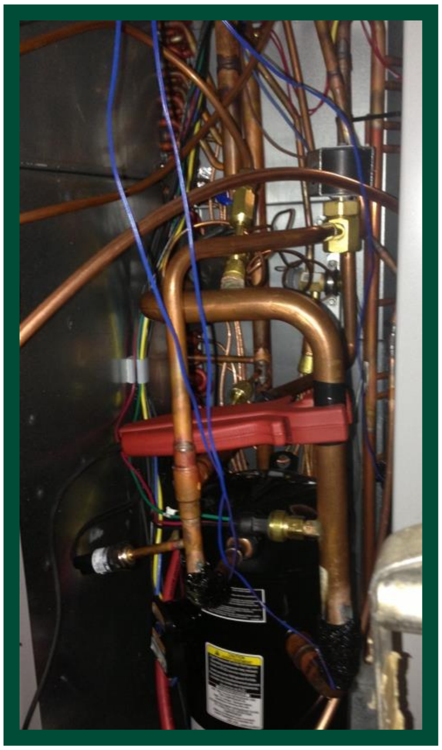
Emerging Technologies

# DEVELOPMENT OF A FAULT DETECTION AND DIAGNOSTICS LABORATORY TEST METHOD FOR A COMMERCIAL PACKAGED ROOFTOP UNIT

*ET13SCE7030*



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Southern California Edison's (SCE's) Technology Test Centers (TTC) is responsible for this project. It was developed as part of Southern California Edison's HVAC Technologies and System Diagnostics Advocacy (HTSDA) program under internal project number HT.11.SCE.002 and subsequently carried over into SCE's Emerging Technologies Program (ET) under the project number ET13SCE7030. TTC project manager Sean Gouw conducted this test method development with overall guidance and management from line manager Ramin Faramarzi, and program manager Jerine Ahmed. For more information on this project, contact sean.gouw@sce.com.

# EXECUTIVE SUMMARY

This project intends to break new ground in the world of fault detection and diagnostics (FDD), and improved heating, ventilating, and air conditioning (HVAC) performance through enhanced maintenance. FDD and HVAC maintenance show ample opportunity to achieve and maintain significant energy and demand savings in support of strategic initiatives, goals, and policies across California. The project's goal is to develop a laboratory test method for FDD technologies for a commercial packaged rooftop unit (RTU) air conditioner. The test method presented in this report is part of many ongoing efforts needed to continually explore solutions to the complex issues inherent with FDD and HVAC maintenance.

The test method imposes single and multiple cooling-mode faults under steady-state conditions. The test method is used to explore the outputs of three FDD technologies in project ET13SCE7040 ("Laboratory Assessment of Fault Detection and Diagnostics Technologies on a Commercial Packaged Rooftop Unit"), and evaluate HVAC fault impacts in project ET13SCE7050 ("Evaluating the Effects of Common Faults on a Commercial Packaged Rooftop Unit"). The Air Conditioning, Heating, and Refrigeration Institute (AHRI) 2008 Standard 210/240 provided the foundation for this test method. This project leveraged industry knowledge through engagement of a Technical Advisory Group (TAG) and the Western HVAC Performance Alliance (WHPA) FDD committee. Experience was also leveraged from a previous FDD/HVAC maintenance study by Southern California Edison's (SCEs) Technology Test Centers (TTC).[VIII] The test method was implemented by a third-party AHRI-certified lab, with some additional follow-up testing conducted at SCE's TTC.

This project successfully developed a steady-state test method suitable for replicating HVAC faults in a laboratory environment, but is not intended to be the final and universal solution to fully understand FDD and HVAC maintenance. This lab test method does not capture transient impacts of faults, and cannot inform of the actual severity, incidence, and prevalence of faults experienced by equipment in the field. The overwhelming permutations of fault severities, fault combinations, indoor/outdoor conditions, and HVAC equipment characteristics make laboratory testing a potentially large burden for directly exploring FDD technologies via lab testing alone.

Industry acceptance of an FDD laboratory test method should continue to be a priority for key stakeholders in the HVAC maintenance/FDD industry (utilities, HVAC manufacturers, HVAC service contractors, FDD developers, etc.), with a clear understanding of how it fits into a combination of other diverse efforts. California utilities should continue their efforts to lead and support these activities. Ideally, field efforts, lab efforts, and simulation efforts across all stakeholders will be cohesively orchestrated and leveraged to best understand and enhance FDD and HVAC maintenance. In this scheme, a larger variety of scenarios can be explored, in an informed, effective manner. An ideal scheme of efforts should include the following:

- Well-trained and experienced field specialists who use best practices and technologies to implement/promote quality HVAC maintenance, and inform laboratory testers and simulation experts.
- Laboratory testers who adhere to a standardized lab test method to generate and compile key data across a variety of important scenarios, and work with field specialists and simulation experts to develop/explore/enhance technologies and best practices.
- Simulation experts who work with laboratory testers and field specialists, and leverage validated simulation/modeling techniques to explore mathematical, field, and lab-generated data to develop/explore/enhance technologies and best practices.

An enhanced understanding of FDD and a standardization of terms and practices allows for broader adoption of reliable, accurate, cost-effective FDD methods and technologies and ultimately widespread enhancement and persistence of HVAC performance. The following activities are recommended with regards to an FDD lab test method:

- Coordinate with industry leaders through venues such as the WHPA FDD committee in a manner in-line with the committee's research roadmap.
- Continue to disseminate findings and engage industry through organizations such as the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), American Council for an Energy-Efficient Economy (ACEEE), and the WHPA FDD committee.
- Support the efforts of ASHRAE SPC207P to ensure a lab test method is developed that generates data that is reliable, repeatable, reasonably representative of field conditions, and helps to enhance the understanding of FDD performance and the objective distinctions of various FDD technologies.
- Use data generated by an industry-accepted lab test method to evaluate FDD technologies that are considered for adoption into utility energy efficiency rebate programs, or California Statewide or Federal Codes and Standards.
- Conduct studies to characterize faults encountered in the field to inform a prioritization of lab test scenarios that should be investigated; characteristics include fault type, severity, prevalence, and incidence.
- Investigate the transient impacts of faults associated with cyclic laboratory testing; consider adoption into the lab test method based on the merits of the results.
- Investigate and enhance current mechanisms to run simulations for FDD and fault impact evaluations, based on reliable lab data generated by an industry-accepted FDD lab test method.
- Investigate the troubleshooting performance of manual diagnostics, by both certified and non-certified technicians, with and without the assistance of FDD technologies.
- Investigate the variances in lab test methods, FDD performance, and fault impacts across key equipment characteristics/configurations, such as (not limited to) refrigerant types, heat exchanger types, expansion device types.

