glides. 14
Figure 5. Illustrative T-s diagram for baseline and improved cycles. 15
Figure 6. P-h Diagrams Illustrating COP Improvement: a) Real Cycle; b) Ideal Cycle (Carnot). 16
Figure 7. System Re-Design Framework, ... 16
Figure 8. Condenser Forms: Unit 1 (left), Unit 10 (center), Unit 6 Cabinet (right). 18
Figure 9. HX Form Examples: L-shape (left), Flat (right). 18
Figure 10. Unit 1 – Modified Systems P-h Diagrams. 28
Figure 11. Unit 4 – Modified Systems P-h Diagrams. 29
Figure 12. Unit 6 – System Level Analysis: Performance Results for R32. 32
Figure 13. Unit 6 – System Level Analysis: Performance Results for R454B. 32
Figure 14. Unit 6 - Comparative System Performance Summary for R452B. 32
Figure 15. Unit 10 – System Level Analysis: Performance Results for R452B. 35
Figure 16. Unit 10 – System Level Analysis: Performance Results for R447B. 35
Figure 17. Unit 10 – System Level Analysis: Performance Results for R454B. 35
Figure 18. Hot Water Thermal Imaging. 36
Figure 19. Test Diagram. 37
Figure 20. OTS Setup: outdoor chamber (left), Unit 10 and frequency converter inside chamber (right). 37
Figure 21. OTS Setup: indoor closed loop left side view (left), refrigerant mass flow meter (right). 38
Figure 22. OTS Setup: indoor closed loop right side view (left), vapor / liquid lines, sight glasses and TXV (right). 38
Figure 23. Unit 6 – R32 Performance Test Summary P-h and T-s Diagrams. 47
Figure 24. Unit 6 – R454B Performance Test Summary P-h and T-s Diagrams. 48
Figure 25. Unit 10 – R447B P-h and T-s Diagrams. 48
Figure 26. Unit 10 – R452B P-h and T-s Diagrams. 49

LIST OF TABLES

Table 1: Activity 2 Summary Modeling Results. 6
Table 2: Tests Summary Results. 7
Table 3: Unit Specifications Summary. 10
Table 4: Units 1, 4, 6 and 10 Components. 12
Table 5: Compressor Models. 17
Table 6: Activity 2 Results. 20
Table 7: Unit 6 Modifications for Testing. 21
Table 8: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 35°C. 22
Table 9: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 46°C. 22
Table 10: Unit 10 Modifications for Testing. 23
Table 8: Unit 10 - Performance Test Summary for R32 Baseline @ 35°C. 23
Table 9: Unit 10 - Performance Test Summary for R32 Baseline @ 46°C. 23
Table 12: Unit 10 – R447B Leak Test Summary Results. 24
Table 13: Unit 1 – Theoretical Cycle Re-Design Summary. 27
Table 14: Unit 1 – HX Analysis Summary 27
Table 15: Unit 1 – Compressor Performance Summary. 27

Table 16: Unit 1 – Expected Modified System Performances. 27
Table 17: Unit 4 – Theoretical Cycle Re-Design Summary. 28
Table 18: Unit 4 – HX Analysis Summary. 28
Table 19: Unit 4 – Compressor Performance Summary. 29
Table 20: Unit 4 – Expected Modified System Performances. 29
Table 21: Unit 6 – Theoretical Cycle Re-Design Summary. 30
Table 22: Unit 6 – HX Analysis for R32 30
Table 23: Unit 6 – HX Analysis for R452B. 30
Table 24: Unit 6 – HX Analysis for R447B. 31
Table 25: Unit 6 – HX Analysis for R454B. 31
Table 26: Unit 6 – Compressor Performance Summary. 31
Table 27: Unit 10 – Theoretical Cycle Re-Design Summary. 33
Table 28: Unit 10 – HX Analysis for R32 33
Table 29: Unit 10 – HX Analysis for R452B. 33
Table 30: Unit 10 – HX Analysis for R447B. 34
Table 31: Unit 10 – HX Analysis for R454B. 34
Table 32. Unit 10 - Compressor Performance Summary. 34
Table 33: Test Scope Change. 37
Table 34: List of Measurements. 39
Table 35: Unit 6 – Performance Tests 39
Table 36: Unit 10 – Performance Tests. 42
Table 37: Unit 10 – R447B Leak Tests 44
Table 38: Unit 6 – Model Verification and Validation for Alternative 1 – R32 @ 46°C. 47
Table 39: Unit 6 – Model Verification and Validation for Alternative 2 – R454B @ 46°C. 47
Table 40: Unit 10 – Model Verification and Validation for Alternative 1 – R447B @ 46°C. 48
Table 41: Unit 10 – Model Verification and Validation for Alternative 2 – R452B @ 46°C. 48

1. Executive Summary

Over the past several years through the Promoting low- Global Warming Potential (GWP) Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA-I) project, 18 different prototypes have been developed and compared to respective baselines to support the assessment of alternative lower-GWP refrigerants for air-conditioning applications. Since the work originally started in 2012, researchers have identified gaps in the performance and operation of the PRAHA-1 prototypes. These gaps include the need to redesign and optimize prototype air-conditioning units, evaluate new alternative refrigerants, and improve component selection. As such, a new project, *Advancing the Designs of PRAHA-I for Meeting or Exceeding the Baseline Designs Performance*, conducted by Optimized Thermal Systems, Inc. (OTS) is herein presented.

The objectives of this project include the following:

1) Evaluate the design limitation of the PRAHA-I prototypes;
2) Optimize and physically evaluate selected prototypes with new refrigerants not evaluated during PRAHA-I; and,
3) Assess potential refrigerant fractionation impact due to leakage.

The project was organized into six activities for which a summary of the results, conclusions and recommendations are presented below:

1) Activity 1: Analyzing the Design of PRAHA-I Prototypes
    a. Certification laboratories, such as the one used for testing the units in PRAHA I, provide limited information for the purposes of product design and development. For future reference it is recommended that for research-oriented efforts such as this one, the units undergo a more rigorous testing process along with full characterization of the system and its individual components operating conditions and performance.
    b. In applications of high ambient temperatures, it is expected that performance will degrade as compared to operating under more temperate conditions and the resultant impact on performance must be considered. The key components for performance improvement identified herein were the compressor, condenser and expansion device.
        i. At higher temperatures, the saturation temperatures and refrigerant density at compressor's suction port can be very different than that from the rated conditions. Larger displacement volumes and efficiency curves optimized for higher pressure lifts might be required. Therefore, the proper selection of the compressor is paramount.
        ii. A better performance condenser will reduce the approach temperature between refrigerant and air, helping the compressor not to discharge refrigerant at very high pressure and temperatures, which degrade performance.
    c. At high ambient conditions, the system is forced to operate in higher pressure lift than at rated conditions, but still requires a certain refrigerant mass flow rate. Passive devices such as capillary tubes and orifices may not be able to provide enough expansion to allow the system to operate in higher temperature conditions. An active expansion device such as EXV's can adequately control operating conditions and maintain stable superheat.

2) Activity 2: Design Improvements (Summary results in Table 1)
    a. R290 and R32 have wider saturation regions allowing the system to operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer. Their cycles

may get closer to that of the ideal Carnot cycle compared to refrigerants with narrower saturation.

b. Refrigerants with high temperature glide may require new heat exchanger (HX) designs, namely condensers. The original designs proved to be sufficiently effective to allow for most systems to operate with the different refrigerants, however, better designs would allow for higher system efficiency and potentially less charge. HX designs are severely constrained by allowed envelope dimensions. A complete system re-design would provide an opportunity for designing HX's with even higher efficiency.
c. The results of this analysis suggest that for an effective refrigerant replacement, a proper compressor selection must be accompanied with it. Higher isentropic efficiencies are desired for higher temperatures, but most importantly, the displacement volume requirements can vary considerably from one refrigerant to another.
d. It is also imperative that having an active expansion device (preferably an Electronic Expansion valve (EXV)) to not only allow for more controlled superheat, but also to enable the unit to run with different refrigerants with very different thermophysical properties.

Table 1: Activity 2 Summary Modeling Results.

General Information			Hardware				Performance	
System	**Rated Capacity (@35°C)**	**System Configuration**	**Compressor**	**Condenser**	**Expansion Device**	**Ref.**	**Cooling Capacity (@46°C)**	**EER (@46°C)**
-	**BTU/hr**	**-**	**Efficiency (-)**	**Type**	**Type**	**-**	**BTU/hr**	**BTU/hr.W**
Unit 1	18000	**Baseline**	**0.66**	**Tube-Fin (5mm Tube)**	**Passive**	**R444B**	**17403**	**7.4**
		Alternate 1	0.7	Same as Baseline	Active (EXV)	R290	17639	8.01
		Alternate 2	0.69			R454C	18104	7.31
		Alternate 3	0.7			R444B	18140	8.14
		Alternate 4	0.68	MCHX		R457A	17749	7.63
Unit 4	24000	**Baseline**	**0.61**	**Tube-Fin (9.5mm Tube)**	**Passive**	**R290**	**17940**	**7.52**
		Alternate 1	0.7	Tube-Fin (5mm Tube)	Active (EXV)	R290	18147	9.12
		Alternate 2	0.7			R290	24120	6.72
Unit 6	24000	**Baseline**	**0.6**	**Tube-Fin (7mm Tube)**	**Passive**	**R32**	**23115**	**8.46**
		Alternate 1	0.65	Tube-Fin (5mm Tube)	Active (EXV)	R32	23798	9.41
		Alternate 2	0.67			R454B	22894	9.71
		Alternate 3	0.7			R452B	23702	9.6
Unit 10	36000	**Baseline**	**0.44**	**Tube-Fin (9.5mm Tube)**	**Passive**	**R32**	**29005**	**6.39**
		Alternate 1	0.65	Tube-Fin (5mm Tube)	Active (EXV)	R447B	30478	9.43
		Alternate 2	0.67			R452B	30796	10.27
		Alternate 3	0.67			R454B	30809	10

3) Activities 3-5: Prototype Modification and Testing (Summary results in Table 2)

a. Unit 6 re-tested baseline exhibited similar performance to that found in PRAHA I testing. It should be stressed that the baseline unit by design had its capillary tube located in the outdoor unit. This would cause liquid refrigerant leaving the outdoor unit to flash. The refrigerant enthalpy at the condenser outlet state was used to calculate the refrigerant-side capacity assuming an isentalphic expansion without heat loss in connecting pipe. This is different from the modified systems of which the capillary tube was removed, and a manual expansion valve was placed at the inlet of the indoor unit. For modified systems, the enthalpy at the expansion valve inlet was used to calculate the refrigerant-side capacity.

b. Unit 10 exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This supports the hypothesis of low compressor efficiency during PRAHA I tests, which also indicates the importance of proper compressor selection.

c. The above is also evidenced by the fact that even with R447B and R452B (zeotropic mixtures), Unit 10 had higher cooling capacity and efficiency than the baseline for the 46°C test condition, as projected in activity 2. The tests at 35°C, however, did not have the same trend.

d. The impact of refrigerant replacement was not very clear, in part due to the hardware change along with it. But because of the differences in saturation curves from the Activity 2 analysis, R32 tends to result in systems with higher efficiency and less charge. The zeotropic mixtures consistently required compressors with larger displacement volumes and even higher mass flow rates for cooling capacities of the same magnitude.

e. Refrigerant fractionation as evidenced by the leak tests, does not appear to a great concern since less than 2% in cooling capacity was observed after the system's re-charge.

f. The Unit 6 modified systems had lower performance than expected from the Activity 2 models. The R32 system configuration exhibited around 10% less flow rate than anticipated, which corresponded to 10% lower capacity. The R454B configuration exhibited a deviation of 5% between model and test due also in part to a 3% flow rate over prediction in the model. Unit 10, on the other hand, exhibited an excellent agreement to the models with less than 2% deviation in cooling capacity.

g. The model's validation adds confidence in the numerical simulation findings and recommendations provided in activity 2.

Table 2: Tests Summary Results.

Syst.	**Test**	**Refrigerant**	**Charge**	**35°C**			**46°C**		
				Cooling Capacity	**Total Power**	**EER**	**Cooling Capacity**	**Total Power**	**EER**
			lb	**BTU/hr**	**kW**	**BTU/hr.W**	**BTU/hr**	**kW**	**BTU/hr.W**
Unit 6	Perfor mance	R32 (Baseline)	3.83	25192	2.43	10.4	23390	3.10	7.54
		R32 (Alternate 1)	4.27	23585	2.12	11.1	21450	2.74	7.84
		R454B (Alternate 2)	5.02	21966	2.10	10.4	21821	2.67	8.17
Unit 10	Perfor mance	R32 (Baseline)*	5.63	34517	3.76	9.18	29005	3.84	7.55
		R447B (Alternate 1)	6.63	32195	3.62	8.88	31073	3.90	7.96
		R452B (Alternate 2)	6.63	28128	3.38	8.33	30292	3.90	7.76
	Liquid Line	Low Charge	4.23	N/A			14216	3.90	3.64
		Re-Charged	6.63				30865	4.16	7.42
	Vapor Line	Low Charge	4.27				15171	3.92	3.87
		Re-Charged	6.77				30587	-	-

**Original baseline values from PRAHA*

4) Conclusions: This report presented a comprehensive set of activities with the objectives of advancing the PRAHA program. The original scope and schedule were modified during the project as new findings and challenges surfaced. The tests that were carried out for PRAHA-I, while sufficient for the purpose of measuring capacity and energy efficiency for the purposes of PRAHA-I, did not have enough essential data to enable a complete cycle evaluation for optimization purposes. This is primarily due to using standard test rig on systems with critical hardware configuration differences. The analyses presented in Activity 2 (design assessment through modeling) provided good insights on adequate component design and/or selection for proper system functioning when using novel refrigerants. The tests in activities 3-5 partially served as validation for the models developed, and as check for previous test data from PRAHA I. The final recommendations for future development are listed as follows:
    a. Establish a baseline system by conducting comprehensive testing including measurements and metrics not typically performed in energy certification tests. Furthermore, testing systems with different configurations require custom test rigs as such to adequately measure working fluid's states to avoid mischaracterization of the operating conditions and performance. Such approach is considerably more labor-intensive which should be factored in the scope in future developments.
    b. Using alternate low-GWP refrigerants is viable and can be competitive to commonly used pure refrigerants but doing so requires proper component design and selection; compressor and expansion device particularly. Drop-in replacement without hardware change is never recommended as evidenced by the change requirements in Activity 2 and performance tests in the subsequent activities.
    c. It is recommended to always perform numerical simulations, and to conduct at least some level of "soft" optimization analyses that will provide information for an educated system re-design / retrofit at much lower costs than gradual trial-and-error changes.
    d. Always test the modified systems with the same instrumentation as the baseline, however mindful of the modifications as such to properly place sensors to obtain adequate readings as suggested in item a above.

2. Introduction

Over the past several years through the Promoting low- Global Warming Potential (GWP) Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA-I) project, 18 different prototypes have been developed and compared to respective baselines to support the assessment of alternative lower-GWP refrigerants for air-conditioning applications. Since the work originally started in 2012, researchers have identified gaps in the performance and operation of the PRAHA-1 prototypes. These gaps include the need to redesign and optimize prototype air-conditioning units, evaluate new alternative refrigerants, and improve component selection. As such, a new project, *Advancing the Designs of PRAHA-I for Meeting or Exceeding the Baseline Designs Performance*, is desired.

The objectives of this project include the following:

4) Evaluate the design limitation of the PRAHA-I prototypes;
5) Optimize and physically evaluate selected prototypes with new refrigerants not evaluated during PRAHA-I; and,
6) Characterize leaks.

The project is divided into six activities namely:

- **Activity 1 – Analyzing the Design of PRAHA-I Prototypes**: evaluate systems performance from selected units tested in PRAHA-I, and assess potential design improvements
- **Activity 2 – Design Improvement**: improve design of specific units targeting higher efficiencies while using alternate low-GWP refrigerants
- **Activity 3 - Prototype Units Fabrication**: modify the a sub-set of the units according to modifications proposed in Activity 2
- **Activity 4 - Evaluation of the Optimized Prototypes**: conduct performance tests on modified units at standard and high ambient temperature conditions (35°C and 46°C)
- **Activity 5 - Analyzing Leaks of Alternatives**: simulate refrigerant leakage and evaluate possible impact of zeotropic mixtures fractionation on performance
- **Activity 6 - Reporting and Data Management**: simulation and test data processing, preparing progress and final reports

3. Activity 1 - Analyzing the Design of PRAHA-I Prototypes

Activity 1 was comprised of three major tasks including: reception of 12 physical units at the OTS facility followed by visual inspection and parts identification; review of performance test reports from PRAHA I tests; and lastly, analyze data and identify, for units of interest, opportunity for improvement targeting higher performance and minimal charge. OTS has completed this activity and an executive summary of the findings are presented herein.

3.1. Physical Units

All 12 units of interest to this project (Table 3) were received on November 8th, 2018. Visual inspection indicated no evident signs of damage. Relevant information to the project such as compressor model, heat exchanger (HX) geometry and circuiting, as well as expansion device were also received.

Table 3: Unit Specifications Summary.