# ABBREVIATIONS AND ACRONYMS

| | |
|---|---|
| AFDD | Automated Fault Detection and Diagnostics |
| AHRI | Air Conditioning, Heating and Refrigeration Institute |
| AMB | Ambient |
| ANSI | American National Standards Institute |
| ASHRAE | American Society of Heating, Refrigerating and Air Conditioning Engineers |
| BACnet | Building Automation and Control Networks (Communications Protocol) |
| Btu | British Thermal Unit |
| CASE | Codes and Standards Enhancement |
| CI | Capacity Index |
| COA | Condensing (temperature) Over Ambient |
| CT | Condensing Temperature |
| CZ | Climate Zone |
| DB | Dry-Bulb Temperature |
| DES | Design and Engineering Services |
| DP | Dew Point |
| EE | Energy Efficiency |
| EER | Energy Efficiency Ratio[1] |
| EI | Efficiency Index |
| ET | Evaporator Temperature (Saturated) |
| ETO | Education, Training, and Outreach |
| °F | Degrees Fahrenheit |

[1] The term 'EER' is used throughout this report as a measure of instantaneous efficiency, across a multitude of possible indoor/outdoor conditions and faults, rather than limited to typical equipment rating conditions.

| | |
|---|---|
| FDD | Fault Detection and Diagnostics |
| hr | Hour |
| HTSDA | HVAC Technologies and System Diagnostics Advocacy |
| HVAC | Heating, Ventilating, and Air Conditioning |
| ITD | Indoor Temperature Drop |
| kW | Kilowatt |
| kWh | Kilowatt-hour(s) |
| lbs. | Pounds |
| oz | Ounce(s) |
| LP | Liquid Pressure |
| LT | Liquid Temperature |
| min | Minute |
| PDA | Personal Digital Assistant |
| PIER | Public Interest Energy Research |
| psi | Pounds per square inch |
| °R | Degrees Rankine |
| RA | Return Air |
| RH | Relative Humidity |
| RTU | Rooftop Unit (Packaged) |
| RWB | Return Wet-Bulb |
| SA | Supply Air |
| SC | Sub-cooling |
| SCE | Southern California Edison |
| SCFM | Standard Cubic Feet per Minute |
| SH | Superheat |

| | |
|---|---|
| SME | Subject Matter Expert |
| SP | Suction Pressure |
| ST | Suction Temperature |
| SWB | Supply Wet-Bulb |
| TAG | Technical Advisory Group |
| TTC | Technology Test Center |
| TxV | Thermostatic Expansion Valve |
| T/C | Thermocouple |
| W | Watt |
| WB | Wet-Bulb Temperature |
| WHPA | Western HVAC Performance Alliance |

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# INTRODUCTION

This project intends to break new ground in the world of fault detection and diagnostics (FDD), and improved heating, ventilating, and air conditioning (HVAC) performance through enhanced optimization. FDD and HVAC optimization show ample opportunity to achieve and maintain significant energy and demand savings in support of strategic initiatives, goals, and policies across California.

## POLICY DRIVERS

The Global Warming Solutions Act of 2006, or Assembly Bill (AB) 32

"In 2006, the Legislature passed and Governor Schwarzenegger signed AB 32, the Global Warming Solutions Act of 2006, which set the 2020 greenhouse gas emissions reduction goal into law. It directed the California Air Resources Board (ARB or Board) to begin developing discrete early actions to reduce greenhouse gases while also preparing a scoping plan to identify how best to reach the 2020 limit. The reduction measures to meet the 2020 target are to be adopted by the start of 2011."[2]

California Long Term Energy Efficiency Strategic Plan

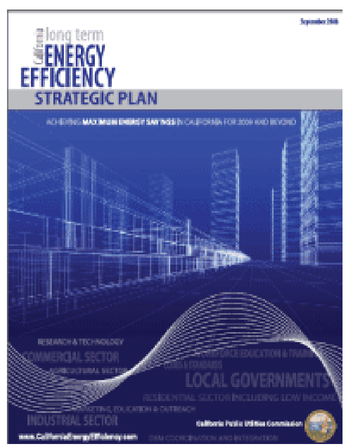


**FIGURE 1. CALIFORNIA LONG TERM ENERGY EFFICIENCY STRATEGIC PLAN**

"On Sept. 18, 2008, the CPUC adopted California's first Long Term Energy Efficiency Strategic Plan, presenting a single roadmap to achieve maximum energy savings across all major groups and sectors in California. The Strategic Plan was subsequently updated in January 2011 to include a lighting chapter.

This comprehensive Plan for 2009 to 2020 is the state's first integrated framework of goals and strategies for saving energy, covering government, utility, and private

[2] http://www.arb.ca.gov/cc/ab32/ab32.htm

sector actions, and holds energy efficiency to its role as the highest priority resource in meeting California's energy needs."[3]

"6. Heating, Ventilation and Air Conditioning

…

Goal 4: New climate-appropriate HVAC technologies (equipment and controls, including system diagnostics) are developed with accelerated marketplace penetration.

The strategies to achieve this goal include:

Commercialize on-board diagnostic systems: Such systems automatically collect data and alert consumers and/or contractors when a fault or negative performance trend is detected. These diagnostics will result in energy benefits by helping ensure that HVAC systems are maintained and operate within design specifications. While many manufacturers currently offer either —on-board systems or hand-held ones that work with all systems, none are widely used by consumers or contractors. Actions to accelerate the commercialization of such diagnostics include:

Prioritizing in-field diagnostic and maintenance approaches based on the anticipated size of savings, cost of repairs, and the frequency of faults occurring.

Benchmarking of existing diagnostic, repair, and maintenance protocols.

Developing nationwide standards and/or guidelines for onboard diagnostic functionality and specifications for designated sensor mount locations.

Aggressive promotion of diagnostic systems as a standard offering on all HVAC equipment."

## THE FDD PROJECT SERIES

Southern California Edison (SCE) initiated a series of six projects under the Heating, Ventilating, and Air Conditioning (HVAC) Technologies and System Diagnostics Advocacy (HTSDA) program. Subsequently, three of these projects are being continued under SCE's Emerging Technologies program. The following projects seek to explore several key items regarding Fault Detection and Diagnostics (FDD) technologies:

- **ET13SCE7030** (~~HT.11.SCE.002~~[4])**:** Development of a Fault Detection and Diagnostics Laboratory Test Method for a Commercial Packaged Rooftop Unit (this report)
- **HT.11.SCE.003:** Development of a Fault Detection and Diagnostics Laboratory Test Method for a Residential Split System[I]
- **ET13SCE7040** (~~HT.11.SCE.004~~[4])**:** Laboratory Assessment of ~~Retrofit~~ Fault Detection and Diagnostics ~~Tools~~ Technologies on a Packaged Unit[II]
- **HT.11.SCE.005:** Laboratory Assessment of Retrofit Fault Detection and Diagnostics Tools on a Residential Split System[III]

[3] http://www.cpuc.ca.gov/PUC/energy/Energy+Efficiency/eesp/

[4] Project number and/or title were updated over the course of the project.

- **ET13SCE7050** (~~HT.11.SCE.006~~[4])**:** Evaluating the Effects of Common Faults on a Commercial Packaged Unit[IV]
- **HT.11.SCE.007:** Evaluating the Effects of Common Faults on a Residential Split System[V]

Projects HT.11.SCE.003, HT.11.SCE.005, and HT.11.SCE.007 focus on a residential split system air conditioner. Projects ET13SCE7030, ET13SCE7040, and ET13SCE7050, (~~HT.11.SCE.002~~, ~~HT.11.SCE.004~~, and ~~HT.11.SCE.006~~[4]) focus on a commercial packaged rooftop unit (RTU) air conditioner. The general strategy behind the residential and commercial projects is to:

- Develop a working laboratory test method;
- Apply the working test method in laboratory assessment projects;
- Update the working test method, as concurrent with lessons learned in the laboratory assessments; and
- Using the data from the laboratory assessment, report on the FDD performance and the observed effects of faults.

## INDUSTRY INPUT

Industry input was important during development and scoping of the commercial FDD project series. Channels such as the Western HVAC Performance Alliance (WHPA) provided the means to provide input. In particular, the WHPA's Onboard/In-Field Fault Detection Diagnostics Committee (formerly known as the "Automated Fault Detection and Diagnostics Subcommittee") played an important role in the realization of the FDD project series by establishing an industry roadmap and bringing together various stakeholders to meet on a regular basis.[5]

Involvement with the FDD committee included frequent updates of concurrent FDD related efforts. One such effort was a Codes and Standards Enhancement (CASE) FDD proposal for Title-24, Part 6 (2013 California Energy Code). Part of this effort included listing of the "highest priority" faults for the CASE proposal to explore. This list, presented and vetted through the FDD committee, is the basis of the scope of faults that this FDD project series will explore.

The following <u>overlying scope of faults</u> was established for the commercial RTU FDD project series:

- Low Refrigerant Charge
- High Refrigerant Charge
- Refrigerant Liquid Line Restrictions
- Refrigerant Non-condensables
- Evaporator Airflow Reduction
- Condenser Airflow Reduction
- Economizer Faults

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[5] http://www.performancealliance.org/Portals/4/Documents/FDD-Committee-Roadmap-Brief-031714%5B1%5D%5B4%5D.pdf

## THE TECHNICAL ADVISORY GROUP

A Technical Advisory Group (TAG) was established to provide support with specialized HVAC and FDD industry expertise. Specifically, feedback was sought regarding the test method and the scope of test scenarios to explore. When establishing the TAG, efforts were made to include as wide a range of participants as possible. This included outreach to industry members from California utilities, academia, and FDD and HVAC manufacturers. The following organizations participated: University of California Davis' Western Cooling Efficiency Center (WCEC), New Buildings Institute (NBI), Portland Energy Conservation Inc. (PECI), National Institute of Standards and Technology (NIST), Climacheck, Field Diagnostics, Pacific Gas and Electric Company (PG&E), Carrier Corporation, Purdue University, Pacific Northwest National Laboratory (PNNL), Sempra utilities, Taylor Engineering, and University of Nebraska. Several TAG members were also active attendees and participants of the WHPA AFDD subcommittee meetings. TAG communication occurred via e-mail, phone calls, discussion in WHPA AFDD subcommittee meetings, and webinars. TAG feedback was obtained prior to conducting the laboratory assessment and finalization of the project reports.

## PROBLEM DEFINITION

The California commercial sector consumes approximately 67 billion kilowatt-hours (kWh) of electricity annually.[6] Ventilation and cooling consumes 18 billion kWh of electricity annually; ventilation and cooling equates to approximately 27% of the total electricity consumed in the California commercial sector.[6] At least 10% of the energy consumed by HVAC is wasted from excessive run time and equipment and control- problems.[VIII] Cooling contributes to approximately 4 megawatts (MW) of peak demand (non-coincident with the power generation's peak demand) in the California commercial sector.[6] Packaged single zone (PSZ) and split single zone (SSZ) systems comprise 70% of the HVAC system types in the California commercial sector; 81% of PSZs and SSZs have "small-sized" cooling capacity ratings of 65,000 British Thermal Units per hour (Btu/h) and lower.[VI]

[6] California Commercial End Use Survey (CEUS) 2007. http://capabilities.itron.com/CeusWeb/Chart.aspx

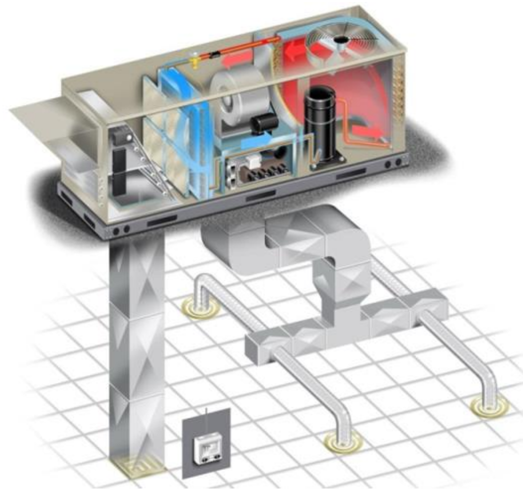

FIGURE 2. PACKAGED ROOFTOP UNIT

Current HVAC maintenance practices face many hurdles and opportunities for enhancement. Traditionally, these practices are open to varying interpretations and are reactive in nature without a preventative maintenance agreement. Even business owners who sign preventative maintenance agreements still face challenges; their agreements are typically restricted due to budget cuts and inadequate knowledge of the maintenance tasks and the frequencies required to preserve HVAC system performance[7]. Regardless, HVAC repair and maintenance is not necessarily aimed at emphasizing optimization of equipment efficiency.

# FAULT DETECTION AND DIAGNOSTICS TECHNOLOGIES

There are no current industry-established classifications or definitions for HVAC FDD technologies. The following FDD definition was observed and generalized for the purposes of this investigation: FDD technologies use an automated means to interpret measurements/parameters to detect symptoms of a faulty operating state, and/or diagnose their root cause(s). For the purposes of this project, FDD technologies are categorized as:

- **"Onboard" FDD:** Technologies permanently installed on HVAC systems for long-term use. These can include products that are factory-installed by the HVAC manufacturer, long-term retrofit products, or FDD-capable thermostats. These technologies may report their findings through a means such as a display on the HVAC system, a thermostat, or some other external system/display.
- **"In-Field" FDD:** Portable technologies that are installed on HVAC systems for temporary use during equipment servicing. These may include dedicated handheld devices or other mobile technologies with FDD software like smart phones, tablets, or laptop computers. These technologies may include their own bundled sensors or have some means to input/interpret measurements/parameters.

FDD technologies have enormous potential to enhance the future of energy efficiency. FDD can provide the information necessary to accurately and reliably

[7] Relationships Matter – Transforming HVAC Through Quality Maintenance. http://www.peci.org/sites/default/files/documents/sce-hvac-aesp.pdf

understand HVAC equipment performance, and improve HVAC maintenance through preventative strategies. Ideally, FDD technologies will be implemented in an automated fashion and outfitted for long-term use with a means for providing remote connectivity. This will enable these technologies to actively inform building operators, homeowners, or service contractors and solicit corrective actions before faults become severe or before critical failures occur. Figure 3 illustrates examples of FDD technologies.

**FIGURE 3. EXAMPLES OF FAULT DETECTION AND DIAGNOSTICS TECHNOLOGIES**

## ANTICIPATED BARRIERS TO ADOPTION OF FDD

Barriers to adopting FDD include:

**Cost Effectiveness:** Cost effectiveness is dependent on the difference between the cost of the FDD technology, and the realized HVAC operating cost reductions. Realizing operating cost reductions are not as straightforward with FDD as it is with other “widget-based” technologies, savings are dependent on:

- Which faults occur in the HVAC system;
- Which faults are detected and diagnosed;
- Which faults are actively corrected; and
- The financial impacts unique to the HVAC owner and application.

**Product Availability and Performance:** The range of commercially available FDD technologies is fairly significant for commercial HVAC. However, the capabilities and performance of these technologies is not well understood and transparent. Currently, industry lacks the means to classify and explore the capabilities and performance of FDD by simulation, laboratory, or field test method. As a result, it is challenging to make comparisons of existing studies of FDD technologies. Additionally, the impacts of HVAC faults are not well understood, especially in scenarios that consist of multiple simultaneous faults.

It is also important to make the distinction between **faults** and **failures**. An HVAC unit may still operate under a fault condition, albeit with significantly detrimental symptoms. Conversely, a failure mode is one that prohibits an HVAC unit from operating at all. It is anticipated that different benefits of FDD are realized through remediation of fault modes rather than failure modes. Failure modes are typically reacted to and resolved regardless of the presence of FDD technologies.