Category	Unit #	Ref.	Designed Capacity Btu/h	Measured Cap. Btu/h	Voltage	Ref. (New designs)	Ref. (Tests)
Window	**1**	**L-20 (R-444B)**	**18,000**	**19,104**	**208-230/60/1**	**L-20, R454C, R290, R457A**	**R290**
	2	L-20 (R-444B)	18,000	16,924	208-230/60/1		
	3	DR-3 (R-454C)	18,000	18,063	208-230/60/1		
Decorative splits	**4**	**R-290**	**24000 (18,000)**	**19,000**	**208-230/60/1**	**R-290**	**R-290**
	5	R-32	24000 (18,000)	19,328	208-230/60/1		
	6	**R-32**	**24,000**	**25,456**	**208-230/60/1**	**R32, R459A**	**R32, R459A**
	7	L-41 (R-447A)	24,000	24,830	208-230/60/1		
	8	L-20 (R-444B)	24,000	22,740	208-230/60/1		
	9	DR-3	24,000	14,638	208-230/60/1		
Ducted splits	**10**	**R-32**	**36,000**	**35,500**	**220-240/50/1**	**R447B, R452B**	**R447B, R452B**
	11	L-20	36,000	36,553	220-240/50/1		
	12	DR-3 (R-454C)	36,000	33,032	220-240/50/1		

3.2. PRAHA-I Performance Reports Assessment

OTS received a complete package of files containing the performance reports for all units tested in PRAHA I. The tests conducted in PRAHA I were meant to assess high-level performance of these units focusing on a large control volume where only total energy in and out was evaluated. As such, these tests were not comprehensive in terms of measurements for cycle analysis required in PRAHA II. Refrigerant side measurements, in most cases, were very limited (few pressure and temperature measurements and no flow rates); thus, it is not possible to fully characterize the cycle and perform energy balances between air and refrigerant sides of the system. Common issues found in the reports include:

- Tag mislabeling and / or mismatching sensor location and tag
- No independent outdoor capacity reported – typically reported the same as indoor capacity
- Missing energy balance checks
- Missing measurement on either airside pressure drop and temperature or fan power
- Inconsistent reported measurements with thermophysical properties for units tested with L-20
- Systematic inconsistency in reported superheat and subcooling
- Missing measurements on refrigerant side at evaporator inlet
- Missing temperature and/or pressure measurements on refrigerant side
- Missing refrigerant mass flow measurements

A summary of the original PRAHA-1 data and results of the data reduction are provided under separate documentation.

3.3. Hardware Improvement Assessment

3.3.1. Heat Exchanger (HX) First Order Analysis (FOA)

This section outlines a FOA for the HXs of Units 1, 4, 6 and 10 to identify improvement potential. The project's objective, as stated above, is to improve performance while minimizing charge. One way of addressing both objectives is by reducing tube / channel diameter. Heat transfer coefficients are inversely proportional to surface hydraulic diameters, however, so is pressure drop. Smaller tubes result in more compact (C = surface area / footprint volume), with reduced internal volume, HXs.

A qualitative analysis using values from literature was carried out to demonstrate the relative impact of diameter over abovementioned metrics, specifically: heat transfer coefficient, compactness and overall thermal conductance (UA). The left-hand side plot in Figure 1 show three curves inversely proportional to the diameter; a 5mm tube can achieve, in this example, 70% greater UA than a conventional 9.5mm, within the same footprint volume (or cabinet).

These are further explored to illustrate the impact on a system level. Systems respond to UA of both condenser and evaporators, but for the purposes of this analysis, condenser only is considered. The UA represents the overall thermal conductance, which will impact the approach temperatures in the system (ΔT_{app}). If the heat rejection is kept constant, the higher the UA, the smaller are the ΔT_{app}'s, thus allowing the condenser to operate in lower pressure levels, which will consequently increase the system performance. An example using a hypothetical R32 cycle with an EER of 12 as base is shown in the right-hand side plot in Figure 1. Performance improvement is limited by the Second Law, when the approach temperatures near zero; however, in this illustration, the EER has potential to increase in over 20% with better condenser design alone.

It is imperative to note that the results presented in this section are for **illustration purposes only**. Further in this report it is presented in more detail a re-design framework, applied to the units of interest in this project, using the metrics outlined in this section.
Unit 1 already had a 5mm condenser, which limits the options for HX re-design. Unit 6 had a 7mm HX on both the indoor and outdoor units, which allows some room for improvement if reducing to 5mm. Lastly, both coils for Unit 10 had 9.5mm tubes, thus there is greater potential for charge reduction and performance improvement for that unit in particular.

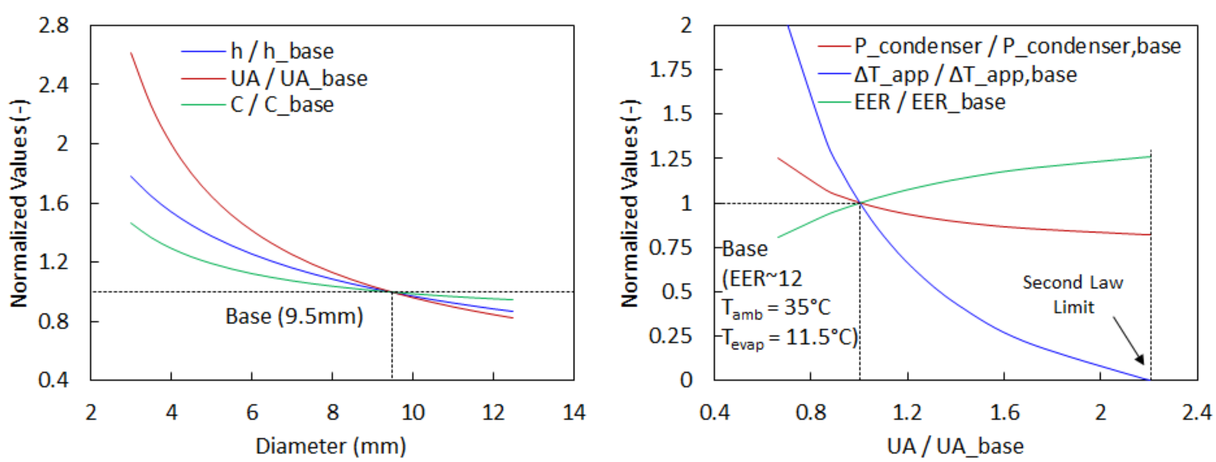

Figure 1. Heat Exchangers FOA.

3.3.2. Compressors

The existing units mostly use compressors sized specifically for R410A or R22 and in some cases custom made for this effort. There is, however, opportunity for a better compressor selection when migrating from R32 to R454B or R447B on Units 6 and 10, respectively.

3.3.3. Expansion Devices

Expansion devices such as TXV's and EXV's may allow for better control and reduced losses in connecting pipes if located near the evaporator. Some units, such as 6 and 10, have a capillary tube in the outdoor unit, which forces the refrigerant to travel in two-phase along the connecting pipes, and at lower temperatures, thus increasing pressure drop and heat gain.

3.3.4. Fan and Blower

Replacing the fan and blower may be necessary if newly designed HXs offer considerable change in pressure drop over the baseline since the flow rates are kept constant. The lack of test data on pressure drop forces us to rely on predicted values only. These will be considered for replacement as a last priority.

3.3.5. Units Component Modification Potential

Table 4 shows the detailed existing components for the units of interest for modification.

Table 4: Units 1, 4, 6 and 10 Components.

System	Unit 1	Unit 4	Unit 6	Unit 10
Refrigerant	**R444B**	**R290**	**R32**	**R32**
Compressor	HIGHLY SL260DG-C8EU	HIGHLY PSH356DG-C8DU3	GMCC KSG226N1UMT	Copeland ZP42K5E-PFJ-XXX
Condenser	5mm Louver TFHX	9.5mm Wavy TFHX	7mm Louver TFHX	9.5mm Louver TFHX
Expansion Device	Capillary Tube	Capillary Tube	Capillary Tube	Capillary Tube
Evaporator	9.5mm Louver TFHX	7mm Louver TFHX	7mm Slit TFHX	9.5mm Louver TFHX

3.4. Conclusions and Recommendations

The first part of this activity regarded data analysis and processing from the original tests conducted in the original PRAHA-I project, which was designed to conduct testing and comparison of cooling capacity vs. EER for the prototypes against the baseline units from same manufacturers. Since limited certification tests were required then, more testing parameters would have been needed to support the optimization and/or redesign process within the scope of PRAHA-II. The second part pertained assessing potential hardware modifications that could result in higher performance and less charge, with the intent of replacing the original refrigerants with alternative, low-GWP ones. The key conclusions and recommendations are:

1- Certification laboratories, such as the one used for testing the units in PRAHA I, provide limited information for the purposes of product design and development. For future reference it is recommended that for research-oriented efforts such as this one, the units undergo a more rigorous testing process along with full characterization of the system and its individual components operating conditions and performance.

2- In applications of high ambient temperatures, it is expected that performance will degrade as compared to operating under more temperate conditions and the resultant impact on performance must be considered. The key components for performance improvement identified herein were the compressor, condenser and expansion device.

a. At higher temperatures, the saturation temperatures and refrigerant density at compressor's suction port can be very different than that from the rated conditions. Larger displacement volumes and efficiency curves optimized for higher pressure lifts might be required. Therefore, the proper selection of the compressor is paramount.
b. A better performance condenser will reduce the approach temperature between refrigerant and air, helping the compressor not to discharge refrigerant at very high pressure and temperatures, which degrade performance.

3- At high ambient conditions, the system is forced to operate in higher pressure lift than at rated conditions, but still requires a certain refrigerant mass flow rate. Passive devices such as capillary tubes and orifices may not be able to provide enough expansion to allow the system to operate in higher temperature conditions. An active expansion device such as EXV's can adequately control operating conditions and maintain stable superheat.

4. Activity 2 - Design Improvements

The details of modeling and simulation results are provided in a separate document submitted in conjunction with this one, while in this section only the summarized performance results are presented.

4.1. Hardware

A general design improvement assessment was presented in the report for Activity 1, focusing on the units of interest to this study. A first order analysis on the HX's showed that moving towards smaller hydraulic diameter tubes can be beneficial from a material savings and charge reduction standpoint. Units 4 and 10 use conventional 9.5mm diameter tube condensers (Table 4), making them good candidates for condenser replacement with either a smaller tube diameter or a microchannel heat exchanger (MCHX). The compressors used on Units 1, 4 and 6 do not have available performance maps making it difficult to assess their fitness for the system. The focus of this study is on proper compressor selection and condenser re-design.

4.2. Refrigerant

R32 and R290 have wide saturation regions (Figure 2 and Figure 3) putting them at an advantage since they may operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer. Their cycles may get closer to that of the ideal Carnot cycle compared to refrigerants with narrower saturation.

Amongst the blends investigated for Unit 1, R444B has the widest saturation region while also having the highest temperature glide (Figure 4). The latter is typically not beneficial, in particular for evaporators, but it may help the condenser. The glide enables the refrigerant temperature profile to get closer to the air temperature profile without crossing (Figure 4). From a thermodynamic perspective, this means R444B can have its condensing pressure reduced further, resulting in higher theoretical COP.

For Units 6 and 10, the investigated blends, although having narrower saturation than the baseline R32, have similar thermophysical characteristics (Figure 3) with lower temperature glides (Figure 4) making them more competitive from a capacity and performance perspective.

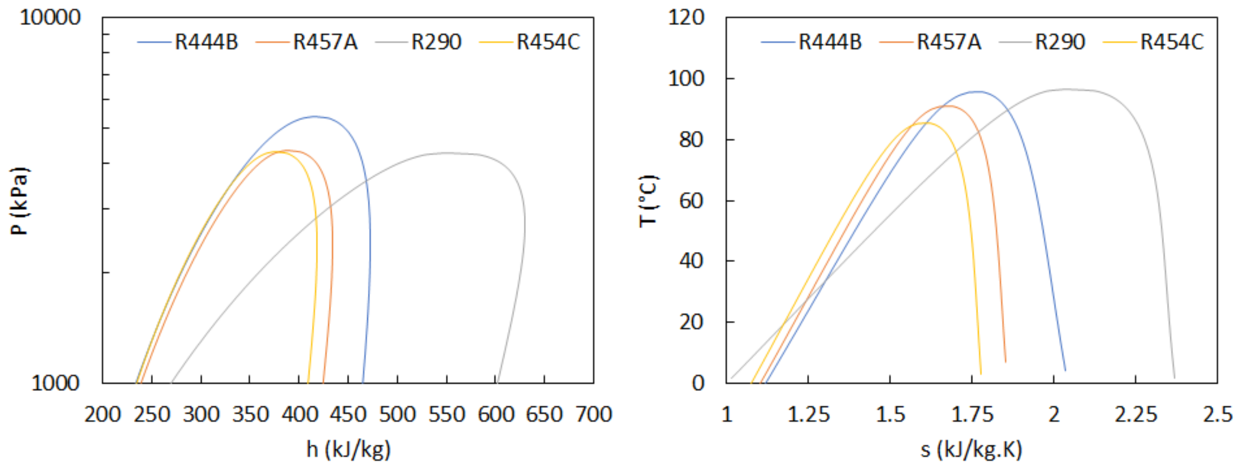

Figure 2. Refrigerants Investigated for Units 1 and 4.

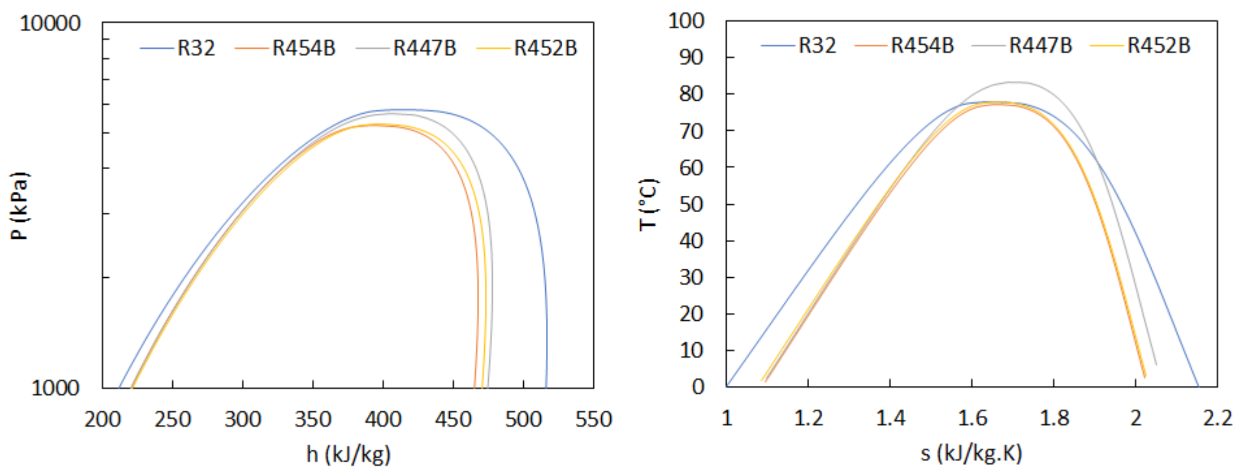

Figure 3. Refrigerants Investigated for Units 6 and 10.

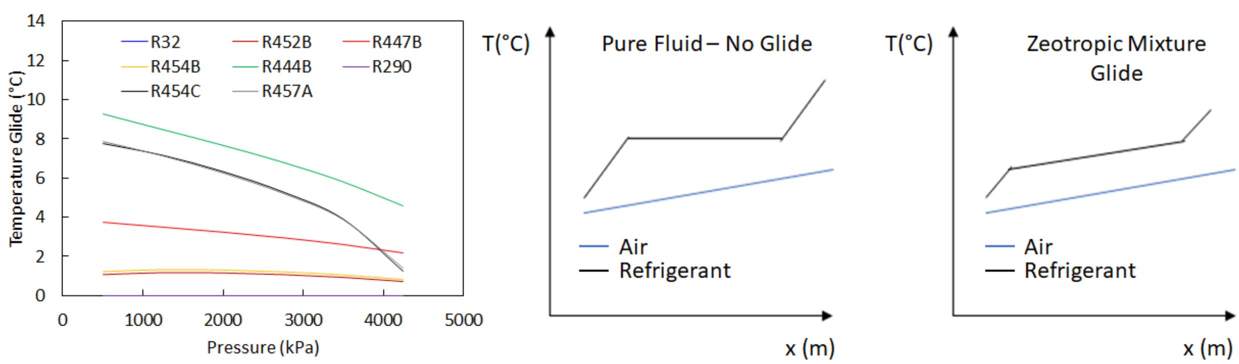

Figure 4. Refrigerant Temperature Glides.

4.3. System Design Modification Framework

The systems' re-design herein presented ultimately consists of a retrofit of the existing units by properly designing and selecting components that can be replaced as drop-ins, with minimal or no modification of the packaging (cabinets). In other words, any component replaced must occupy the same envelope as the baseline component. The focus of the re-design is on:

- Compressor
- Condenser, and
- Expansion valve

The evaporator designs were not changed for two main reasons: a) some are custom-made wrap-around the blower units, such as in Unit 6, making it harder to quickly find an off-the-shelf option; and, b) the goal is to deliver the same cooling capacity while improving efficiency. For the latter, there's more room for improvement in the condenser by reducing condensing pressure, assuming the evaporator can already deliver the expected capacity.

The fans and blowers were also not considered for change, in part due to the lack of information on the performance curves from the baseline models, but also due to potential high cost and lead time for replacement with secondary impact on performance since 80-90% of the power consumed comes from the compressor.

The first step to assess the level of performance required for each component is to investigate an improved theoretical cycle, which will indicate how much COP improvement can be expected, as well as refrigerant flow rate needs and HX size (UA). To improve the performance of a vapor compression cycle, the pressure lift between evaporating and condensing pressures must be reduced. Consequently, the approach temperatures between air and refrigerant will be reduced as well (Figure 5), thus the thermal capacitance of the heat exchangers must increase. Furthermore, the closer to the saturation region, the closer the cycle reaches the ideal Carnot efficiency (Figure 6).

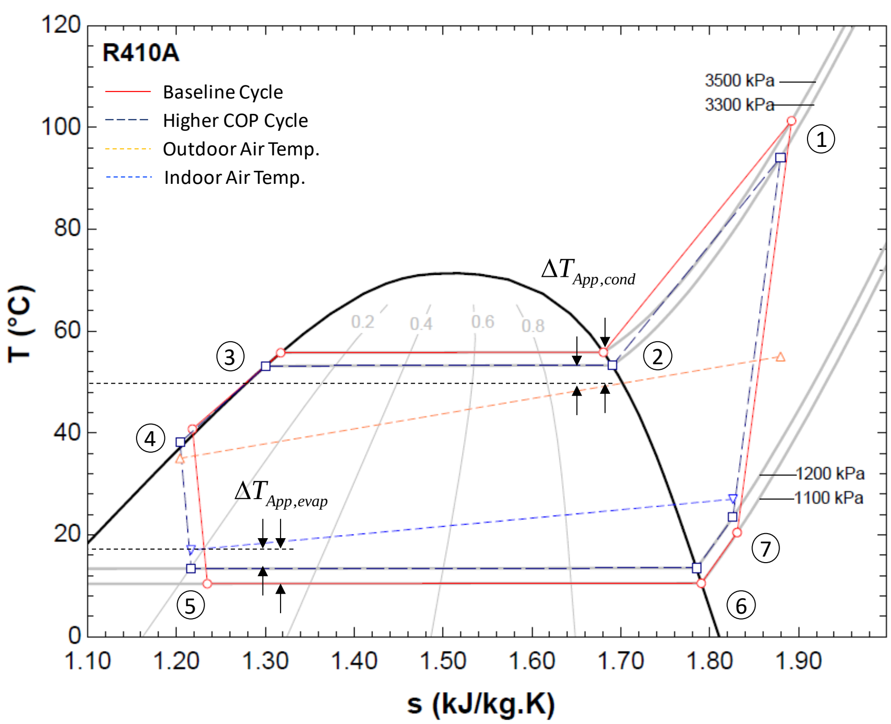

Figure 5. Illustrative T-s diagram for baseline and improved cycles.

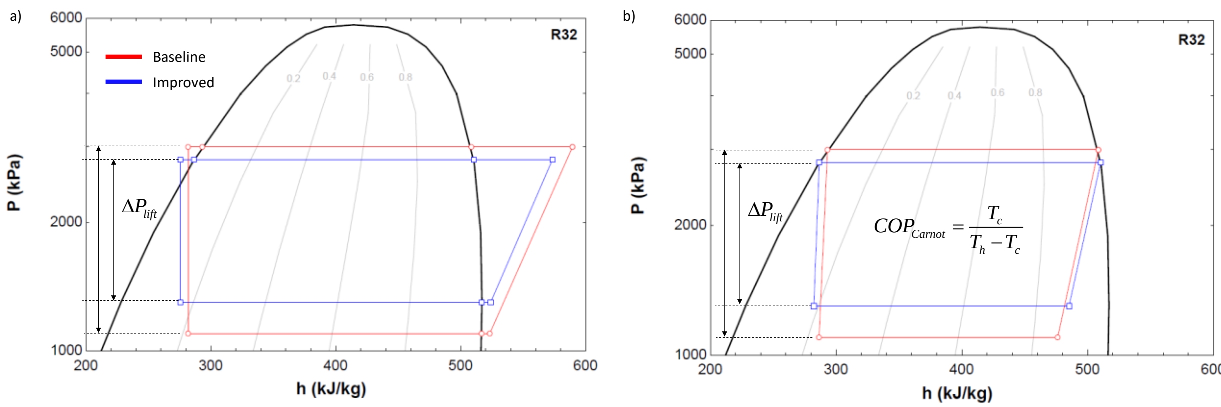

Figure 6. P-h Diagrams Illustrating COP Improvement: a) Real Cycle; b) Ideal Cycle (Carnot).

The system design framework is performed according to Figure 7.

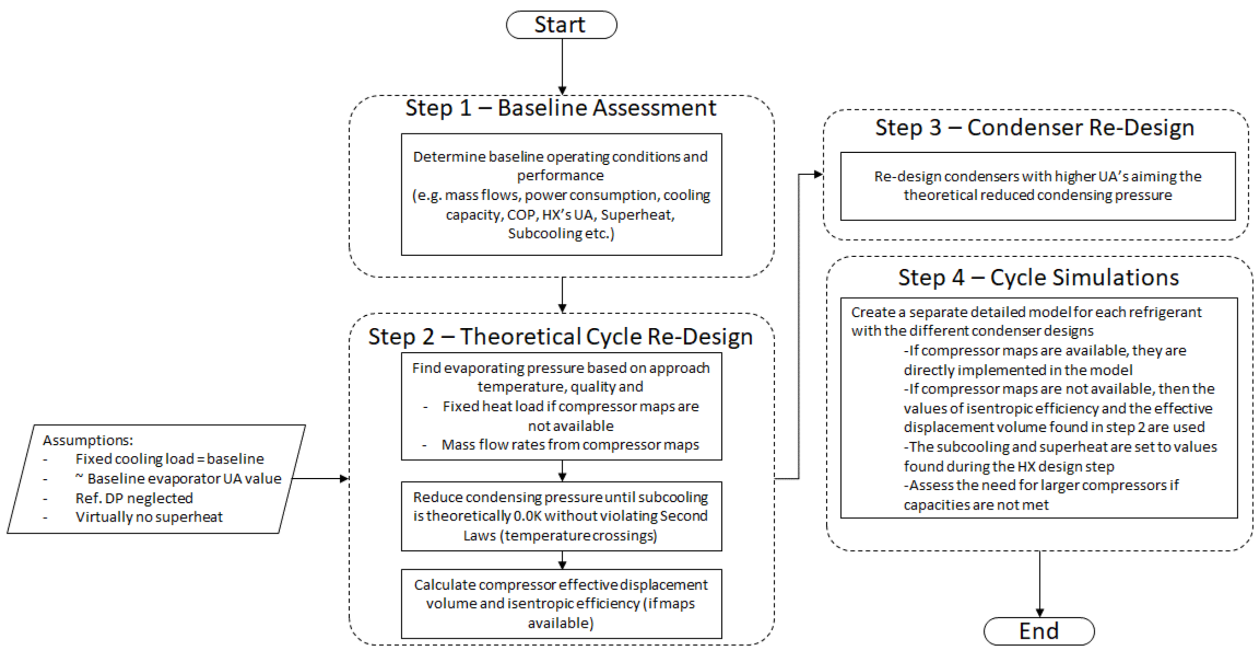

Figure 7. System Re-Design Framework,

4.3.1. Compressors

Modeling compressors are handled in two possible ways, as suggested previously: using performance maps when available or using fixed isentropic efficiency and effective displacement volume. For the larger capacity units (6 and 10), performance maps were provided. Although these compressors were originally designed for R410A refrigerant they may operate – not necessarily optimally – with other refrigerants. Compressor manufacturers supporting this project used proprietary simulation tools, with aid from available empirical data (tests with other refrigerants), to develop theoretical maps for the various refrigerants of interest (Table 5) and made them available to OTS for modeling purposes. It is understood that the predictions are for reference only, and the compressor manufacturer does not guarantee performance for any refrigerants for which the compressors haven't been fully tested.

Table 5: Compressor Models.

Model	Capacity (BTU/hr)	Frequency (Hz)	Refrigerants
ZP20K5E-PFV	24,000	60	R32, R459A, R454B, R410A
ZP21K5E-PFV	24,000		
ZP31K6E-PFV	36,000	50/60	R447B, R452B, R454B, R410A
ZP34K6E-PFV	36,000		

For the smaller units (1 and 4), which were re-designed using R290 (Propane), compressor performance maps were not available. The approach for these units then was to set a target isentropic efficiency of 0.7 (baseline data suggests that the compressor efficiencies ranged from 0.55 to 0.65). The required mass flow rate is calculated based on capacity in the theoretical cycle model described above. From there, the effective displacement volume can be determined (eq. (1))[1]. The latter serves to determine whether a system can use the same compressors for different refrigerants.

$$V_{eff} = \eta_{vol} \cdot V_{disp} = \frac{\dot{m}_{required}}{f \cdot \rho_{suction}} \tag{1}$$

4.3.2. HX Design and Selection

The condensers design procedure takes into consideration the following:

- **Face area**: baseline face area must be preserved or at most reduced. Furthermore, the aspect ratio must also match that of the baseline so the HX can be drop-in replaced in the same cabinet.
    - Find the number of tube rows and tube length to match as closely as possible to tube face area and aspect ratio

- **Airside pressure drop and flow rate**: the test data from reports contain only air flow rate measurements, while no information on pressure drop is provided. Additionally, the fan performance curves are also not available, which limits the ability to find the exact operating condition. The baseline models provide an estimate prediction for the pressure drop, which is used as reference.

- **Thermal performance**: this step must be iteratively conducted with the previous step, as such for each design change the air flow rate and capacity are evaluated under the new conditions found in the theoretical cycle re-design.
    - Gradually increment the condensing pressure until attainable performance is achieved. This process is done iteratively using the theoretical cycle model, to find new expected operating conditions for evaporating pressure, superheat, subcooling and refrigerant flow rate.

- **HX Form**: as indicated previously, the HX design is constrained by cabinet dimensions as well as form. In the case of units 1 and 4, the condensers are flat coils placed 90° inside the cabinet (Figure 8), which makes it simpler for drop-in replacement as long as new designs have the same overall dimensions. For units 6 and 10, however, the condensers are L-shaped inside the cabinet (Figure 8). Forming coils is widely done, however, for custom coils it may be a challenge, in particular for MCHX. For this reason, the MCHX designs for units 6 and 10 are sized for a full-face area, assuming the coil can be formed, and a second design that is a single flat slab placed in longer side of the "L" shape(Figure 9).

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[1] Variable definitions in the Nomenclature list after final conclusions section in this document.

- **HX Name Tag Convention:** for practical purposes, the HX's will be tagged according to the following W XX YY Z
    - **W:** B = Baseline or N = New Design
    - **XX:** TF **=** Tube-Fin or MC = Microchannel
    - **YY:** D# = Tube Diameter or Height
    - **Z:** R = Reduced Face Area
    - **Example:** New Tube Fin Design with 5.0mm diameter with same face area as the baseline → NTFD5

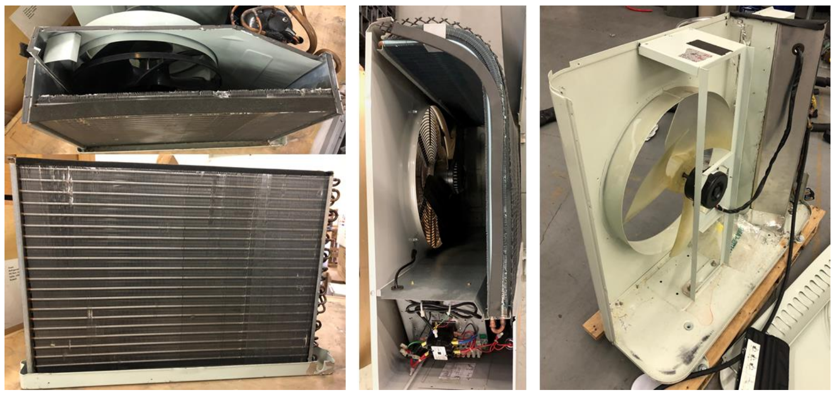

Figure 8. Condenser Forms: Unit 1 (left), Unit 10 (center), Unit 6 Cabinet (right).

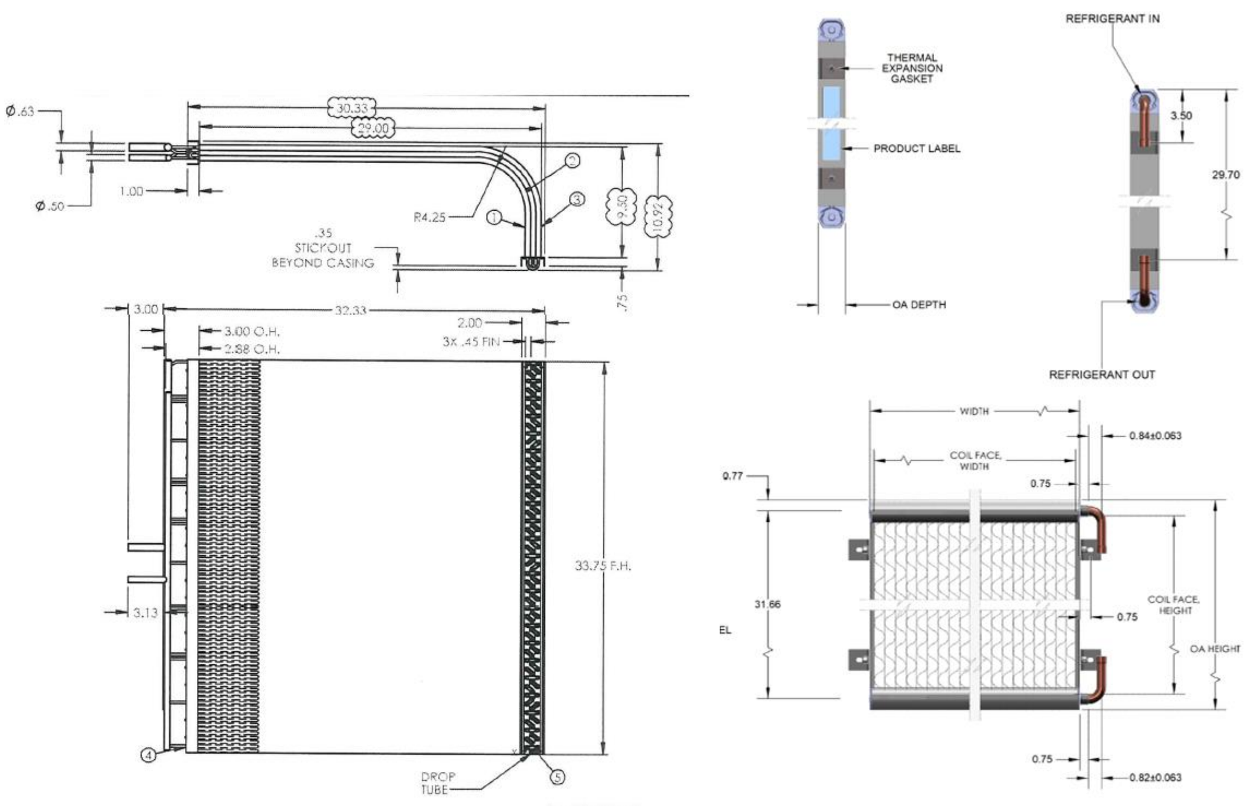

Figure 9. HX Form Examples: L-shape (left), Flat (right).

4.3.3. System Design

In the final step, the modified systems are evaluated holistically through system level modeling and simulation using an in-house Steady-State vapor compression cycle software that has the capability to integrate with the HX and compressor models (performance maps, generic etc.). For each modified system and each refrigerant, a system model was created.

4.4. Modified Systems Results Summary

The final results of Activity 2 are summarized in Table 6. For more detailed results in the framework steps refer to APPENDIX A .

4.5. Conclusions and Recommendations

This section presents a systematic approach based on first order analysis providing educated guidance towards the direction of more efficient systems with fewer simulations and minimal changes to the systems. The study includes a wide variety of refrigerants as well as condenser designs and compressor model options. Given the challenges with original test data the baseline models serve as a numerical reference only. The findings are strictly valid to comparisons against the baseline models and OTS does not guarantee that results would be reflected in actual systems as herein reported. The key conclusions and recommendations are:

1- R290 and R32 have wider saturation regions allowing the system to operate with smaller superheat and subcooling, while benefiting from two-phase heat transfer.

2- Refrigerants with high temperature glide may require new heat exchanger (HX) designs, namely condensers. The original designs proved to be sufficiently effective to allow for most systems to operate with the different refrigerants, however, better designs would allow for higher system efficiency and potentially less charge. HX designs are severely constrained by allowed envelope dimensions. A complete system re-design would provide an opportunity for designing HX's with even higher efficiency.

3- The results of this analysis suggest that for an effective use of alternate low-GWP refrigerant, a proper compressor selection must be accompanied with it. Higher isentropic efficiencies are desired for higher temperatures, but most importantly, the displacement volume requirements can vary considerably from one refrigerant to another.

4- It is also imperative that having an active expansion device (preferably an EXV) to not only allow for more controlled superheat, but also to enable the unit to run with different refrigerants with very different thermophysical properties.

Table 6: Activity 2 Results.

General Information			Hardware					Ref.	Performance			
System	Rated Capacity (@35°C)	System Configuration	Compressor		Condenser		Expansion Device		Cooling Capacity (@46°C)		EER (@46°C)	
-	BTU/hr	-	Effective Disp. Vol. (cm³)*	Efficiency (-)	Type	Effectiveness (-)	Type	-	BTU/hr	%	BTU/hr. W	%
Unit 1	18000	**Baseline**	**19.8**	**0.66**	**Tube-Fin (5mm Tube)**	**0.20**	**Passive**	**R444B**	**17403**	**0.00%**	**7.4**	**0.00%**
		Alternate 1	25.9	0.70	Same as Baseline	0.35	Active (EXV)	R290	17639	1.40%	8.01	8.20%
		Alternate 2	24.8	0.69		0.26		R454C	18104	4.00%	7.31	-1.30%
		Alternate 3	19.6	0.70		0.23		R444B	18140	4.20%	8.14	9.90%
		Alternate 4	25.3	0.68	MCHX	0.24		R457A	17749	2.00%	7.63	3.10%
Unit 4	24000	**Baseline**	**26.4**	**0.61**	**Tube-Fin (9.5mm Tube)**	**0.24**	**Passive**	**R290**	**17940**	**0.00%**	**7.52**	**0.00%**
		Alternate 1	26.3	0.70	Tube-Fin (5mm Tube)	0.26	Active (EXV)	R290	18147	1.20%	9.12	21.40%
		Alternate 2	37.9	0.70		0.20		R290	24120	34.40%	6.72	-10.60%
Unit 6	24000	**Baseline**	**16.0**	**0.60**	**Tube-Fin (7mm Tube)**	**0.12**	**Passive**	**R32**	**23115**	**0.00%**	**8.46**	**0.00%**
		Alternate 1	16.9	0.65	Tube-Fin (5mm Tube)	0.15	Active (EXV)	R32	23798	3.00%	9.41	11.20%
		Alternate 2	18.4	0.67		0.19		R454B	22894	-1.00%	9.71	14.80%
		Alternate 3	19.0	0.70		0.17		R452B	23702	2.50%	9.6	13.50%
Unit 10	36000	**Baseline**	**19.6**	**0.44**	**Tube-Fin (9.5mm Tube)**	**0.13**	**Passive**	**R32**	**29005**	**0.00%**	**6.39**	**0.00%**
		Alternate 1	22.3	0.65	Tube-Fin (5mm Tube)	0.25	Active (EXV)	R447B	30478	5.10%	9.43	47.50%
		Alternate 2	23.0	0.67		0.25		R452B	30796	6.20%	10.27	60.70%
		Alternate 3	23.3	0.67		0.25		R454B	30809	6.20%	10	56.50%

** Product of displacement volume and volumetric efficiency*

5. Activities 3, 4 & 5 - Prototype Units Fabrication, Evaluation of the Optimized Prototypes and Analyzing Leaks of Alternatives

Activities 3-5 officially began in April 2019 when the first round of tests on modified Unit 6 were carried out. Initial tests resulting in unsuccessful outcomes leading OTS to change the system modifications and the scope. Additional information found in APPENDIX B . The detailed test data and charge optimization for Units 6 and 10 are presented in APPENDIX C through APPENDIX E . Comparisons between Activity 2 model validations and experimental data are presented in APPENDIX F .