**End-User Need and Interaction:** One potential benefit of FDD technologies is the removal of uncertainties regarding varying interpretations/diagnostics. However, one must consider that there may not be suitable technological replacements for the creative/critical thinking abilities inherent with manual analysis of complex problems. The level of FDD (the extent of manual involvement and automated technologies selected) appropriate for HVAC FDD in a given application needs to be met with the level of need defined for that application. Justified levels remain to be seen through continuing explorations of FDD technologies, the impacts of common faults, and the unique economics that characterize each application. Additionally, many behavioral factors influence whether or not a diagnostic (regardless of its uncertainty) is acted on appropriately and resolved.

# OBJECTIVE

The objective of this project is to develop a laboratory test method for FDD technologies. The test method details procedures to generate faults, and explore the response of FDD to those faults. This report presents the final updated test methodology used in ET13SCE7040, and examines the specific issues and lessons learned in the laboratory assessment.

This test method is developed with the intention of informing SCE's Energy Efficiency Programs, as well as other developing FDD-related efforts such as Codes and Standards Enhancement (CASE) studies for the California Code of Regulations, the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard Project Committee 207P, or the WHPA FDD Committee. Investigation into FDD directly supports the big bold energy efficiency strategies contained in the California Long-Term Energy-Efficiency Strategic Plan (CLTEESP) and supports the goals established by California's AB32.

# TEST METHOD DEVELOPMENT STRATEGY

Consistency with current applicable HVAC testing methodologies is important to the industry acceptance and success of an FDD test method. For this reason, the intention is to leverage as much existing knowledge as possible. The Air Conditioning, Heating and Refrigeration Institute (AHRI) establishes standards for HVAC equipment testing. The AHRI is widely recognized and represents more than 300 heating, water heating, ventilation, air conditioning, and commercial refrigeration manufacturers within the global HVAC industry.

AHRI 210/240-2008[VII] and its incumbent referenced standards (such as ASHRAE Standard 37) were chosen as a basis for the FDD test method to build on. The FDD test method is used for FDD technologies suitable for unitary air-conditioners and air-source unitary heat pumps with nominal capacity under 65,000 Btu/h. The FDD test method will leverage steady state wet-coil cooling mode testing, analogous to tests outlined in AHRI 210/240.

Furthermore, a previous investigation of FDD and HVAC faults was conducted on a packaged RTU at SCE's Technology Test Centers (TTC). This data supported a Public Interest Energy Research (PIER) project[VIII] as well as HVAC maintenance projects[IX] conducted under SCE's Education, Training, and Outreach (ETO) program. The procedures used for that evaluation directly fed into the development of this test method. The resultant draft test method was screened both through various subject matter experts (SMEs) at SCE's TTC, as well as through a TAG, composed of various key industry members.

# THE HVAC TEST UNIT

The HVAC test unit is a high-efficiency (12.45 EER/15.2 SEER), 5-ton (nominal) packaged RTU air conditioner (see Figure 2). The RTU contains a gas-fired heating system; however, it was not used as the laboratory assessment that focused on cooling mode operation only. The test RTU is a pilot-unit from the manufacturer, contains onboard FDD, an economizer, is fixed-capacity (fixed-speed fans and compressor), uses R-410a refrigerant, and uses a thermostatic expansion valve (TxV).

FIGURE 4. HVAC TEST RTU

Various HVAC units exist in commercial applications, comprising a number of different possible physical configurations. This unit is just one possible configuration. It represents a better-than-standard-efficiency unit that is fitted with premium options and that's relevant to the current generation of products that will be aging. Typically, higher efficiency/high-capacity units are more likely to contain premium features like onboard FDD. Other options to explore may include (but are not limited to) those that feature increased cooling capacity (multiple or single stage), R-22 refrigerant, fixed orifice expansion devices, electronic expansion devices, standard-efficiency units, or higher-efficiency units (larger or micro-channel heat exchangers, more efficient compressors, fans, etc.). Ultimately, field studies are needed to best characterize the various equipment types and inform industry-wide FDD and HVAC maintenance efforts (ASHRAE SPC207P, WHAP FDD committee, utility energy efficiency rebate programs, California Statewide or Federal Codes and Standards) about what is most prevalent in the field.

# FDD TEST UNITS

Three FDD technologies were chosen as test units for this assessment. Two are characterized as in-field technologies and one is characterized as onboard. Each test unit has a wide library of possible diagnostic messages available to convey to end users. Not all diagnostic messages across the three FDD test units are pertinent to the scope of faults chosen for this study. Table 1 characterizes the faults that are pertinent to each FDD test unit.

TABLE 1. BASELINE TEST SCENARIOS

| FAULT | FDD TEST UNIT A | FDD TEST UNIT B | FDD TEST UNIT C |
|---|---|---|---|
| Low Charge | ✓ | ✓ | ✓ |
| High Charge | ✓ | ✓ | ✓ |
| Liquid Line Restrictions | ✓ | ✓ | ✓ |
| Non-Condensables | X | ✓ | X |
| Evaporator Airflow Reduction | ✓ | ✓ | ✓ |
| Condenser Airflow Reduction | ✓ | ✓ | ✓ |
| Economizer Mechanical/Communications Faults | ✓ | X | X |

The onboard FDD came installed on the test RTU as a factory option. It features its own physical display interface that is accessible with the removal of one of the RTU's side panels. The interface allows a user to browse through fault codes and various measured and calculated parameters. In addition to the diagnostics messages, the following parameters were selected for consideration, as per their relevance to this FDD study[8]:

1. Supply Air Temperature (SAT), °F
2. Outside Air Temperature (OAT), °F
3. Saturated Suction Temperature (SST), °F
4. Saturated Condensing Temperature (SCT), °F
5. Saturated Suction Pressure (SSP), pounds per square inch (psig)
6. Saturated Condensing Pressure (SCP), psig

The onboard FDD contains 46 unique diagnostic messages. Many of these messages indicate a series of several probable causes; others are traced back to a single cause. Six of the onboard FDD's diagnostic messages are relevant to the scope of faults of this study (see Table 2). The onboard FDD allows for historical access to the last 20 diagnostic messages. Logging for the other parameters is not available. The onboard FDD technology's diagnosis of the HVAC system was recorded for each test scenario. In addition, ten "spot measurements" were recorded for the key parameters that were displayed during each test scenario.

[8] The onboard FDD technology has many different additional accessible parameters that are outside of the scope of these FDD discussions.

**TABLE 2. ONBOARD FDD TEST DIAGNOSTIC MESSAGES**

| DIAGNOSTIC MESSAGE # | DESCRIPTION | PROBABLE CAUSE(S) | |
|---|---|---|---|
| 1A or 1B | Circuit A or Circuit B - Loss of Charge | I | Low refrigerant |
| | | II | Faulty suction pressure transducer |
| 2A or 2B | Circuit A or Circuit B - High Discharge Pressure | I | An overcharged system |
| | | II | High outdoor ambient temperature coupled with dirty outdoor coil |
| | | III | Plugged filter drier |
| | | IV | A faulty high---pressure switch |
| 3A or 3B | Circuit A or Circuit B - Low Refrigerant Pressure | I | Low refrigerant charge |
| | | II | Dirty filters |
| | | III | Evaporator fan turning backwards |
| | | IV | Loose or broken fan belt |
| | | V | Plugged filter drier |
| | | VI | Faulty transducer |
| | | VII | Excessively cold return air |
| | | VIII | Stuck open economizer when the ambient temperature is low. |
| 4 | Loss of communication with the Economizer Actuator | Communication wiring problem with actuator | |
| 5 | Dirty Air Filter | Dirty Air Filter | |
| 6 | Economizer Damper Fault | I | Economizer Damper Actuator Out of Calibration |
| | | II | Economizer Damper Actuator Torque Above Load Limit |
| | | III | Economizer Damper Actuator Hunting Excessively |
| | | IV | Economizer Damper Stuck or Jammed |
| | | V | Economizer Damper Actuator Mechanical Failure |
| | | VI | Economizer Damper Actuator Direction Switch Wrong |

One of the in-field FDD test units is a package of items, intended for use as an enhancement to the service technician’s toolset. These items are familiar in use and setup to those typically used by HVAC service contractors; training is available from the manufacturer. A significant amount of HVAC maintenance best-practices-related information is also available through reference literature and training provided by the FDD manufacturer. The package includes:

- (1) personal digital assistant (PDA) mobile device
- (2) Air-side probes (supply air and return air) with each measuring both dry-bulb (DB) and wet-bulb (WB) temperatures
- (1) Air-side sensor that measures DB temperature (condenser inlet air)
- (2) Clamp-on thermocouple (T/C) sensors (suction and liquid line refrigerant temperatures)

- (3) Refrigerant pressure hoses (high and low side system pressures, for general charging/recovery/evacuation purposes)
- (1) Digital refrigerant manifold

For the purposes of this evaluation, this unit's hoses and manifold were not used for charging/recovery/evacuation. The PDA displays several screens of measurements and calculations. The tool steps through its internal algorithms and displays its diagnosis in real-time fashion. The tool has approximately 49 different diagnostic messages. Twelve of the in-field FDD's diagnostics messages are relevant to the scope of faults of this study. Measurements, calculations, and FDD messages were observed to be simultaneously populated approximately once every three seconds. This tool has zero or limited logging capability; it is able to log one set of readings, which may be uploaded to an online server for reporting. The technology was provided as new, as calibrated from the manufacturer. The PDA displays the following 19 measurements and calculations:

1. Suction pressure (SP), psig
2. Liquid pressure (LP), psig
3. Suction temperature (ST), degrees Fahrenheit (°F)
4. Liquid temperature (LT), °F
5. Ambient air temperature (AMB), °F
6. Return air (RA) DB temperature, °F
7. Return air WB(RWB) temperature, °F
8. Supply air (SA) DB temperature, °F
9. Supply air WB (SWB) temperature, °F
10. **Evaporator temperature (ET), °F**
11. **Superheat (SH), °F**
12. **Condensing temperature over ambient (COA), °F**
13. **Sub-cooling (SC), °F**
14. **Indoor temperature drop (ITD), °F**
15. Efficiency Index (EI)
16. Capacity Index (CI)
17. Power, kilowatts (kW)
18. Runtime, hours
19. Dollar ($) Savings

Measurement and calculation items 1 through 14 were used for testing. For items 10 – 14, (marked in **bold)** the tool has pre-established ranges to detect whether the reported parameter is considered "Low", "Ok (-)", "Ok", "Ok (+)" or "High".

The in-field FDD technology's predominant diagnosis of the HVAC system was recorded. In addition, ten "spot measurements" were recorded for the key parameters that were displayed. However, it was impossible to capture all 19 reported parameters in unison with the FDD technology's refresh rate of about three seconds.

Comment

*Diagnostic messages instantaneously take all measurements and calculations into account and represent the "bottom-line" interpretation of system performance. It is of great interest to evaluate FDD technologies based on what diagnostic is reported.*

*Significant FDD output variance was observed with real-time operation. This, in combination with a three second display refresh rate, presented logistical challenges* *in recording the outputs of the FDD Test Unit. It becomes difficult to tie "spot readings" of measurements and calculations back to the overall diagnosis message, when they all cannot be recorded simultaneously.*

*Access to all 19 reported parameters requires navigation through several different screens. The action of moving from one screen to another requires the user to wait until the next refresh of the display screen. Each screen displays a distinct portion of the 19 parameters and the overall diagnostic message.*

The other in-field FDD test unit is a handheld device. The device contains embedded algorithms for analysis of system parameters that may be obtained either through direct attachment of several types of accessory heads, or manual input from separate independent measurements. Accessory heads can be used wirelessly to send remote measurements to the device. Up to 12 parameters can be measured wirelessly at one time and sent to the device for live viewing and analysis.

The device leads technicians step-by-step through critical tests including: Target Evaporator Exit Temp, Target Superheat, Superheat, Subcooling, Combustion, and a diagnostics program. The diagnostics program is chosen as the focus for this investigation. The algorithms built into the diagnostics program are based on field data of over 250,000 air conditioners. Tolerances built into the algorithms may be adjusted by the user. The diagnostics program contains approximately 36 different diagnostic messages; the diagnostics program has the capability to output combinations of these 36 messages. Thirteen of the in-field FDD's diagnostics messages are relevant to the scope of faults for this study. Table 3 details the categories and descriptions of the 13 diagnostic messages chosen for this study. Some diagnostic messages were not considered because it:

- Required the use of a TrueFlow® grid;
- Applied to HVAC equipment that use fixed orifice metering devices (test RTU featured a TXV);
- Indicated a lack of available measurement data. (All relevant data was input into the device (with the exception of TrueFlow® grid measurement)); or
- Applied to a fault that was not covered in this study.

**TABLE 3. SELECTED SCOPE OF FDD TEST UNIT C DIAGNOSTIC MESSAGES (IN-FIELD)**

| DIAGNOSTIC MESSAGE CATEGORY | # | DESCRIPTION |
|---|---|---|
| Indoor Coil Airflow Diagnosis | 1 | **Probable OK airflow:** The indoor coil airflow was tested by an indirect means (temperature split) and is probably OK. |
| | 2 | **Low airflow, increase airflow until actual temp split matches target temp split. Actual temp split is __°F and target temp split __°F:** The indoor coil airflow is low based on the temperature split. Check the filter and coil and inspect for any restrictions and blockages. Make sure all registers are open. If the airflow remains low, consider increased blower speed and duct system modifications. Supply and return plenum static pressures can be used to diagnose the causes of low airflow. |
| | 3 | **Low capacity or possible high airflow, measure airflow directly:** The temperature split is low. This usually means that the capacity of the system has been reduced due to incorrect refrigerant charge. Higher than expected airflow is rare, but does occur |