5.1. Unit 6

Some modifications were made to Unit 6 to improve its efficiency. The baseline compressor was replaced with alternate models to account for the change in refrigerant and to improve efficiency. The compressor used with R454B had a higher displacement volume than the one used with R32. Furthermore, the capillary tubes were replaced with a manual TXV that was installed directly at the evaporator inlet to increase the cooling capacity of the evaporator. A summary of the design modifications evaluated for Unit 6 is listed in Table 7, while Table 8 and Table 9 show the performance of Unit 6 for baseline and modifications at 35°C and 46°C ambient, respectively. The baseline system performed similar, within 2%, to reported performance in PRAHA I. There is a discrepancy in the measurements from condenser outlet to expansion inlet in the baseline case, since the capillary tube (removed in the modified systems) was located in the outdoor unit The expansion causes the refrigerant to flash in the liquid line thus compromising the readings at the expansion device. For calculation purposes, the condenser outlet enthalpy was used instead of the expansion inlet.

Table 7: Unit 6 Modifications for Testing.

System	Unit 6		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R32	R454B
Compressor	GMCC KSG226N1UMT	Copeland ZP20K5E	Copeland ZP21K5E
Expansion Device	Capillary Tube (Outdoor unit)	Manual Valve (Indoor Unit)[2]	Manual Valve (Indoor Unit)[2]

Cooling capacity for the modified unit with either refrigerant was consistently lower by 6-12% than the baseline. The modified R32 system reportedly showed lower mass flow rate than expected, likely the main cause for the lower-than-expected thermal performance. The R454B system resulted in a poorer performance but was less sensitive to ambient temperature than its R32 counterpart - i.e. cooling capacity was near the same at both 35°C and 46°C, while for R32 there was a ~2,000BTU/hr reduction with the temperature increase. It is also possible that there is a mismatch between thermophysical property library and actual refrigerant properties for R454B which can happen with newer fluids. The libraries need periodic update as more test data become available.

[2] A manual valve was used to mimic a TXV or EXV recommended as component modification in these systems configurations.

Table 8: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**R32**	**R454B**	**-**	**-**
Charge	lb	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	25192	23585	21966	-6.4%	-12.8%
Energy Balance	%	-2.28%	-4.66%	-3.06%	-	-
Compressor Power	kW	2.11	1.79	1.77	-15.1%	-16.2%
Fan Power	kW	0.32	0.33	0.33	2.2%	4.2%
Total Power	kW	2.43	2.12	2.10	-12.8%	-13.5%
EER	BTU/hr.W	10.37	11.12	10.44	7.2%	0.68%

Table 9: Unit 6 - Performance Test Summary for R32 Baseline (OTS) @ 46°C.

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**R32**	**R454B**	**-**	**-**
Charge	lb	3.83	4.27	5.02	11.5%	31.1%
Cooling Capacity	BTU/hr	23390	21450	21821	-8.3%	-6.7%
Energy Balance	%	-1.78%	-4.42%	-7.61%	-	-
Compressor Power	kW	2.71	2.32	2.25	-14.2%	-16.6%
Fan Power	kW	0.40	0.42	0.42	5.3%	5.3%
Total Power	kW	3.10	2.74	2.67	-11.7%	-13.8%
EER	BTU/hr.W	7.55	7.84	8.17	3.8%	8.2%

5.2. Unit 10

Applying what was learned in the initial modifications to Unit 6, modifications to Unit 10 were limited to include the compressor and expansion device only. Unlike Unit 6, however, the re-test of the baseline system was not successful; refer to APPENDIX D for additional information. However since Unit 6 baseline re-test showed good reproducibility from original data, it is assumed that the Unit 10 original baseline is appropriate for comparison against the modified system configurations. A summary of the design modifications evaluated for Unit 10 is listed in Table 10. The detailed test data is presented in APPENDIX E .

At 35°C the modified units exhibited almost 20% less cooling capacity with 10% less power consumption, resulting in up to 11% less EER (Table 11). These results were not unexpected since the modified units were re-designed using the 46°C temperature, when the baseline system's performance showed a great degradation of performance. At 46°C condition, the tests exhibited 2-5% greater cooling capacity with up to 12% less power consumption compared to the baseline, which was equivalent to 13-17% greater system performance.

In Activity 2 the compressor power consumptions were underestimated, as well as the total fan power consumption, leaving the impression the overall performance improvement would considerably be greater than the observed. The cooling capacity, on the other hand, was predicted with less than 2% deviation from test data, validating at least the models created.

Table 10: Unit 10 Modifications for Testing.

System	Unit 10		
	Baseline	Alternate 1	Alternate 2
Refrigerant	R32	R447B	R452B
Compressor	Copeland ZP42K6E	Copeland ZP34K5E	Copeland ZP31K5E
Expansion Device	Orifice	Manual Valve	Manual Valve

Table 11: Unit 10 - Performance Test Summary for R32 Baseline @ 35°C.

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**447B**	**452B**	**-**	**-**
Charge	lb	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	35543	32195	28128	-9.42%	-20.86%
Energy Balance	%	---	7.52%	-3.29%	-	-
Compressor Power	kW	-	2.67	2.4	-	-
Fan Power	kW	-	0.95	0.98	-	-
Total Power	kW	3.761	3.62	3.38	-3.75%	-10.13%
EER	BTU/hr.W	9.451	8.894	8.322	-5.89%	-11.94%

Table 12: Unit 10 - Performance Test Summary for R32 Baseline @ 46°C.

		Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)	Alt. 1 vs. Baseline	Alt. 2 vs. Baseline
Refrigerant	**-**	**R32**	**447B**	**452B**	**-**	**-**
Charge	lb	5.625	6.625	6.625	17.78%	17.78%
Cooling Capacity	BTU/hr	29633	31073	30292	4.86%	2.22%
Energy Balance	%	---	4.21%	1.21%	-	-
Compressor Power	kW	---	3.18	2.93	-	-
Fan Power	kW	---	0.95	0.97	-	-
Total Power	kW	4.466	4.13	3.9	-7.52%	-12.67%
EER	BTU/hr.W	6.64	7.52	7.76	13.33%	16.95%

5.3. Leak Tests

In the interest of time the leak tests were conducted only on Unit 10 for R447B. The choice of refrigerant was based on temperature glide, where R447B exhibits the highest glide amongst the refrigerants evaluated between Unit 6 and Unit 10 (refer to Figure 4). The leak tests were conducted to closely represent field operation. The procedure applied includes the following steps:

1- Run unit until steady-state is achieved (repeat 46°C performance test), monitoring capacity and subcooling
2- Gradually remove refrigerant from vapor line until capacity is reduced to approximately 50%, if possible
3- Store and weigh removed refrigerant
4- Re-charge with new refrigerant until same subcooling is achieved
5- Compare cooling capacities; if more than 5% deviation is observed, repeat steps 1-4, however in step 2, reduce capacity to 25% only
6- Repeat steps 1-5 for the liquid line

The comparison herein presented refers to a leakage of approximately 30% of charge, while reducing capacity by approximately 50% based on airside only. The leak tests showed less than 2% deviation in cooling capacity after re-charge from both vapor and liquid lines (Table 13). Since the capacity deviation was less than 5%, no further testing for 25% capacity reduction was conducted. The results suggest little impact due to fractionation.

Table 13: Unit 10 – R447B Leak Test Summary Results.

System			*Liquid Line Leak*		*Vapor Line Leak*	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	-	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Charge	lb	6.625	4.27	6.625	4.23	6.77
Cooling Capacity	BTU/hr	31073	14216	30865	15171	30587
Energy Balance	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Compressor Power	kW	3.18	2.93	3.18	2.94	-[3]
Fan Power	kW	0.95	0.98	0.98	0.98	0.98
Total Power	kW	4.13	3.90	4.16	3.92	-[3]
EER	BTU/hr.W	7.52	3.64	7.42	3.87	-[3]

5.4. Conclusions and Recommendations

This section presented the performance tests conducted on units 6 and 10. The key conclusions and recommendations are:

1- Unit 6 re-tested baseline exhibited similar performance to that found in PRAHA I testing. It should be stressed that the baseline unit by design had its capillary tube located in the outdoor unit. This would cause liquid refrigerant leaving the outdoor unit to flash. The refrigerant enthalpy at the condenser outlet state was used to calculate the refrigerant-side capacity assuming an isentalphic expansion without heat loss in connecting pipe. This is different from the modified systems of which the capillary tube was removed, and a manual expansion valve was placed at the inlet of the indoor unit. For modified systems, the enthalpy at the expansion valve inlet was used to calculate the refrigerant-side capacity.

2- Unit 10 exhibited a considerable reduction in power consumption at the high ambient test condition (46°C) as compared to the original test data. This also indicates the importance of proper compressor selection.

3- The higher-than-expected power consumption in the Unit 10 baseline tests is also evidenced by the fact that even with zeotropic mixtures (R447B and R452B), Unit 10 had higher cooling capacity and efficiency than the baseline for the 46°C test condition, as projected in activity 2.

4- Because of the differences in saturation curves from the Activity 2 analysis, R32 tends to result in systems with higher efficiency and less charge when no modifications to the hardware are made. The results showed however, that making appropriate component selection, such as compressors with larger displacement volumes and higher mass flow rates for the zeotropic mixtures, cooling capacities and overall performance were of the same order of magnitude.

5- Refrigerant fractionation as evidenced by the leak tests, does not appear to be a great concern since less than 2% deviation in cooling capacity was observed after the system's re-charge.

6- The Unit 6 modified systems had lower performance than expected from the Activity 2 models. The R32 system configuration exhibited more than 10% less flow rate than anticipated due to performance

[3] Compressor power consumption was not properly recorded for this test; the error was identified after the fact and the team was unable to retrieve that information. While that compromises the assessment of the overall system performance, the deviations are expected to be marginal. The leak test on liquid line suggest minimal impact on power consumption after re-charge, while cooling capacity was reportedly fully recovered after recharge on both leak tests.

maps overprediction, which corresponded to 10% lower capacity. The R454B configuration exhibited a deviation of 5% between model and test due also in part to a 3% flow rate over prediction in the model. Unit 10, on the other hand, exhibited an excellent agreement to the models with less than 2% deviation in cooling capacity.

7- The model's validation adds confidence in the numerical simulation findings and recommendations provided in activity 2.

6. Conclusions

This report presents a comprehensive set of activities with the objectives of advancing the PRAHA program. The original scope and schedule were modified during the project as new findings and challenges surfaced. The tests that were carried out for PRAHA-I, while sufficient for the purpose of measuring capacity and energy efficiency for the purposes of PRAHA-I, did not have enough essential data to enable a complete cycle evaluation for optimization purposes. This is primarily due to using standard test rig on systems with critical hardware configuration differences. The analyses presented in Activity 2 (design assessment through modeling) provided good insights on adequate component design and/or selection for proper system functioning, when using novel refrigerants.

The final recommendations for future development are listed as follows:

1- Establish a baseline system by conducting comprehensive testing including measurements and metrics not typically performed in energy certification tests. Furthermore, testing systems with different configurations require custom test rigs as such to adequately measure working fluid's states to avoid mischaracterization of the operating conditions and performance. Such approach is considerably more labor-intensive which should be factored in the scope in future developments.

2- Using alternate low-GWP refrigerants is viable and can be competitive to presently used refrigerants but doing so requires proper component design and selection; compressor and expansion device particularly. Drop-in replacement without hardware change is never recommended as evidenced by the change requirements in Activity 2 and performance tests in the subsequent activities.

3- It is recommended to always perform numerical simulations, and to conduct at least some level of "soft" optimization analyses that will provide information for an educated system re-design / retrofit at much lower costs than gradual trial-and-error changes.

4- Always test the modified systems with the same instrumentation as the baseline, however mindful of the modifications as such to properly place sensors to obtain adequate readings as suggested in item 1 above.

Nomenclature

COP	Coefficient of Performance	-
D_o	Tube Outer Diameter	mm
f	Frequency	Hz
FPI	Fins per Inch	1/in
h	Enthalpy	kJ/kg
h_t	Tube Height	mm
HX	Heat Exchanger	-
$\dot{m}$	Mass Flow Rate	kg/s
MCHX	Microchannel Heat Exchanger	-
P	Pressure	kPa
P_l	Tube Longitudinal Pitch	mm
P_t	Tube Transverse Pitch	mm
s	Entropy	kJ/kg.K
T	Temperature	°C
TFHX	Tube-Fin Heat Exchanger	-
UA	Thermal Conductance	kW/K
V	Volume	m^3
w_t	Tube Width	mm
η_{vol}	Volumetric Efficiency	-
ρ	Density	kg/m^3

APPENDIX A - Activity 2 Design Framework Results

Table 14: Unit 1 – Theoretical Cycle Re-Design Summary.

System		*Baseline*	*Alternative 1*	*Alternative 2*	*Alternative 3*	*Alternative 4*
Case	-	**Simulation**	**Target**			
Refrigerant	**-**	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Condenser	-	BTFD5	-	-	-	-
Compressor	-	SL260DG-C8EU	-	-	-	-
Cooling Capacity	BTU/hr	17403	17477	17477	17477	17477
Compressor Power	kW	1.92	1.49	1.49	1.33	1.43
Fan Power	kW	0.43	0.43	0.43	0.43	0.43
Total Power	kW	2.35	1.92	1.93	1.76	1.86
COP	-	2.17	2.66	2.66	2.91	2.75
COP Gain	-	1.00	1.23	1.23	1.34	1.27

Table 15: Unit 1 – HX Analysis Summary

Condenser		**R444B**		**R290**		**R454C**		**R457A**	
Inputs		***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***	***BTFD5***	***NMCD2***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56
Refrigerant Pressure	kPa	2875.0	2875.0	2170.7	2170.7	2436.4	2436.4	2183.9	2183.9
Saturation Temperature at Inlet	°C	61	61	61	61	61	61	61	61
Refrigerant Temperature	°C	110.00	110.00	110.00	110.00	110.00	110.00	110.00	110.00
Mass Flow Rate	kg/s	0.03	0.03	0.02	0.02	0.03	0.03	0.03	0.03
Outputs									
Heat Load	W	7512.9	7441.2	8232.4	8016.6	6168.0	6040.0	6592.0	6429.0
Air Dry-Bulb Temperature	°C	58.6	58.2	59.7	59.6	56.3	56.3	57.0	56.9
Refrigerant Temperature	°C	46.7	48.1	50.3	53.8	47.2	49.5	48.0	51.1
LMTD	°C	12	15	19	23	14	18	16	21
UA	W/K	635.57	482.84	439.36	350.35	451.67	327.93	424.35	313.48
NTU	-	1.04	0.79	0.72	0.57	0.74	0.53	0.69	0.51
Effectiveness	-	0.1915	0.1896	0.2098	0.2043	0.1572	0.1539	0.1680	0.1638
Refrigerant Pressure Drop	kPa	78.2	1.4	85.0	1.7	79.3	1.4	87.2	1.7
Airside DP	Pa	75.1	75.5	75.1	75.5	75.1	75.5	75.1	75.5
Air Heat Transfer Coefficient (Average)	$W/m^2.K$	130.0	148.3	130.0	148.3	130.0	148.3	130.0	148.3
Refrigerant Heat Transfer Coefficient (Average)	$W/m^2.K$	3341.0	1721.0	4113.0	2033.0	3040.0	1382.0	3423.0	1601.0
Subcooling	°C	13.20	13.14	8.96	7.35	6.77	5.93	5.34	4.05
Charge	kg	0.3822	0.1143	0.1079	0.0352	0.3097	0.094	0.2522	0.0764

Table 16: Unit 1 – Compressor Performance Summary.

Compressor		Baseline				
Refrigerant	-	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Isentropic efficiency	-	0.66	0.70	0.69	0.70	0.68
Power	kW	1.9175	1.7682	2.0449	1.7966	1.8932
Pressure Lift	kPa	2284.8	1556.0	2087.7	1902.2	1904.9
Effective Displacement Volume	cm^3	19.80	25.87	24.80	19.64	25.35
Rotation Speed	RPM	3600	3600	3600	3600	3600

Table 17: Unit 1 – Expected Modified System Performances.

System		*Baseline*				
Case	-	**Simulation**	**Expected**			
Refrigerant	**-**	**R444B**	**R290**	**R454C**	**R444B**	**R457A**
Condenser	-	BTFD5	BTFD5	BTFD5	BTFD5	NMCD2
Compressor	-	SL260DG-C8EU	-	-	-	-
Cooling Capacity	BTU/hr	17403	17639	18104	18140	17749
Compressor Power	kW	1.92	1.77	2.04	1.80	1.89
Fan Power	kW	0.43	0.43	0.43	0.43	0.43
Total Power	kW	2.35	2.20	2.48	2.23	2.33
COP	-	2.17	2.35	2.14	2.38	2.24
COP Gain	-	1.00	1.08	0.99	1.10	1.03

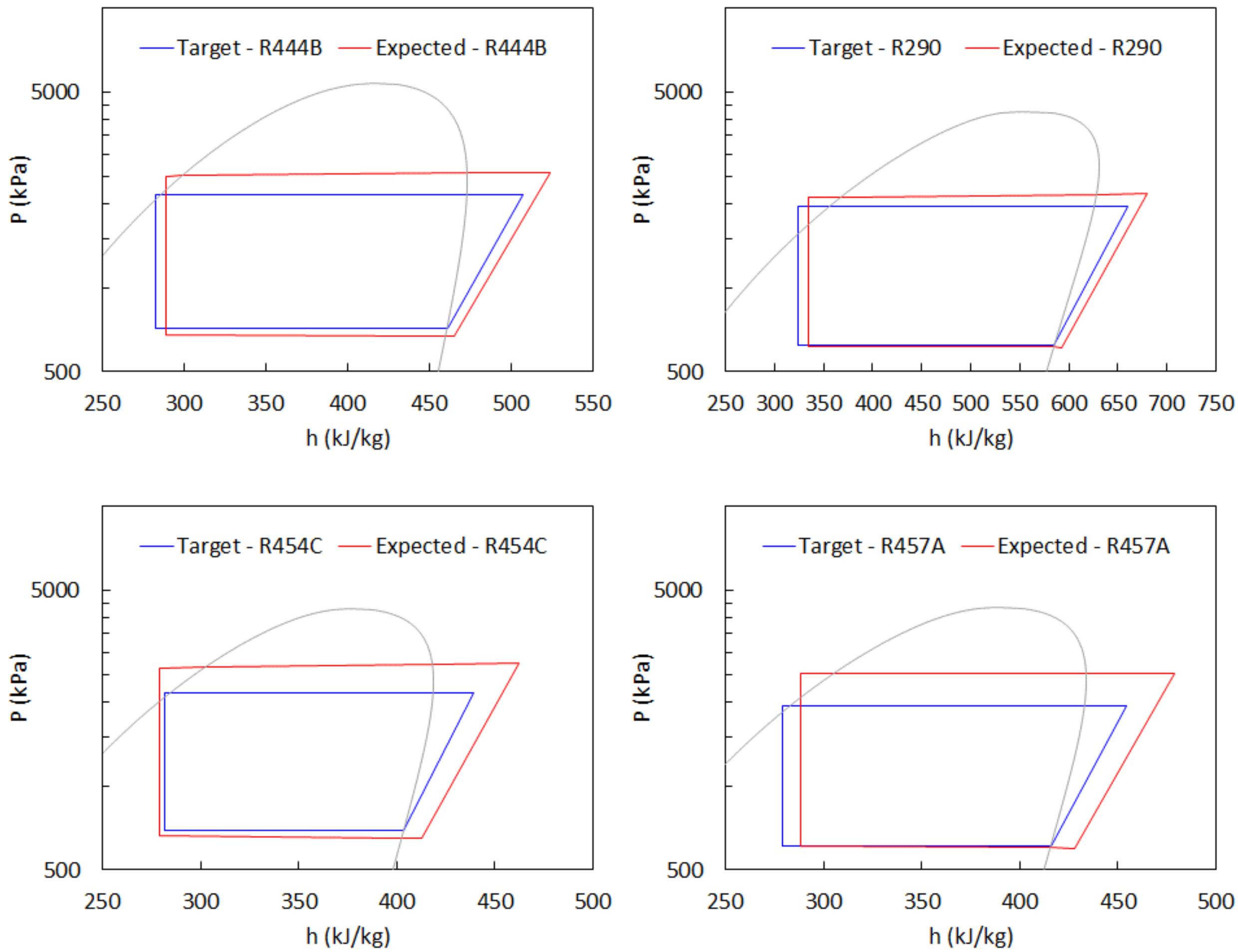

Figure 10. Unit 1 – Modified Systems P-h Diagrams.

Table 18: Unit 4 – Theoretical Cycle Re-Design Summary.

System		***Baseline***	***Alternative 1***	***Alternative 2***
			Target	
Refrigerant	-	**R290**	**R290**	**R290**
Condenser	-	BTFD9	-	-
Compressor	-	PSH356DG-C8DU4	-	-
Cooling Capacity	BTU/hr	17940	17940	23920
Compressor Power	kW	2.11	1.40	3.23
Fan Power	kW	0.28	0.28	0.28
Total Power	kW	2.39	1.68	3.51
COP	-	2.20	3.14	2.00
COP Gain	-	1.00	1.42	0.91

Table 19: Unit 4 – HX Analysis Summary.

Condenser		**R290 - 18kBTU**		**R290 - 24kBTU**	
Inputs		***BTFD9***	***NTFD5***	***BTFD9***	***NTFD5***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m^3/s	0.81	0.76	0.81	0.76
Refrigerant Pressure	kPa	2875	2875	2875	2875
Saturation Temperature at Inlet	°C	75.5	75.5	75.5	75.5

Condenser		R290 - 18kBTU		R290 - 24kBTU	
Inputs		***BTFD9***	***NTFD5***	***BTFD9***	***NTFD5***
Refrigerant Temperature	°C	110	110	110	110
Mass Flow Rate	kg/s	0.02	0.02	0.03	0.03
Outputs					
Heat Load	W	8139	8148	12080	12190
Air Dry-Bulb Temperature	°C	55.0	56.1	59.5	61.2
Refrigerant Temperature	°C	46.2	46.0	47.7	46.4
LMTD	°C	9.6	7.4	14.3	10.0
UA	W/K	848	1097	846	1216
NTU	-	0.97	1.34	0.97	1.48
Effectiveness	-	0.15	0.16	0.22	0.23
Refrigerant Pressure Drop	kPa	4.2	13.4	11.0	35.2
Airside DP	Pa	16.0	15.9	16.0	15.9
Air Heat Transfer Coefficient (Average)	W/m².K	82.9	100.7	82.9	100.7
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	1535.2	1493.7	2382.4	2505.6
Subcooling	°C	29.2	29.2	27.6	28.4
Charge in Tubes	kg	0.90	0.46	0.76	0.39

Table 20: Unit 4 – Compressor Performance Summary.

Compressor		**Baseline**	**18kBTU/Hr**		**24kBTU/Hr**	
Refrigerant	**-**	**R290**	**R290**	**R290**	**R290**	**R290**
Isentropic efficiency	-	0.61	0.70	0.70	0.70	0.70
Power	kW	2.1067	1.7364	1.7093	3.3152	3.31
Pressure Lift	kPa	1457.6	1556.3	1513.7	2947.1	2937.4
Effective Displacement Volume	cm³	26.394	26.309	26.309	37.866	37.866
Rotation Speed	RPM	3600	3600	3600	3600	3600

Table 21: Unit 4 – Expected Modified System Performances.

System		***Baseline***	***Alternative 1***		***Alternative 2***	
			Expected			
Refrigerant	**-**	**R290**	**R290**	**R290**	**R290**	**R290**
Condenser	-	BTFD9	BTFD9	NTFD5	BTFD9	NTFD5
Compressor	-	PSH356DG-C8DU4	-	-	-	-
Cooling Capacity	BTU/hr	17940	17991	18147	24045	24120
Compressor Power	kW	2.11	1.74	1.71	3.32	3.31
Fan Power	kW	0.28	0.28	0.28	0.28	0.28
Total Power	kW	2.39	2.02	1.99	3.60	3.59
COP	-	2.20	2.61	2.67	1.96	1.97
COP Gain	-	1.00	1.19	1.21	0.89	0.89

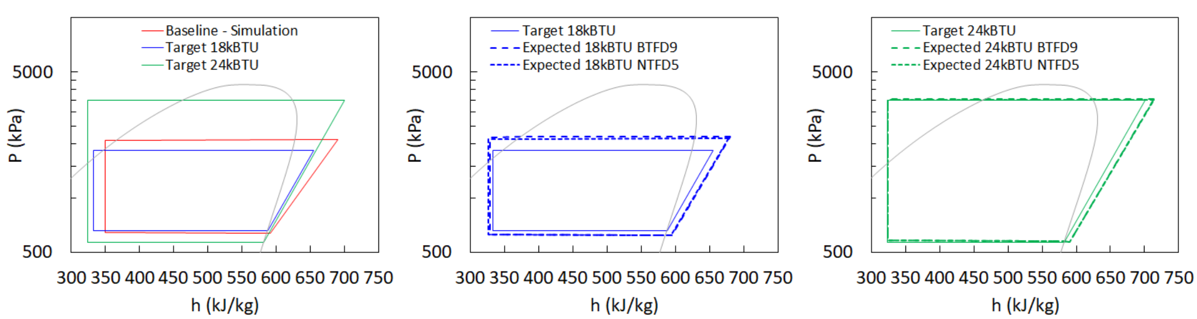

Figure 11. Unit 4 – Modified Systems P-h Diagrams.

Table 22: Unit 6 – Theoretical Cycle Re-Design Summary.

System			*Alternate 1*	*Alternate 2*	*Alternate 3*
		Simulation	**Target**		
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R452B**
Condenser	-	BTFD9	-	-	-
Compressor	-	GMCC KSG226N1UMT	ZP20K5E	ZP21K5E	-
Cooling Capacity	BTU/hr	23115	23114	23114	23115
Compressor Power	kW	2.73	2.37	2.29	2.04
Fan Power	kW	8.46	9.75	10.10	11.31
Total Power	kW	2.73	2.37	2.29	2.04
COP	-	2.48	2.86	2.96	3.32
COP Gain	-	1.00	1.15	1.19	1.34

Table 23: Unit 6 – HX Analysis for R32

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3562	3562	3562	3562
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	9159	9416	9332	9113
Air Dry-Bulb Temperature	°C	53.63	55.35	54.27	55.24
Refrigerant Temperature	°C	49.78	46.15	47.40	50.47
LMTD	°C	19.94	9.46	15.13	20.57
UA	W/K	459.40	995.12	616.75	443.09
NTU	-	0.39	0.97	0.52	0.43
Refrigerant Pressure Drop	kPa	100.98	26.10	3.06	4.70
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	5543.00	2624.00	2353.00	2978.00
Subcooling	°C	4.48	9.04	8.10	5.07
Charge	kg	0.39	0.71	0.17	0.11

Table 24: Unit 6 – HX Analysis for R452B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3247	3247	3247	3247
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7876	7964	7936	7866
Air Dry-Bulb Temperature	°C	52.52	53.94	53.06	53.99
Refrigerant Temperature	°C	47.41	46.05	46.53	47.61
LMTD	°C	15.49	8.09	12.37	15.72
UA	W/K	508.37	984.95	641.46	500.33
NTU	-	0.43	0.96	0.55	0.49
Refrigerant Pressure Drop	kPa	71.90	21.03	2.60	3.70
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	4252.00	2077.00	2103.00	2112.00
Subcooling	°C	6.14	8.20	7.99	6.89
Charge	kg	0.55	0.90	0.21	0.15

Table 25: Unit 6 – HX Analysis for R447B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3025	3025	3025	3025
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7607	8241	8157	7914
Air Dry-Bulb Temperature	°C	52.41	54.19	53.25	54.04
Refrigerant Temperature	°C	50.00	46.24	47.63	51.40
LMTD	°C	20.58	10.45	15.92	22.14
UA	W/K	369.65	788.34	512.32	357.47
NTU	-	0.31	0.77	0.44	0.35
Refrigerant Pressure Drop	kPa	185.90	27.30	3.18	4.90
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	5396.00	2439.00	2397.00	3281.00
Subcooling	°C	0.00	6.05	5.17	1.22
Charge	kg	0.33	0.70	0.16	0.11

Table 26: Unit 6 – HX Analysis for R454B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46.01	46.01	46.01	46.01
Relative Humidity	%	16.37	16.37	16.37	16.37
Air Flowrate	m³/s	1.08	0.94	1.08	0.94
Refrigerant Pressure	kPa	3204	3204	3204	3204
Saturation Temperature at Inlet	°C	55.53	55.53	55.53	55.53
Refrigerant Temperature	°C	112.00	112.00	112.00	112.00
Mass Flow Rate	kg/s	0.03	0.03	0.03	0.03
Outputs					
Heat Load	W	7993	8094	8060	7976
Air Dry-Bulb Temperature	°C	52.61	54.06	53.16	54.10
Refrigerant Temperature	°C	47.59	46.06	46.61	47.91
LMTD	°C	15.95	8.28	12.72	16.40
UA	W/K	501.09	977.17	633.67	486.37
NTU	-	0.43	0.96	0.54	0.48
Refrigerant Pressure Drop	kPa	74.70	22.02	2.70	4.10
Airside DP	Pa	26.30	29.30	27.70	28.90
Air Heat Transfer Coefficient (Average)	W/m².K	109.57	126.69	128.70	130.84
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	4445.93	2140.00	2008.00	2201.00
Subcooling	°C	5.75	8.03	7.75	6.43
Charge	kg	0.51	0.87	0.20	0.14

Table 27: Unit 6 – Compressor Performance Summary.

		Baseline	***Alternate 1***	***Alternate 2***	***Alternate 3***
Refrigerant		***R32***	***R32***	***R454B***	***R452B***
Isentropic Efficiency	-	0.60	0.64	0.66	0.70
Volumetric Efficiency	-	-	0.87	0.90	-
Displacement Volume	cm³	-	19.34	20.31	-
Frequency	Hz	60	60	60	60
Effective Displacement	cm³	16.0	16.8	18.3	19.0
Compressor Power	kW	2.4	2.3	2.3	2.1

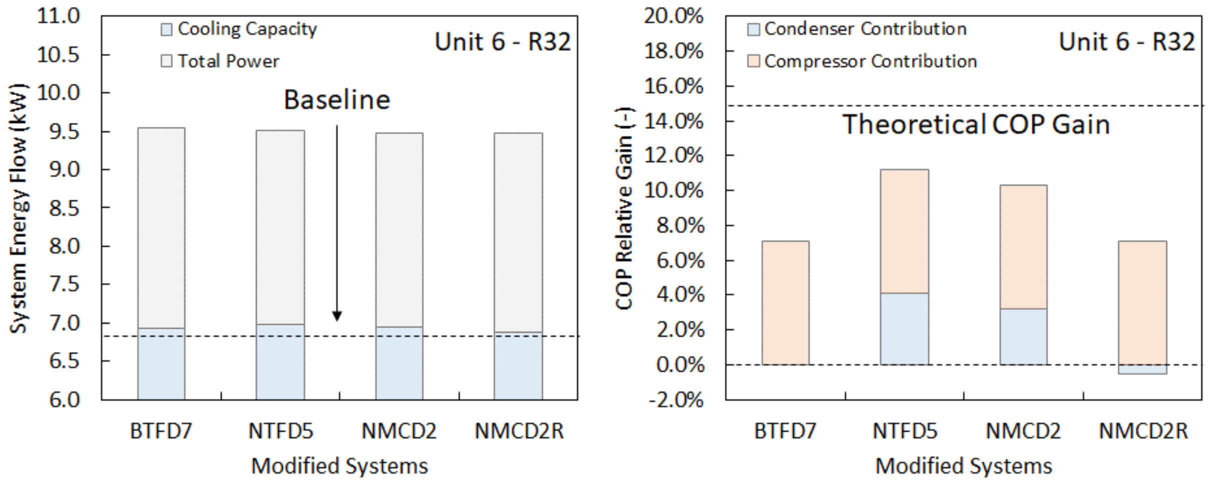

Figure 12. Unit 6 – System Level Analysis: Performance Results for R32.

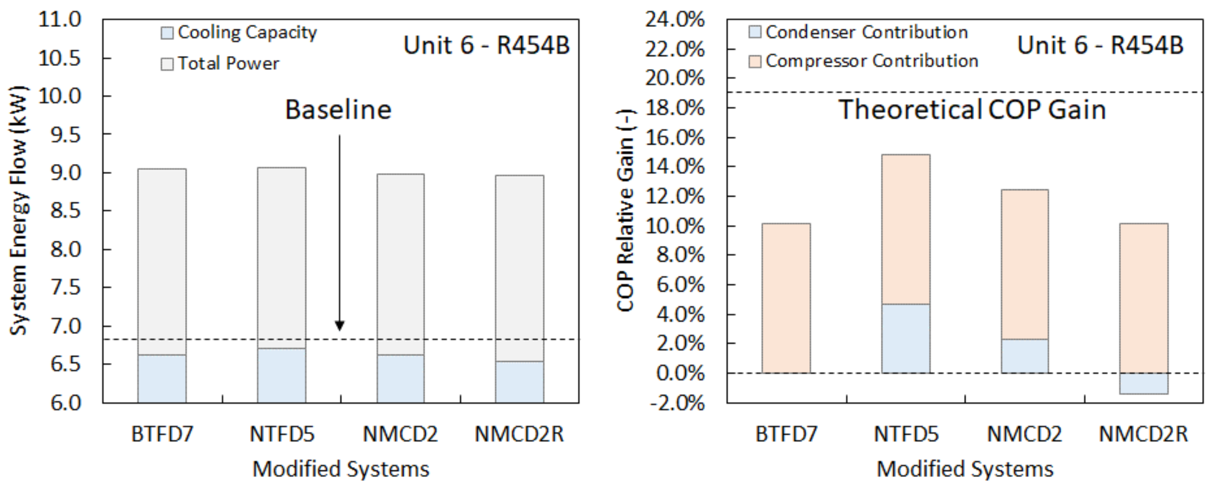

Figure 13. Unit 6 – System Level Analysis: Performance Results for R454B.

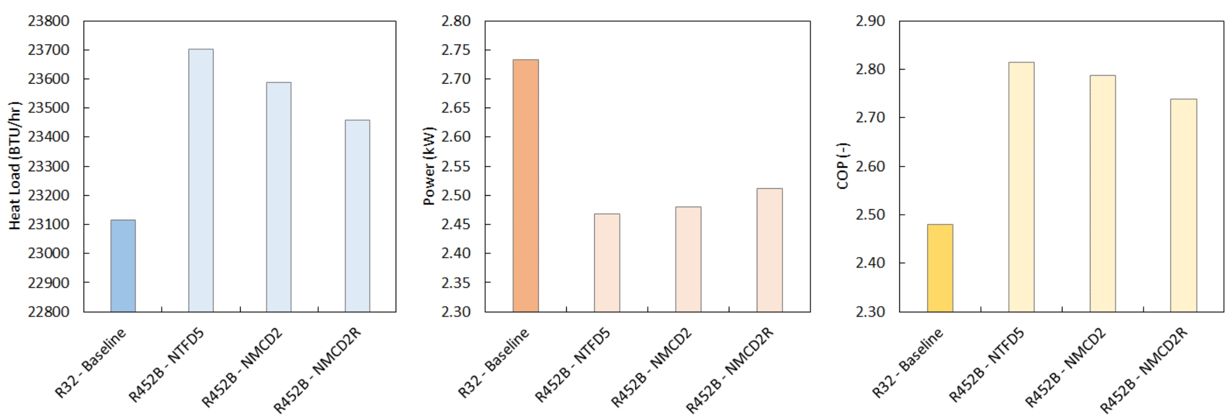

Figure 14. Unit 6 - Comparative System Performance Summary for R452B.

Table 28: Unit 10 – Theoretical Cycle Re-Design Summary.