| Diagnostic message Category | # | Description |
|---|---|---|
| | | occasionally. Measuring the airflow directly will identify whether or not high airflow is the cause of the low temperature split. |
| Refrigerant Charge Diagnosis | 4 | **Charge OK:** Refrigerant charge was tested using the appropriate method, and it is OK. |
| | 5 | **Possible OK charge:** The primary indicator of refrigerant charge (subcooling for TXV/EXV or superheat for non-TXV) indicates the refrigerant level was OK. However, a secondary indicator reduces the confidence in that diagnosis. Check out any other potential problems indicated. |
| | 6 | **Possible undercharge, possibly add refrigerant:** Try fixing other conditions first and retesting but if this diagnosis persists the system may be undercharged; if no other conditions are triggered, consider adding refrigerant to correct. The amount of refrigerant to add will vary based on the size of the system and the difference between Target and Actual superheat/subcooling. |
| | 7 | **Possible overcharge, possibly remove refrigerant:** Try fixing other conditions first and retesting, but if this diagnosis persists, the system may be overcharged. If no other conditions are triggered, consider recovering refrigerant to correct. The amount of refrigerant to recover will vary based on the size of the system and the difference between Target and Actual superheat/subcooling. |
| | 8 | **Overcharged, recover refrigerant until actual subcooling reaches target subcooling. Actual subcooling is __°F and target subcooling is __°F:** There is too much refrigerant in this TXV/EXV system. Remove refrigerant until the actual subcooling is within ±3°F (Grant = None) of the target subcooling. The closer the actual subcooling is to the target subcooling, the better. |
| | 9 | **Undercharged, add refrigerant until actual subcooling reaches target subcooling. Your actual subcooling is __°F and your target subcooling is __°F:** This TXV/EXV system is low on refrigerant. Add refrigerant until the subcooling is within ±3°F (Grant = None) of the target subcooling. The closer the actual subcooling is to the target subcooling, the better. |
| Refrigerant Lines and Metering Devices Diagnosis | 10 | **Possible liquid line restriction, check liquid line:** Make sure the service shut-off valves are open. Check the liquid line for kinks, tight bends or sections that may have been stepped on or crushed. Check for a large temperature difference between the liquid line at the compressor and at the metering device. |
| Condenser Coil Performance Diagnosis | 11 | **Condenser airflow OK:** The condenser airflow and capacity indications are OK. |
| | 12 | **Low condenser airflow, clean condenser, check condenser fan:** There is insufficient airflow going across the condenser for the needed heat transfer. Check that the condenser coils and fins are clean, aligned, and free of nearby obstructions. Check the fan motor bearings to ensure that the fan is rotating freely. |
| Outdoor Unit Amp Draw Diagnosis | 13 | **Outdoor amp draw OK:** The outdoor unit is running at the proper amperage for the current conditions. |
| | 14 | **High outdoor amp draw, probable excessive compressor friction:** Check other possible causes of high amp draw (low condenser airflow and refrigerant overcharge) before condemning the compressor. Check that condenser coils and fins are clean, aligned, and free of nearby obstructions. |

| Diagnostic message Category | # | Description |
|---|---|---|
| | 15 | **Low outdoor amp draw, possible compressor valve or motor problem:** Check the refrigerant charge before condemning the compressor. |
| Cooling Capacity Diagnosis | 16 | **Low capacity:** This unit is operating under its expected capacity. Check the refrigerant charge, repair if needed, and retest. |

# THE TEST METHOD

In this section, the test method is presented, as well as various discussions on specific key lessons learned by conducting the laboratory assessment.

## SELECTING FAULT THRESHOLDS FOR ANALYSIS

To explore FDD outputs, a fault threshold must be selected to establish a clear performance-based boundary for what is considered a fault. In reality, the acceptable levels will vary depending on the unique needs that exist throughout a wide range of commercial end-users. For the purposes of this study, the fault threshold chosen is when EER degradation (air-side or refrigerant-side) is greater than 10%; if EER degradation (air-side or refrigerant-side) is less than or equal to 10%, it is not considered a fault. For more details about the steady-state performance impacts of faults on the RTU, please see the ET13SCE7050 report.

Additionally, overcharge greater than 5% is considered a fault. This is due to the fact that although steady-state performance impacts from overcharge are not as pronounced as other faults, system reliability concerns still exist (slugging the compressor).

## ANALYZING FDD OUTPUTS

FDD performance is analyzed according to functionality that was specified by the corresponding FDD manufacturer/developer. There is little to no consistency in the functionality witnessed in the FDD technologies selected for this study. The following five potential outputs are generalized for discussion purposes related analysis of FDD[x]:

- **No response**: The FDD protocol cannot be applied for a given input scenario, or does not give an output because of excessive uncertainty.
- **Correct**: The operating condition, whether faulted or unfaulted, is correctly identified.
- **False alarm**: No significant fault is present, but the protocol indicates the presence of a fault.
- **Misdiagnosis**: A significant fault is present, but the protocol misdiagnoses what type of fault it is.
- **Missed Detection**: A significant fault is present, but the protocol indicates that no fault is present.

It is also important to make a clear distinction between fault diagnosis, and symptom detection. For this investigation, the following definitions are adopted:

- **Symptom detection**: The function of FDD to identify the magnitude of deviation in one or more operating parameters, from what may be considered typical or expected from normal operation. Examples of operating parameters in an HVAC system are high-side and low-side pressures, superheat, sub-cooling, indoor and outdoor unit airflow rate, or cooling capacity. “Low cooling capacity” is a type of FDD output that is considered symptom detection.
- **Fault diagnosis:** The function of FDD to identify the presence of one or more specific faults that cause an HVAC system to exhibit symptoms. Faults may be

indicated in varying degrees of specificity. For example, a high level could be considered as "RTU fault exists", down to "a condenser issue exists" down to a "condenser circuit 1 heat exchanger fouling exists". A hierarchy of fault families can be conceptualized to varying degrees.

Faults are considered to be the root cause and symptoms are the deviations in specific HVAC system parameters. Some faults may exhibit similar symptoms. For example, a low charge fault and an evaporator airflow reduction fault may exhibit symptoms of reduced sub-cooling, reduced low-side refrigerant pressure, and reduced cooling capacity. Discussion will identify if the FDD clearly indicate a specific fault, or if it indicates symptoms with multiple possible causes.

For the purposes of this study, which covers both single and multiple fault scenarios, the following logic is adopted and illustrated in Figure 5 below. FDD may have the capability to output multiple simultaneous messages. In this case, all messages are grouped together and considered as one overall message. In the case where multiple messages occur, but not simultaneously (transient fluctuations), the most prevailing message was selected for analysis.