System		*Baseline*	*Alternate 1*	*Alternate 2*	*Alternate 3*
		Simulation		**Target**	
Refrigerant	**-**	**R32**	**R452B**	**R447B**	**R454B**
Condenser	-	BTFD9	-	-	-
Compressor	-	ZP42K5E	ZP31K5E	ZP34K5E	ZP31K5E
Cooling Capacity	BTU/hr	29005	34311	31611	34608
Compressor Power	kW	3.84	2.81	2.31	2.65
Fan Power	kW	0.70	0.70	0.70	0.70
Total Power	kW	4.54	3.51	3.01	3.35
COP	-	1.87	2.87	3.08	3.03
COP Gain	-	1.00	1.53	1.64	1.62

Table 29: Unit 10 – HX Analysis for R32

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3562	3562	3562	3562
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	10693	11074	11435	10669
Air Dry-Bulb Temperature	°C	54.1	57.0	54.9	55.8
Refrigerant Temperature	°C	55.2	52.9	49.3	55.4
LMTD	°C	22.8	19.8	15.9	22.5
UA	W/K	468	560	717	475
NTU	-	0.35	0.55	0.54	0.42
Refrigerant Pressure Drop	kPa	26.7	67.1	6.8	10.1
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	3823	4239	3050	3991
Subcooling	°C	0.00	1.75	6.17	0.00
Charge	kg	0.61	0.43	0.17	0.11

Table 30: Unit 10 – HX Analysis for R452B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3247	3247	3247	3247
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9549	9812	9751	9500
Air Dry-Bulb Temperature	°C	53.2	55.8	53.6	54.8
Refrigerant Temperature	°C	49.5	46.4	47.1	50.1
LMTD	°C	16.7	9.2	12.2	17.1
UA	W/K	573	1067	802	557
NTU	-	0.43	1.04	0.60	0.49
Refrigerant Pressure Drop	kPa	17.2	47.1	5.6	8.2
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	2974	3038	2537	2812
Subcooling	°C	4.82	7.51	7.34	4.38
Charge	kg	0.83	0.79	0.23	0.15

Table 31: Unit 10 – HX Analysis for R447B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3025	3025	3025	3025
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9016	9632	9923	9085
Air Dry-Bulb Temperature	°C	52.9	55.6	53.8	54.4
Refrigerant Temperature	°C	52.4	51.7	49.9	52.7
LMTD	°C	20.4	18.9	17.1	20.3
UA	W/K	441	510	579	448
NTU	-	0.33	0.50	0.43	0.40
Refrigerant Pressure Drop	kPa	29.2	67.3	7.2	10.8
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	3528	3833	2999	3458
Subcooling	°C	0.00	0.00	2.67	0.00
Charge	kg	0.56	0.45	0.17	0.10

Table 32: Unit 10 – HX Analysis for R454B

Condenser					
Inputs		***BTFD7***	***NTFD5***	***NMCD2***	***NMCD2R***
Air Dry-Bulb Temperature	°C	46	46	46	46
Relative Humidity	%	16.4	16.4	16.4	16.4
Air Flowrate	m³/s	1.23	0.94	1.23	1.04
Refrigerant Pressure	kPa	3204	3204	3204	3204
Saturation Temperature at Inlet	°C	56	56	56	56
Refrigerant Temperature	°C	100	100	100	100
Mass Flow Rate	kg/s	0.04	0.04	0.04	0.04
Outputs					
Heat Load	W	9634	9953	9901	9597
Air Dry-Bulb Temperature	°C	53.3	55.9	53.8	54.9
Refrigerant Temperature	°C	50.4	46.7	47.3	50.8
LMTD	°C	17.9	10.5	12.7	18.0
UA	W/K	537	952	782	532
NTU	-	0.40	0.93	0.59	0.47
Refrigerant Pressure Drop	kPa	18.8	51.1	5.9	8.7
Airside DP	Pa	29.6	26.7	25.7	26.0
Air Heat Transfer Coefficient (Average)	W/m².K	100.4	117.0	124.8	125.3
Refrigerant Heat Transfer Coefficient (Average)	W/m².K	3095	3211	2633	2942
Subcooling	°C	3.71	6.98	6.98	3.40
Charge	kg	0.78	0.71	0.22	0.14

Table 33. Unit 10 - Compressor Performance Summary.

Compressor			Copeland ZP31K5E-PFV	Copeland ZP34K5E-PFV	Copeland ZP31K5E-PFV
Refrigerant		***R32***	***R452B***	***R447B***	***R454B***
Isentropic Efficiency	-	0.439	0.638	0.662	0.662
Volumetric Efficiency	-		0.760	0.803	0.790
Displacement Volume	cm³		29.350	29.350	29.350
Frequency	Hz	50	50	50	50
Effective Displacement Volume	cm³	19.646	22.301	23.581	23.183

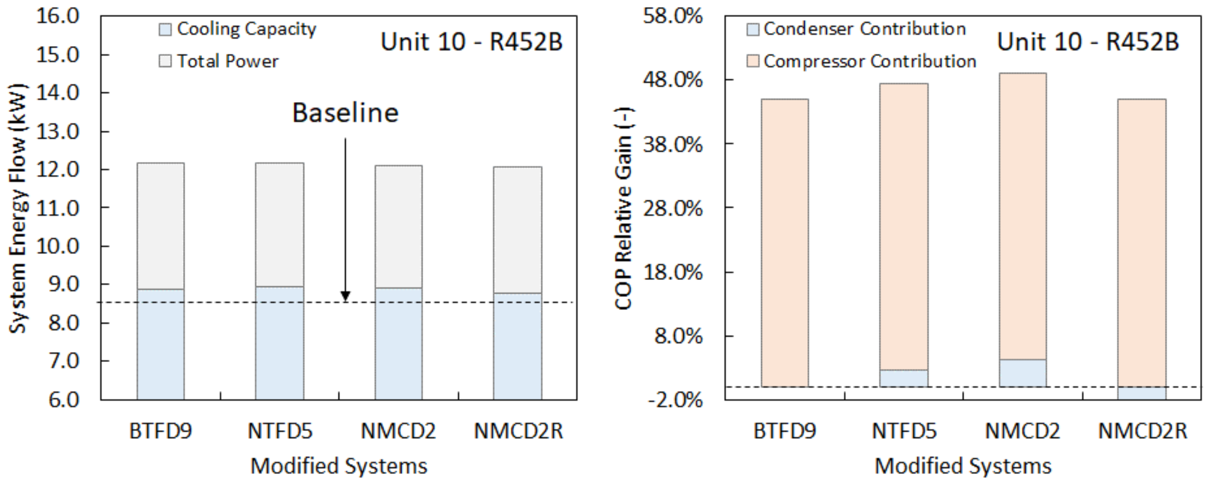

Figure 15. Unit 10 – System Level Analysis: Performance Results for R452B.

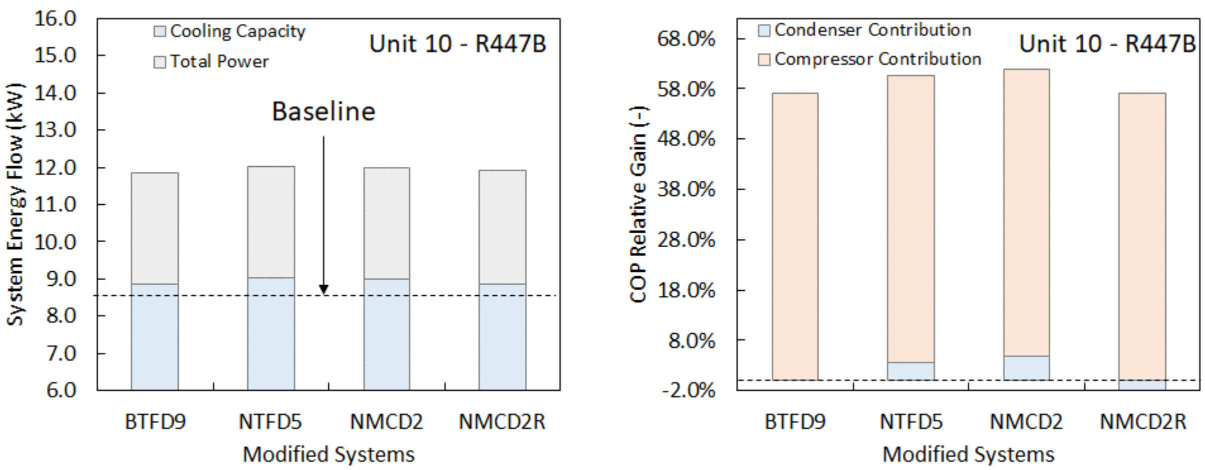

Figure 16. Unit 10 – System Level Analysis: Performance Results for R447B.

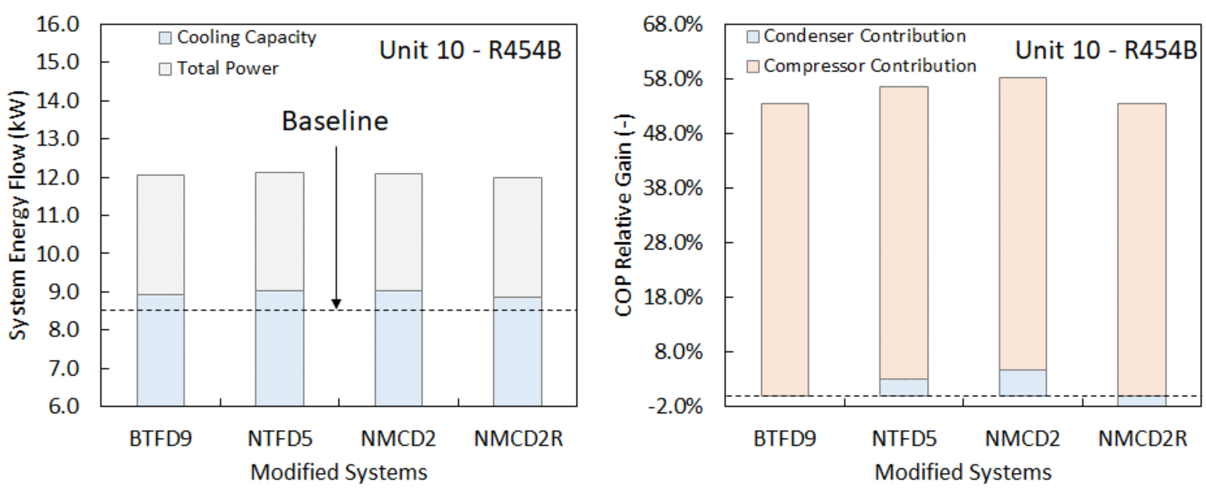

Figure 17. Unit 10 – System Level Analysis: Performance Results for R454B.

APPENDIX B – Unit 6 Initial Tests, Scope Change and Test Setup

Unit 6 was initially modified and tested at a separate facility and the test results exhibited a considerably lower cooling capacity than expected (~20%). Power consumption was also greater than designed. The condensing pressures were 20-30% above expectations, and the refrigerant pressure drop across the condenser was at least twice as high as expected. The outlet conditions of the condenser for R32 were possibly in two-phase. The condenser airflow rate was 10%-15% lower than expected. Superheat hardly met the setpoint values.

OTS formulated a hypothesis that the degraded performance was due to the condenser not being fully active; i.e. some regions were not transferring heat. One way for this to happen is by having severe maldistribution thus impeding heat transfer, increasing pressure drop – thus the condensing pressure – and possibly reducing the flow rate as well; all of which were observed in the test data. OTS tested the hypothesis by running hot water through the HX and observing with a thermal camera (Figure 18), which revealed the "dead zones". Upon inspection by the manufacturer, it was confirmed there were blockages in some of the tubes. A new HX was built, but the same pattern was observed, forcing OTS to remove the condenser replacement from the scope given the project schedule.

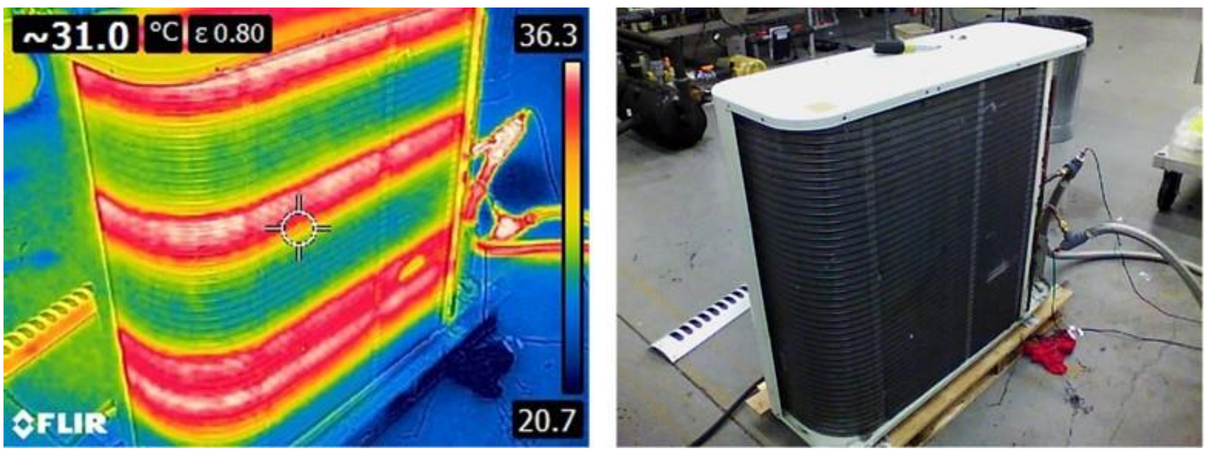

Figure 18. Hot Water Thermal Imaging.

Given the challenges with the initial tests and unit modification, the scope was re-defined. The original test plan was changed to accommodate time and resources as appropriate. Table 34 outlines the major changes to the scope. The tests were conducted at the OTS laboratory (Figure 19 to Figure 22). A summary of the key differences between the test setups (original and at OTS) is presented in Table 35.

Table 34: Test Scope Change.

Unit	Refrigerant	Test	Original Scope		New Scope	
			Planned	Actual	Planned	Actual
Unit 1	R290	Charge Optimization	Yes	No	No	No
		Performance Tests	Yes	No	No	No
Unit 6	R32 (Baseline)	Charge Optimization	No	No	Yes	Yes
		Performance Tests	No	No	Yes	Yes
	R32 (Modified)	Charge Optimization	Yes	Yes	Yes	Yes
		Performance Tests	Yes	Yes	Yes	Yes
	R454B	Charge Optimization	Yes	Yes	Yes	Yes
		Performance Tests	Yes	Yes	Yes	Yes
		Leak Tests	Yes	Yes	No	No
Unit 10	R32 (Baseline)	Charge Optimization	No	No	Yes	Yes*
		Performance Tests	No	No	Yes	Yes*
	R447B	Charge Optimization	Yes	No	Yes	Yes
		Performance Tests	Yes	No	Yes	Yes
		Leak Tests	Yes	No	Yes	Yes
	R452B	Charge Optimization	Yes	No	Yes	Yes
		Performance Tests	Yes	No	Yes	Yes
		Leak Tests	Yes	No	No	No

** Tests were conducted; however, no useful data was obtained (see section 5.2)*

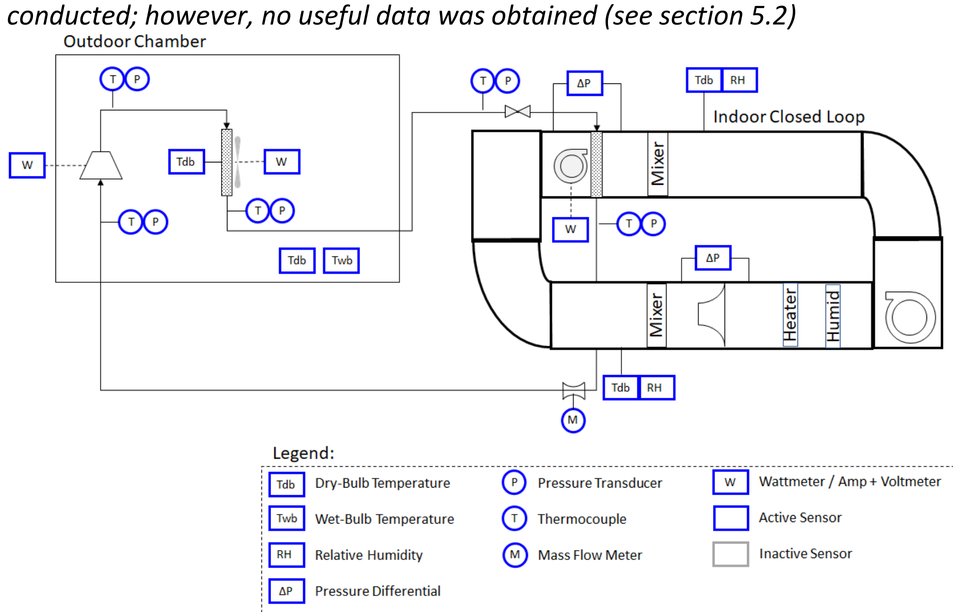

Figure 19. Test Diagram.

Figure 20. OTS Setup: outdoor chamber (left), Unit 10 and frequency converter inside chamber (right).

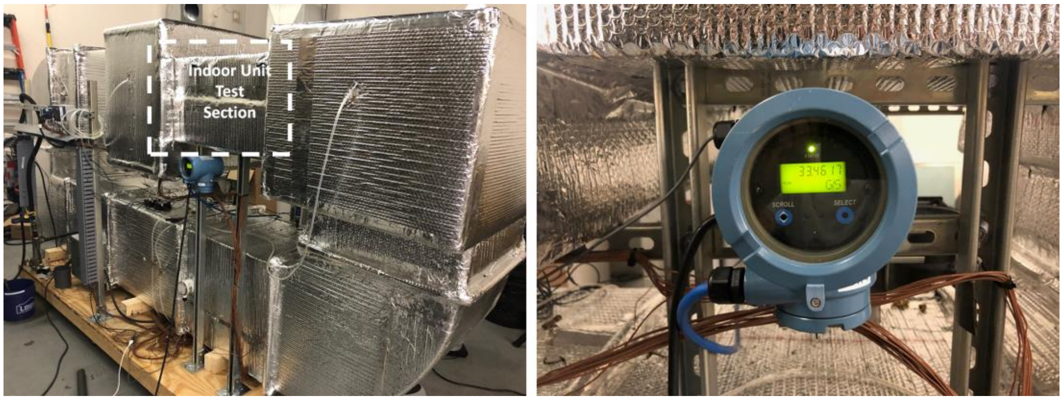

Figure 21. OTS Setup: indoor closed loop left side view (left), refrigerant mass flow meter (right).