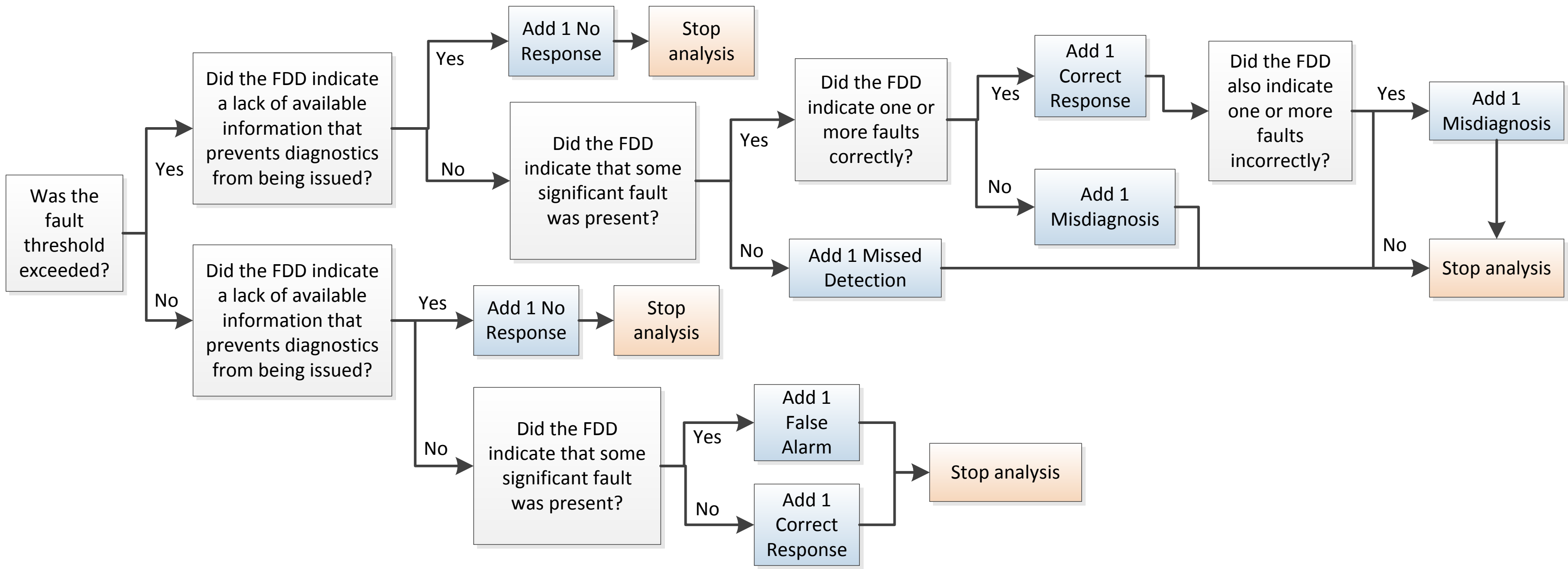


**FIGURE 5. FLOWCHART OF ANALYZING FDD OUTPUTS (PER TEST SCENARIO)**

# TEST EQUIPMENT INSTALLATION, INSTRUMENTATION, AND DATA ACQUISITION

Testing was conducted at a third-party, AHRI-certified, private laboratory. The RTU was installed with guidance from manufacturer-provided literature, and the specifications of AHRI 210/240-2008 and its incumbent referenced standards ASHRAE 37-2009 ("Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment"), and ASHRAE Standard 41.2-1987, ("Standard Methods for Laboratory Airflow Measurement", etc.), with the exception of the gas hook-up for heating.

A refrigerant mass flow meter was installed on the liquid line. A ball valve was installed on the liquid line, after the mass flow meter, for liquid line restriction testing. Sight-glasses were also installed to assist in identifying the presence of mixed-phase refrigerant flow in the liquid line. Figure 6, Figure 7, and Figure 8 are presented to highlight the key measured state points on the refrigerant-side and the air-side. Figure 9 and Figure 10 show photographs of the test setup. Table 4 and Table 5 summarize the types of measurements available at each air-side and refrigerant-side state point for all laboratory sensors, onboard FDD sensors, and in-field FDD sensors. In this manner, key measurements and similarly located sensors are tracked and presented. Manufacturer literature and standard laboratory practice guided the installation of the in-field FDD technology and placement of its corresponding sensors. The locations of onboard FDD sensors were left unchanged from the factory setup.

Evaporator volumetric airflow was measured using an ASHRAE Airflow Measurement Apparatus, located downstream of the supply air duct. Condenser volumetric airflow was measured using an ASHRAE Airflow Measurement Apparatus, located downstream of an air duct that is placed directly over the condenser fan leaving airstream. Evaporator and condenser volumetric airflow was measured in units of Standard Cubic Feet per Minute (SCFM). The outside air intake remains blocked off for the duration of all non-economizer fault tests. In addition, condensate from the evaporator was plumbed to a separate container outside of the test chamber. This container was periodically weighed with a scale to determine total condensate collected for a given test scenario, and manually recorded. No agreement could be established between condensate-based latent cooling capacity calculations and psychrometrics-based latent cooling capacity calculations. Condensate-based calculations were ultimately not used as they were deemed unreliable.