Figure 22. OTS Setup: indoor closed loop right side view (left), vapor / liquid lines, sight glasses and TXV (right).

Table 35: List of Measurements.

Component	Refrigerant Side			Air Side		
	Measurement	Original Scope	New Scope	Measurement	Original Scope	New Scope
Condenser	Inlet Temperature	Yes	Yes	Air Flow Rate	Yes	**No**
	Inlet Pressure	Yes	Yes	Air Pressure Drop	No	No
	Outlet Temperature	Yes	Yes	Fan Power	No	**Yes**
	Outlet Pressure	Yes	Yes	Inlet Dry-bulb	Yes	Yes
	Subcooling	Yes*	**Yes**	Inlet Wet-Bulb / RH	Yes	Yes
				Outlet Dry-bulb	Yes	Yes
				Outlet Wet-Bulb / RH	Yes	Yes
Evaporator	Inlet Temperature	No	No	Air Flow Rate	Yes	Yes
	Inlet Pressure	No	No	Air Pressure Drop	No	**Yes****
	Outlet Temperature	Yes	Yes	Blower Power	No	**Yes**
	Outlet Pressure	Yes	Yes	Inlet Dry-bulb	Yes	Yes
	Superheat	Yes*	**Yes**	Inlet Wet-Bulb / RH	Yes	Yes
	Refrigerant Mass Flow Rate	No	**Yes**	Outlet Dry-bulb	Yes	Yes
				Outlet Wet-Bulb / RH	Yes	Yes
Compressor	Suction Temperature	Yes	Yes			
	Suction Pressure	Yes	Yes			
	Discharge Temperature	Yes	Yes			
	Discharge Pressure	Yes	Yes			
	Compressor Power	No	**Yes**			
Expansion Device	Suction Temperature	Yes	Yes			
	Suction Pressure	Yes	Yes			
	Discharge Temperature	No	No			
	Discharge Pressure	No	No			

Charge Optimization

The charge optimization procedure as originally scoped was not implemented due to the following:

a. The systems responded less sensitively to charge on subcooling and superheat, which were difficult to control with charging alone. A manual valve was added (Unit 10 exhibited little expansion) such that superheat could be better controlled. The valve also allowed for better control over the pressure levels compared to charge levels alone.

b. For the modified systems, the charge was gradually increased, departing from the original charge from PRAHA I tests, until it was observed that the superheat and subcooling better matched design conditions for validation purposes.

c. For the refrigerant blends, removing charge could result in fractionation (evaluated as a separate task), so it was decided to only incrementally increase charge, without removing it. For this procedure, a small gradual increment is necessary to avoid overcharging.

APPENDIX C - Unit 6 Raw and Processed Tested Data

Table 36: Unit 6 – Performance Tests

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R32**	**R32**	**R454B**
Charge	lb	3.83	4.27	5.02	3.83	4.27	5.02
Cooling Capacity	BTU/hr	25193	23585	21966	23390	21450	21821
Energy Balance	%	-2.28%	-4.66%	-3.06%	-1.78%	-4.42%	-7.61%
Compressor Power	kW	2.11	1.79	1.77	2.71	2.32	2.25
Fan Power	kW	0.32	0.33	0.33	0.40	0.42	0.42
Total Power	kW	2.43	2.12	2.10	3.10	2.74	2.67
EER	BTU/hr.W	10.36	11.12	10.44	7.54	7.84	8.17
Evaporator							
Airside							
Inlet							
Air Flow Rate	m^3/s	0.31	0.31	0.31	0.31	0.31	0.30

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R32**	**R32**	**R454B**
Temperature	°C	27.0	27.0	27.0	29.0	29.0	29.0
Wet Bulb	°C	19.68	19.68	19.68	21.33	21.33	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.011	0.011	0.011	0.013	0.013	0.013
Density	kg/m³	1.15	1.15	1.15	1.14	1.14	1.14
Enthalpy	kJ/kg	56.3	56.2	56.2	61.9	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0	1.0
Outlet							
Air Flow Rate	m³/s	0.29	0.29	0.29	0.29	0.29	0.29
Temperature	°C	14.3	15.1	15.8	16.9	17.7	18.1
Wet Bulb	°C	14.35	14.35	14.35	14.35	14.35	14.35
Relative Humidity	%	83.6	82.4	80.0	84.5	83.3	81.3
Humidity Ratio	kg/kg	0.008	0.009	0.009	0.010	0.011	0.011
Density	kg/m³	1.21	1.20	1.20	1.19	1.19	1.19
Enthalpy	kJ/kg	35.8	37.5	38.5	42.7	44.7	45.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0	1.0
Refrigerant Side							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	4.58	6.19	4.76	7.49	8.33	8.47
Pressure	kPa	939.13	986.90	876.76	1026.70	1053.10	979.34
Quality	-	0.16	0.19	0.20	0.20	0.25	0.27
Enthalpy	kJ/kg	273.64	269.78	268.60	301.30	291.37	289.89
Entropy	kJ/kg.K	1.20	1.25	1.30	1.27	1.32	1.37
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	8.08	9.26	9.46	9.08	13.54	11.80
Pressure	kPa	939	987	877	1027	1053	979
Superheat	K	3.50	3.07	4.89	1.59	5.20	3.58
Enthalpy	kJ/kg	520.49	520.22	473.43	518.52	523.27	472.93
Entropy	kJ/kg.K	2.15	2.15	2.03	2.13	2.15	2.02
HX Level							
Average Cooling Capacity	kW	7.384	6.912	6.438	6.855	6.287	6.395
Energy Balance (Qair - Qref)/Qref	%	-2.28%	-4.66%	-3.06%	-1.78%	-4.42%	-7.61%
Sensible Heat Ratio	-	0.64	0.66	0.65	0.64	0.67	0.66
Superheat	K	3.500	3.066	4.885	1.593	5.205	3.582
LMTD	K	13.783	12.822	14.015	13.985	12.184	13.041
UA	kW/K	0.573	0.539	0.459	0.550	0.516	0.490
Air Pressure Drop	Pa	N/A	N/A	N/A	N/A	N/A	N/A
Refrigerant Pressure Drop	kPa	N/A	N/A	N/A	N/A	N/A	N/A
Fan Power	kW	0.120	0.127	0.134	0.196	0.217	0.217
Condenser							
Airside							
Inlet							
Air Flow Rate	m³/s	0.9516	0.9838	1.0091	0.9580	0.9735	1.0613
Temperature	°C	35.01	34.76	35.12	46.06	45.93	46.05
Wet Bulb	°C	20.0	19.8	20.0	27.4	27.3	27.4
Humidity Ratio	kg/kg	0.008	0.008	0.009	0.015	0.015	0.015
Density	kg/m³	1.13	1.13	1.13	1.08	1.08	1.08
Enthalpy	kJ/kg	57.0	56.4	57.2	86.2	85.8	86.2
Specific Heat	kJ/kg.K	1.01	1.01	1.01	1.02	1.02	1.02
Outlet							
Air Flow Rate	m³/s	0.98	1.01	1.03	0.98	1.00	1.09
Temperature	°C	43.40	42.29	42.08	54.74	53.60	53.19
Wet Bulb	°C	22.4	22.0	22.1	29.3	29.0	29.0
Humidity Ratio	kg/kg	0.008	0.008	0.009	0.015	0.015	0.015
Density	kg/m³	1.10	1.10	1.10	1.05	1.05	1.05
Enthalpy	kJ/kg	65.6	64.1	64.3	95.2	93.7	93.6
Specific Heat	kJ/kg.K	1.01	1.01	1.01	1.02	1.02	1.02

		Baseline (35°C)	Alternate 1 (35°C)	Alternate 2 (35°C)	Baseline (46°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R32**	**R32**	**R454B**	**R32**	**R32**	**R454B**
Refrigerant Side							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	89.78	82.73	78.33	109.00	107.24	90.75
Pressure	kPa	2724.15	2643.18	2360.90	3464.77	3365.88	3010.13
Superheat	K	45.9	40.1	35.9	54.7	54.2	38.0
Enthalpy	kJ/kg	580.73	573.07	523.39	594.42	593.52	528.90
Entropy	kJ/kg.K	2.20	2.18	2.08	2.21	2.21	2.07
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	39.17	34.52	34.68	51.79	45.63	45.79
Pressure	kPa	2675.81	2598.75	2310.89	3416.39	3324.50	2958.91
Subcooling	K	4.00	7.44	5.59	1.89	6.84	5.07
Enthalpy	kJ/kg	273.6	264.0	266.4	301.3	287.0	287.8
Entropy	kJ/kg.K	1.24	1.21	1.28	1.33	1.28	1.34
HX Level							
Heat Rejection	kW	9.19	8.53	8.08	9.25	8.31	8.42
Subcooling	K	4.00	7.44	5.59	1.89	6.84	5.07
Refrigerant Pressure Drop	kPa	48.34	44.43	50.01	48.38	41.38	51.22
Fan Power	kW	0.20	0.20	0.20	0.20	0.20	0.20
TXV							
Refrigerant							
Inlet		[4]			[4]		
Temperature	°C	30.64	37.31	35.83	39.70	47.55	46.78
Pressure	kPa	1991.01	2587.20	2301.38	2528.52	3317.42	2945.62
Subcooling	°C	*(Two-Phase)	4.47	4.27	*(Two-Phase)	4.83	3.88
Enthalpy	kJ/kg	*(Two-Phase)	269.8	268.6	*(Two-Phase)	291.4	289.9
Entropy	kJ/kg.K	*(Two-Phase)	1.233	1.284	*(Two-Phase)	1.299	1.349
Compressor							
Refrigerant							
Inlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	11.57	12.55	12.76	13.81	17.63	13.07
Pressure	kPa	936.06	984.95	874.98	1024.91	1052.17	969.56
Superheat	K	7.09	6.43	8.26	6.38	9.32	5.18
Enthalpy	kJ/kg	524.9	524.4	477.3	524.6	528.3	474.8
Entropy	kJ/kg.K	2.170	2.161	2.048	2.156	2.166	2.028
Outlet							
Mass Flow Rate	kg/s	0.030	0.028	0.031	0.032	0.027	0.035
Temperature	°C	89.8	82.7	78.3	109.0	107.2	90.8
Pressure	kPa	2724.2	2643.2	2360.9	3464.8	3365.9	3010.1
Superheat	K	45.9	40.1	35.9	54.7	54.2	38.0
Enthalpy	kJ/kg	580.7	573.1	523.4	594.4	593.5	528.9
Entropy	kJ/kg.K	2.200	2.183	2.084	2.205	2.207	2.074
Compressor Level							
Power Consumption	kW	2.11	1.79	1.77	2.71	2.32	2.25
Isentropic Efficiency	-	0.80	0.84	0.73	0.74	0.76	0.69
Frequency	Hz	60	60	60	60	60	60

[4] The baseline configuration does not have an expansion valve, the state point herein presented refers to measurement readings at indoor unit inlet.

APPENDIX D - Unit 10 Baseline Re-Test

Prior to modifying Unit 10, it was tested in its received, baseline condition with the components used to test during PRAHA I. Given the results of the data review in Activity 1, and the challenges experienced in the initial testing of Unit 6, the project team agreed that testing the units in their baseline configuration would be important for more accurate comparison.

The electrical components for Unit 10 have phase mismatch, i.e. the fan and blower are three-phase while the compressor is single-phase, but all operate in 50Hz. OTS does not have a Variable Frequency Drive (VFD) for single-phase motors, requiring the use of a frequency converter to reduce the compressor speed. According to the baseline data from PRAHA 1, the total power consumption of Unit 10 varied between 3.5-4.5kW; OTS has a 5.0kW converter, which should be sufficiently large to meet testing needs.

Initial tests suggested that the compressor peak start current exceeds the converter threshold, causing the latter to trip and shut off. Although the blower and the fan run normally with the converter, the compressor alone does not. The compressor motor was tested at 60Hz direct from the grid and it works, thus confirming that the issue is indeed the peak current. A soft starter was acquired with the objective to mitigate the issue. The soft starter capacitors weren't fast enough to smooth the peak current, however, thus requiring manual charging, which eventually lead to component failure.

The last tentative to run the baseline was connecting the compressor to 60Hz and the fans to 50Hz. The refrigerant mass flow rate was too high impeding full condensation and full evaporation. A manual TXV was added along with two sight glasses in the liquid and vapor lines and reasonable data was obtained for the 35°C ambient temperature condition. While attempting to test the system under the 46°C ambient temperature, the compressor overheats and shuts down. Heavier gauge wire, new contactors and switch bypass were unsuccessfully employed. In the interest of time, the baseline re-tests were discontinued. The analysis will be carried out using the original baseline performance for comparison purposes.

APPENDIX E - Unit 10 Raw and Processed Tested Data

Table 37: Unit 10 – Performance Tests.

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	-	**R447B**	**R452B**	**R447B**	**R452B**
Charge	lb	6.625	6.625	6.625	6.625
Cooling Capacity	BTU/hr	32195	28128	31073	30292
Energy Balance	%	7.52%	-3.29%	4.21%	1.21%
Compressor Power	kW	2.67	2.40	3.16	2.93
Fan Power	kW	0.95	0.98	0.95	0.97
Total Power	kW	3.62	3.38	4.11	3.90
EER	BTU/hr.W	8.88	8.33	7.55	7.76
Evaporator					
Airside					
Inlet					
Air Flow Rate	m³/s	0.74	0.73	0.74	0.73
Temperature	°C	27.0	27.0	29.0	29.0
Wet Bulb	°C	19.68	19.69	21.33	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.011	0.011	0.013	0.013
Density	kg/m³	1.15	1.15	1.14	1.14
Enthalpy	kJ/kg	56.2	56.3	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R447B**	**R452B**	**R447B**	**R452B**
Outlet					
Air Flow Rate	m³/s	0.72	0.71	0.71	0.70
Temperature	°C	17.4	19.1	19.7	19.8
Wet Bulb	°C	15.80	16.64	17.91	18.06
Relative Humidity	%	85.1	78.5	84.7	84.5
Humidity Ratio	kg/kg	0.011	0.011	0.012	0.012
Density	kg/m³	1.19	1.18	1.18	1.18
Enthalpy	kJ/kg	44.3	46.8	50.7	51.1
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0
Refrigerant Side					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	9.81	5.53	12.90	13.09
Pressure	kPa	996.41	907.20	1085.49	1133.86
Quality	-	0.19	0.19	0.27	0.25
Enthalpy	kJ/kg	272.43	264.74	296.09	288.71
Entropy	kJ/kg.K	1.32	1.30	1.40	1.38
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	15.22	25.20	16.76	23.36
Pressure	kPa	996	907	1085	1134
Superheat	K	5.79	19.82	4.42	10.47
Enthalpy	kJ/kg	477.29	485.20	476.43	477.36
Entropy	kJ/kg.K	2.04	2.09	2.03	2.03
HX Level					
Average Cooling Capacity	kW	9.436	8.244	9.107	8.878
Energy Balance (Qair - Qref)/Qref	%	7.52%	-3.29%	4.21%	1.21%
Sensible Heat Ratio	-	0.81	0.85	0.83	0.87
Superheat	K	5.794	19.818	4.422	10.474
LMTD	K	9.534	5.829	9.222	6.171
UA	kW/K	0.990	1.414	0.988	1.439
Air Pressure Drop	Pa	N/A	N/A	N/A	N/A
Refrigerant Pressure Drop	kPa	N/A	N/A	N/A	N/A
Fan Power	kW	0.502	0.523	0.501	0.519
Condenser					
Airside					
Inlet					
Air Flow Rate	m³/s	1.44	1.50	1.44	1.42
Temperature	°C	35.03	35.08	46.14	46.22
Wet Bulb	°C	20.0	20.0	27.4	27.5
Humidity Ratio	kg/kg	0.008	0.009	0.016	0.016
Density	kg/m³	1.13	1.13	1.08	1.07
Enthalpy	kJ/kg	57.0	57.2	86.5	86.7
Specific Heat	kJ/kg.K	1.01	1.01	1.02	1.02
Outlet					
Air Flow Rate	m³/s	1.47	1.53	1.48	1.45
Temperature	°C	41.90	40.83	53.36	53.26
Wet Bulb	°C	22.0	21.7	29.0	29.1
Humidity Ratio	kg/kg	0.008	0.009	0.016	0.016
Density	kg/m³	1.10	1.11	1.05	1.05
Enthalpy	kJ/kg	64.0	63.0	94.0	94.0
Specific Heat	kJ/kg.K	1.01	1.01	1.02	1.02
		0.00010	0.00038	0.00011	-0.00001
Refrigerant Side					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047

		Alternate 1 (35°C)	Alternate 2 (35°C)	Alternate 1 (46°C)	Alternate 2 (46°C)
Refrigerant	**-**	**R447B**	**R452B**	**R447B**	**R452B**
Temperature	°C	78.84	92.46	93.29	97.45
Pressure	kPa	2493.84	2600.61	3199.13	3357.43
Superheat	K	31.5	46.5	35.3	40.4
Enthalpy	kJ/kg	522.20	532.28	529.64	527.68
Entropy	kJ/kg.K	2.09	2.11	2.08	2.07
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	40.68	35.54	53.44	48.65
Pressure	kPa	2481.63	2599.27	3187.26	3351.92
Subcooling	K	3.37	9.26	1.62	7.33
Enthalpy	kJ/kg	274.8	266.6	300.2	291.9
Entropy	kJ/kg.K	1.32	1.29	1.39	1.37
HX Level					
Heat Rejection	kW	11.39	9.94	11.59	11.10
Energy Balance (Qair - Qref)	kW	N/A	N/A	N/A	N/A
Subcooling	K	3.37	9.26	1.62	7.33
Air Pressure Drop	Pa	-	-	-	-
Refrigerant Pressure Drop	kPa	12.21	1.34	11.87	5.51
Fan Power	kW	0.45	0.45	0.45	0.45
TXV					
Refrigerant					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	39.42	34.55	51.55	47.11
Pressure	kPa	2462.98	2583.59	3166.49	3331.97
Subcooling	°C	4.31	9.99	3.21	8.59
Enthalpy	kJ/kg	272.4	264.7	296.1	288.7
Entropy	kJ/kg.K	1.310	1.284	1.382	1.358
Compressor					
Refrigerant					
Inlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	16.84	26.01	17.17	24.96
Pressure	kPa	993.13	902.34	1082.17	1128.72
Superheat	K	7.52	20.81	4.94	12.23
Enthalpy	kJ/kg	479.3	486.2	477.0	479.4
Entropy	kJ/kg.K	2.052	2.090	2.035	2.042
Outlet					
Mass Flow Rate	kg/s	0.046	0.037	0.051	0.047
Temperature	°C	78.8	92.5	93.3	97.5
Pressure	kPa	2493.8	2600.6	3199.1	3357.4
Superheat	K	31.5	46.5	35.3	40.4
Enthalpy	kJ/kg	522.2	532.3	529.6	527.7
Entropy	kJ/kg.K	2.087	2.112	2.082	2.073
Compressor Level					
Power Consumption	kW	2.67	2.40	3.16	2.93
Isentropic Efficiency	-	0.72	0.83	0.68	0.77
Frequency	Hz	60	60	60	60

Table 38: Unit 10 – R447B Leak Tests

System			***Liquid Line Leak***		***Vapor Line Leak***	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	**-**	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Charge	lb	6.625	4.27	6.625	4.23	6.77

System			Liquid Line Leak		Vapor Line Leak	
		Full Charge	Low Charge	Re-Charged	Low Charge	Re-Charged
Refrigerant	**-**	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Cooling Capacity	BTU/hr	31073	14216	30865	15171	30587
Energy Balance	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Compressor Power	kW	3.18	2.93	3.18	2.94	-
Fan Power	kW	0.95	0.98	0.98	0.98	0.98
Total Power	kW	4.13	3.90	4.16	3.92	-
EER	BTU/hr.W	7.52	3.64	7.42	3.87	-
Evaporator						
Airside						
Inlet						
Air Flow Rate	m³/s	0.74	0.73	0.74	0.73	0.74
Temperature	°C	29.0	29.0	29.0	29.0	29.0
Wet Bulb	°C	21.33	21.34	21.34	21.34	21.34
Relative Humidity	%	51.0	51.0	51.0	51.0	51.0
Humidity Ratio	kg/kg	0.013	0.013	0.013	0.013	0.013
Density	kg/m³	1.14	1.14	1.14	1.14	1.14
Enthalpy	kJ/kg	62.0	62.0	62.0	62.0	62.0
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0
Outlet						
Air Flow Rate	m³/s	0.71	0.72	0.71	0.72	0.71
Temperature	°C	19.7	23.3	19.6	23.2	19.7
Wet Bulb	°C	17.91	19.87	18.08	19.77	18.05
Relative Humidity	%	84.7	73.1	86.3	73.6	86.0
Humidity Ratio	kg/kg	0.012	0.013	0.012	0.013	0.012
Density	kg/m³	1.18	1.16	1.18	1.16	1.18
Enthalpy	kJ/kg	50.7	57.0	51.2	56.7	51.1
Specific Heat	kJ/kg.K	1.0	1.0	1.0	1.0	1.0
Refrigerant Side						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	12.90	2.61	12.94	2.81	12.75
Pressure	kPa	1085.49	794.22	1086.62	799.23	1080.50
Quality	-	0.27	0.30	0.27	0.30	0.27
Enthalpy	kJ/kg	296.09	291.52	296.48	290.79	296.24
Entropy	kJ/kg.K	1.40	1.40	1.41	1.40	1.41
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	16.76	28.23	17.07	27.95	17.01
Pressure	kPa	1085	794	1087	799	1080
Superheat	K	4.42	26.24	4.70	25.76	4.82
Enthalpy	kJ/kg	476.43	496.65	476.77	496.25	476.88
Entropy	kJ/kg.K	2.03	2.14	2.03	2.13	2.03
HX Level						
Average Cooling Capacity	kW	9.107	4.167	9.046	4.446	8.965
Energy Balance (Qair – Qref)/Qref	%	4.21%	-34.72%	0.35%	-31.55%	1.87%
Sensible Heat Ratio	-	0.83	1.18	0.90	1.12	0.89
Superheat	K	4.422	26.235	4.695	25.756	4.823
LMTD	K	9.222	6.051	9.065	6.501	9.217
UA	kW/K	0.988	0.689	0.998	0.684	0.973
Fan Power	kW	0.501	0.524	0.524	0.524	0.524
Condenser						
Airside						
Inlet						
Air Flow Rate	m³/s	1.44	1.49	1.42	1.48	1.42
Temperature	°C	46.14	46.08	46.21	45.77	46.02
Wet Bulb	°C	27.4	27.4	27.5	27.2	27.4
Humidity Ratio	kg/kg	0.016	0.015	0.016	0.015	0.015
Density	kg/m³	1.08	1.08	1.07	1.08	1.08
Enthalpy	kJ/kg	86.5	86.3	86.7	85.3	86.1
Specific Heat	kJ/kg.K	1.02	1.02	1.02	1.02	1.02

System			***Liquid Line Leak***		***Vapor Line Leak***	
		Full Charge	**Low Charge**	**Re-Charged**	**Low Charge**	**Re-Charged**
Refrigerant	**-**	**R447B**	**R447B**	**R447B**	**R447B**	**R447B**
Outlet						
Air Flow Rate	m³/s	1.48	1.52	1.46	1.50	1.46
Temperature	°C	53.36	51.27	53.52	51.05	53.28
Wet Bulb	°C	29.0	28.6	29.1	28.4	29.0
Humidity Ratio	kg/kg	0.016	0.015	0.016	0.015	0.015
Density	kg/m³	1.05	1.06	1.05	1.06	1.05
Enthalpy	kJ/kg	94.0	91.7	94.3	90.8	93.6
Specific Heat	kJ/kg.K	1.02	1.02	1.02	1.02	1.02
Refrigerant Side						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	93.29	121.77	94.07	120.31	94.34
Pressure	kPa	3199.13	2846.79	3200.02	2847.47	3175.47
Superheat	K	35.3	68.9	36.1	67.4	36.7
Enthalpy	kJ/kg	529.64	569.70	530.67	567.95	531.39
Entropy	kJ/kg.K	2.08	2.20	2.08	2.20	2.09
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	53.44	50.27	53.37	50.13	53.28
Pressure	kPa	3187.26	2843.00	3188.61	2843.11	3164.31
Subcooling	K	1.62	-0.33	1.71	-0.19	1.45
Enthalpy	kJ/kg	300.2	293.2	300.0	293.2	299.9
Entropy	kJ/kg.K	1.39	1.37	1.39	1.37	1.39
HX Level						
Heat Rejection	kW	11.59	8.60	11.57	8.69	11.49
Energy Balance (Qair – Qref)	kW	N/A	N/A	N/A	N/A	N/A
Subcooling	K	1.62	-0.33	1.71	-0.19	1.45
Refrigerant Pressure Drop	kPa	11.87	3.79	11.40	4.36	11.16
Fan Power	kW	0.45	0.45	0.45	0.45	0.45
TXV						
Refrigerant						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	51.55	49.15	51.74	48.80	51.60
Pressure	kPa	3166.49	2827.45	3168.66	2827.31	3144.31
Subcooling	°C	3.21	0.54	3.06	0.89	2.84
Enthalpy	kJ/kg	296.1	291.5	296.5	290.8	296.2
Entropy	kJ/kg.K	1.382	1.369	1.383	1.366	1.382
Compressor						
Refrigerant						
Inlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	17.17	29.26	18.00	28.98	18.47
Pressure	kPa	1082.17	793.15	1082.65	797.99	1076.58
Superheat	K	4.94	27.30	5.75	26.83	6.41
Enthalpy	kJ/kg	477.0	497.7	478.0	497.3	478.8
Entropy	kJ/kg.K	2.035	2.140	2.038	2.138	2.041
Outlet						
Mass Flow Rate	kg/s	0.051	0.031	0.050	0.032	0.050
Temperature	°C	93.3	121.8	94.1	120.3	94.3
Pressure	kPa	3199.1	2846.8	3200.0	2847.5	3175.5
Superheat	K	35.3	68.9	36.1	67.4	36.7
Enthalpy	kJ/kg	529.6	569.7	530.7	568.0	531.4
Entropy	kJ/kg.K	2.082	2.200	2.085	2.195	2.087
Compressor Level						
Power Consumption	kW	3.18	2.93	3.18	2.94	0.00
Isentropic Efficiency	-	0.68	0.68	0.68	0.69	0.68
Frequency	Hz	60	60	60	60	60

System			Liquid Line Leak		Vapor Line Leak	
		Full Charge	Low Charge	Re-Charged	Low Charge	Re-Charged
Refrigerant	-	R447B	R447B	R447B	R447B	R447B

APPENDIX F - Model Verification and Validation

Table 39: Unit 6 – Model Verification and Validation for Alternative 1 – R32 @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	27	31	14%
Cooling Capacity	BTU/hr	21450	23653	10%
Total Power	kW	2.74	2.67	-2%
EER	BTU/hr.W	7.84	8.86	13%

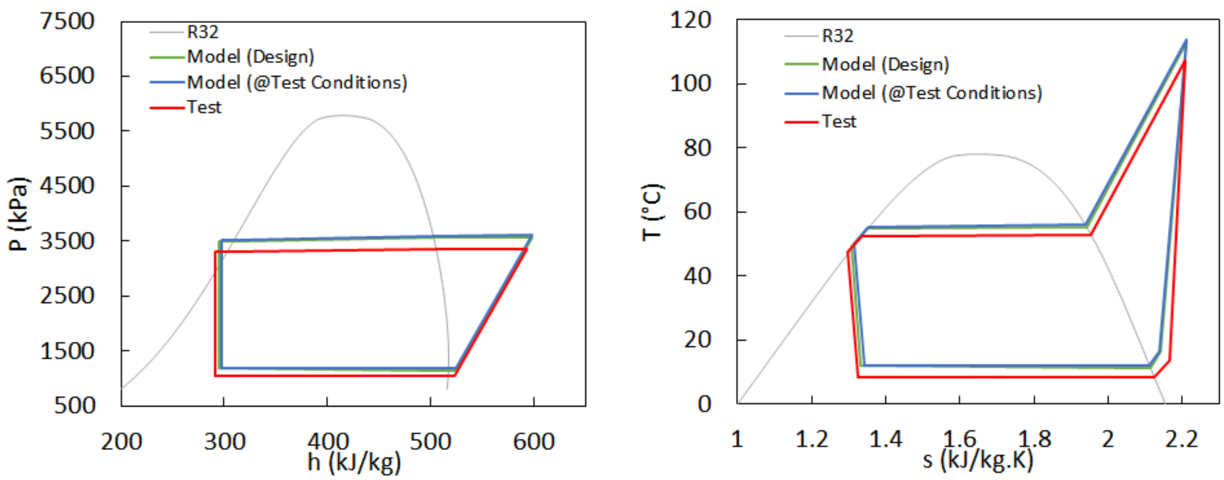

Figure 23. Unit 6 – R32 Performance Test Summary P-h and T-s Diagrams.

Table 40: Unit 6 – Model Verification and Validation for Alternative 2 – R454B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	35	36	3%
Cooling Capacity	BTU/hr	21821	22969	5%
Total Power	kW	2.67	2.49	-7%
EER	BTU/hr.W	8.17	9.24	13%

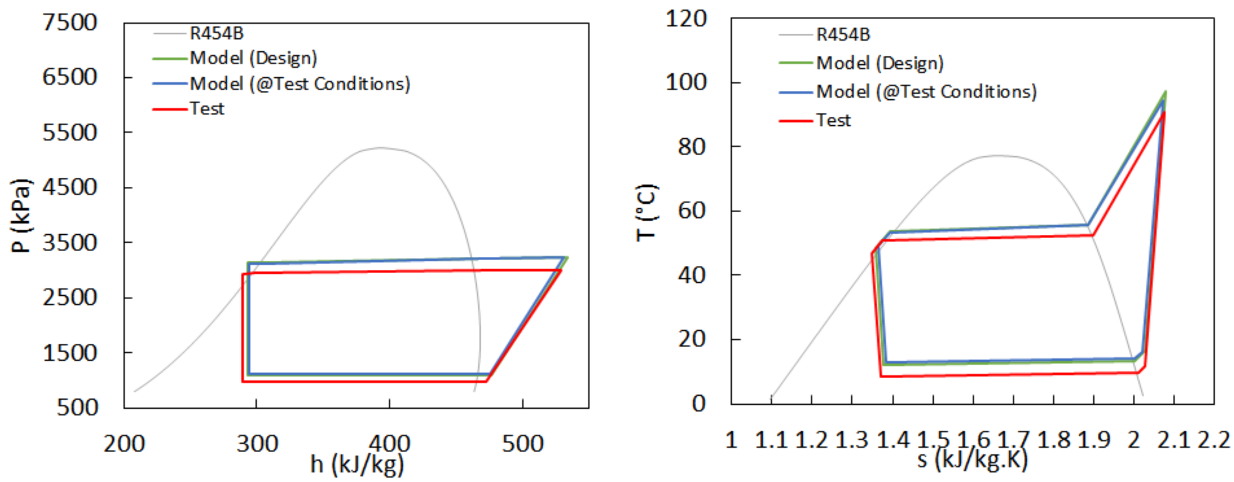

Figure 24. Unit 6 – R454B Performance Test Summary P-h and T-s Diagrams.

Table 41: Unit 10 – Model Verification and Validation for Alternative 1 – R447B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	51	49	-3%
Cooling Capacity	BTU/hr	31169	31026	-0.5%
Total Power	kW	2.70	3.00	11%
EER	BTU/hr.W	11.54	10.34	-10%

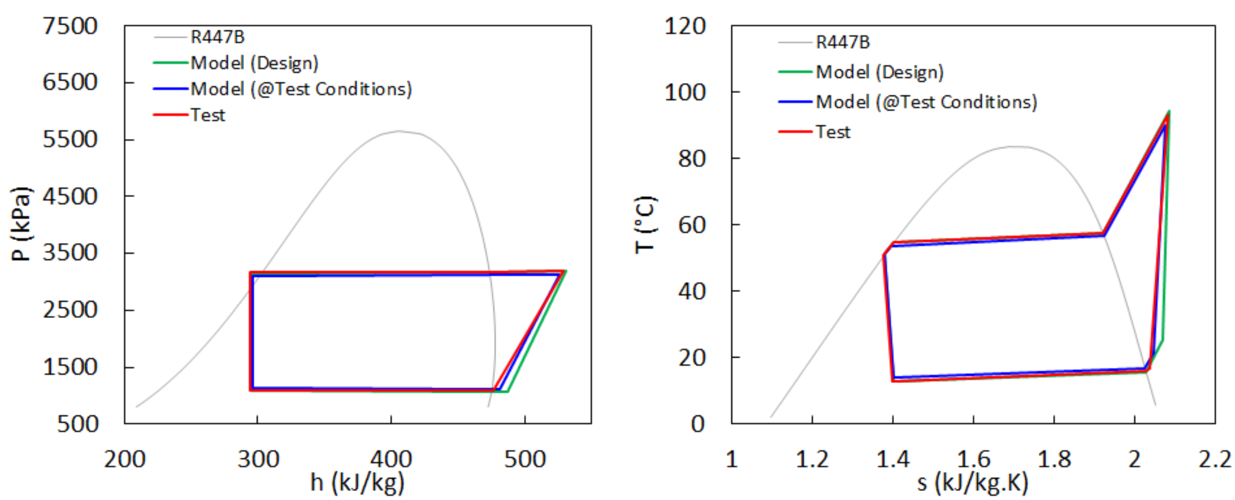

Figure 25. Unit 10 – R447B P-h and T-s Diagrams.

Table 42: Unit 10 – Model Verification and Validation for Alternative 2 – R452B @ 46°C.

		Test	Model (Test Conditions)	Relative Difference
Refrigerant Mass Flow Rate	g/s	47	48	2%
Cooling Capacity	BTU/hr	30292	30704	1.4%
Total Power	kW	3.90	3.34	-14%
EER	BTU/hr.W	7.76	9.19	18%

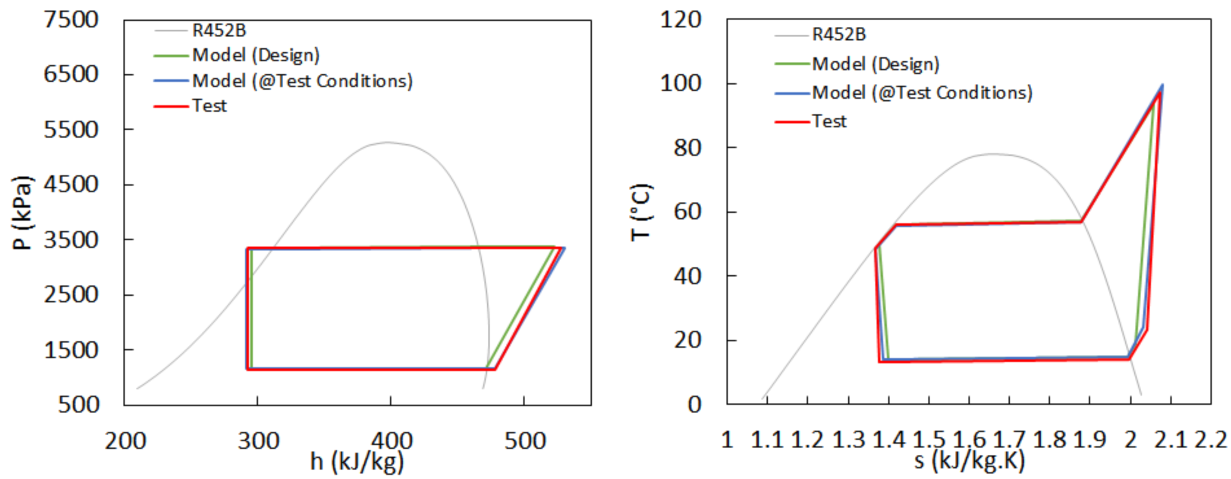

Figure 26. Unit 10 – R452B P-h and T-s Diagrams.