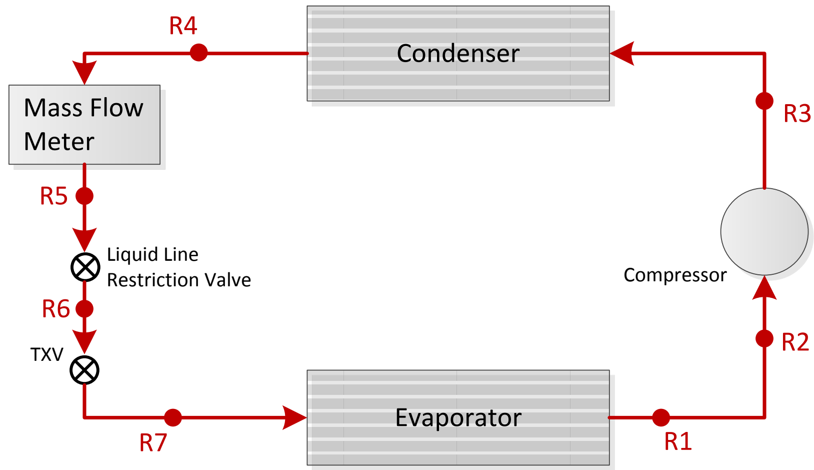


**FIGURE 6. SIMPLIFIED REFRIGERANT-SIDE STATE POINT DIAGRAM**

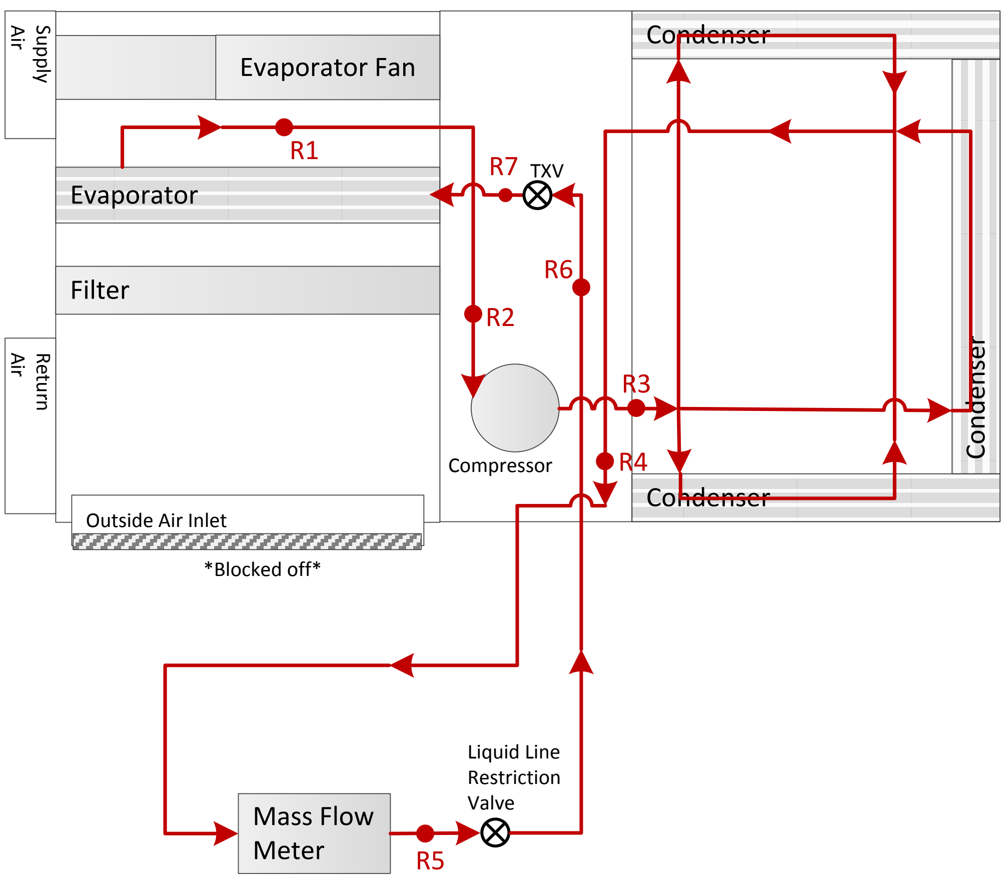


**FIGURE 7. REFRIGERANT-SIDE STATE POINTS**

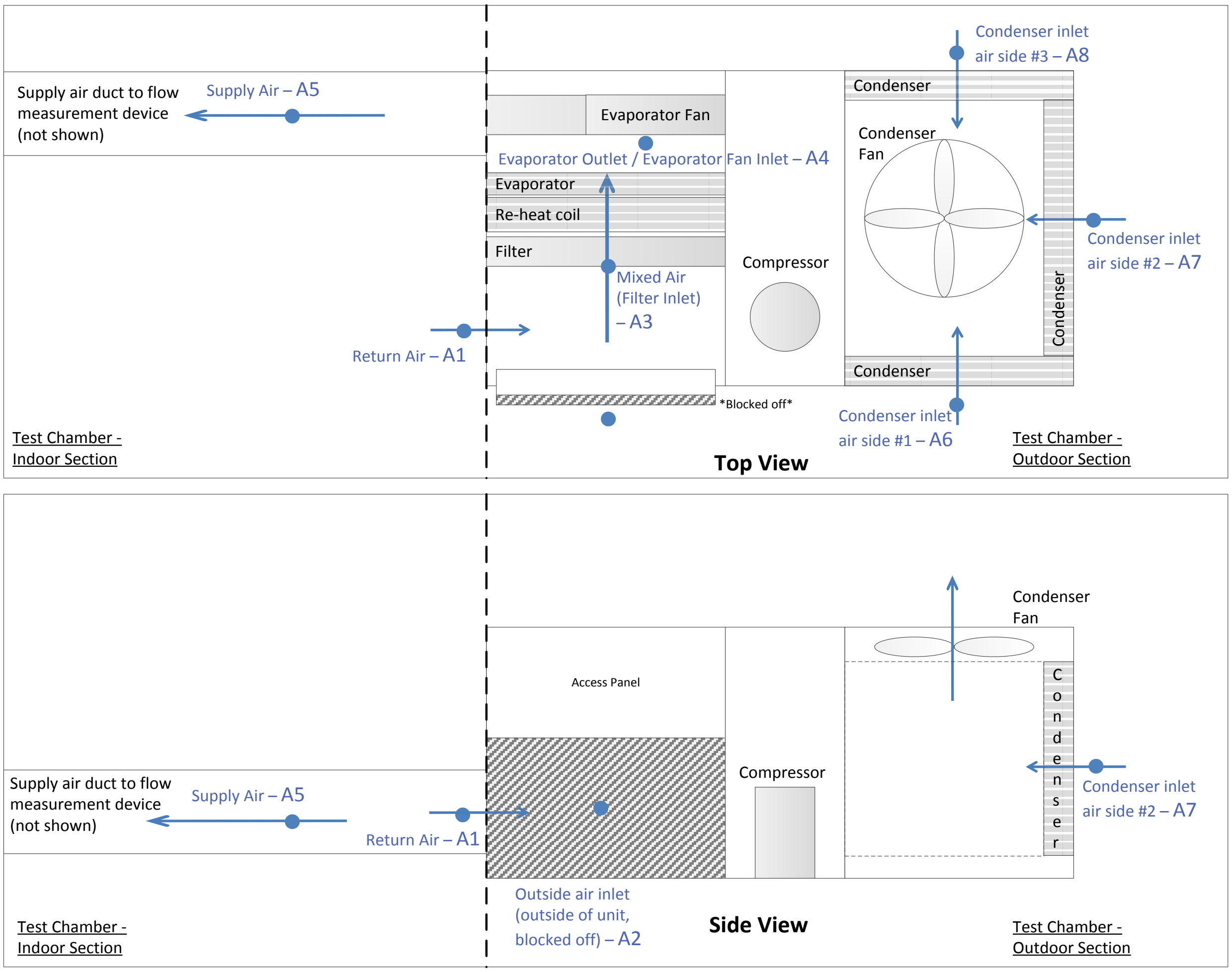


**FIGURE 8. AIR-SIDE STATE POINTS**

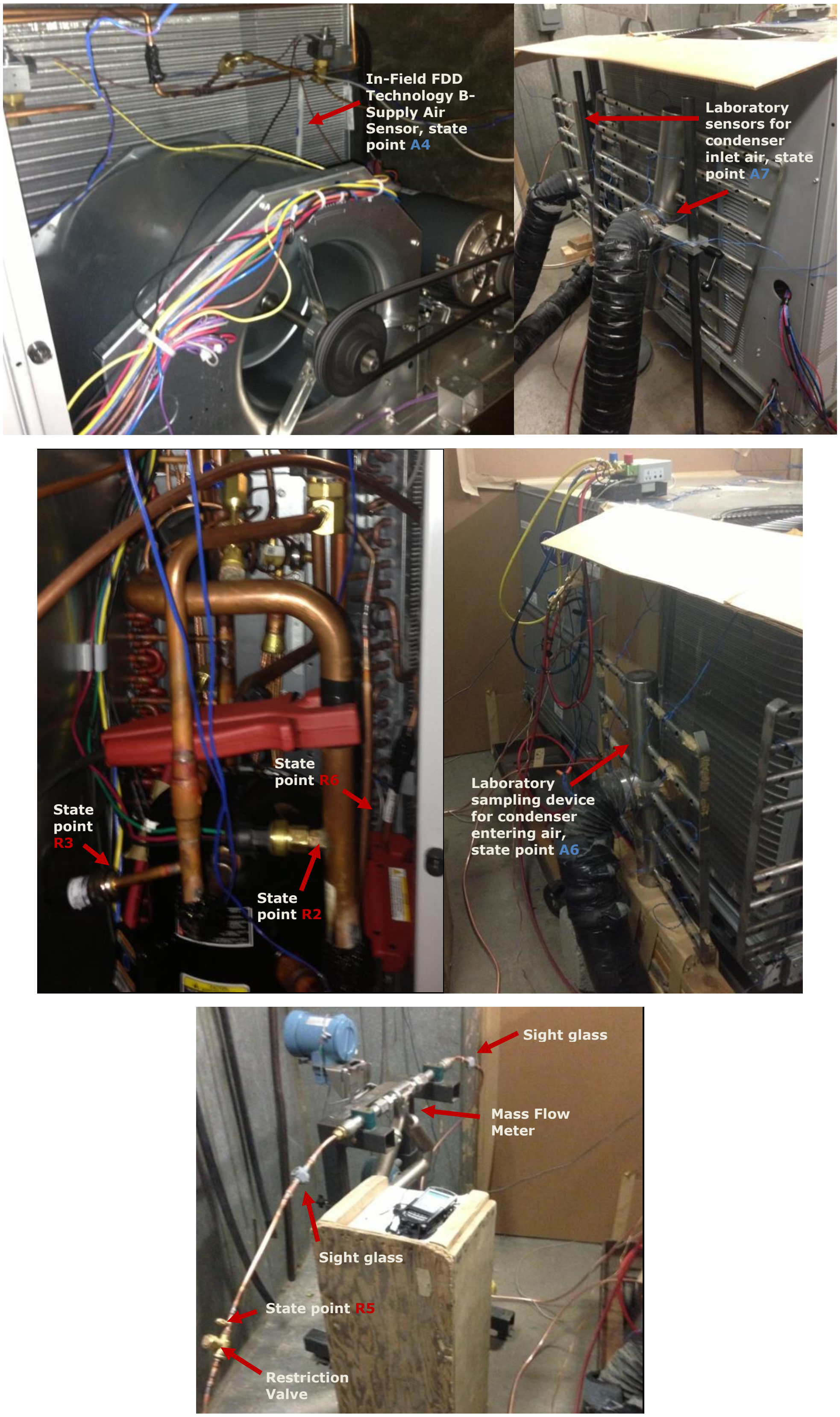


**FIGURE 9. TEST SETUP-OUTDOOR SECTION**

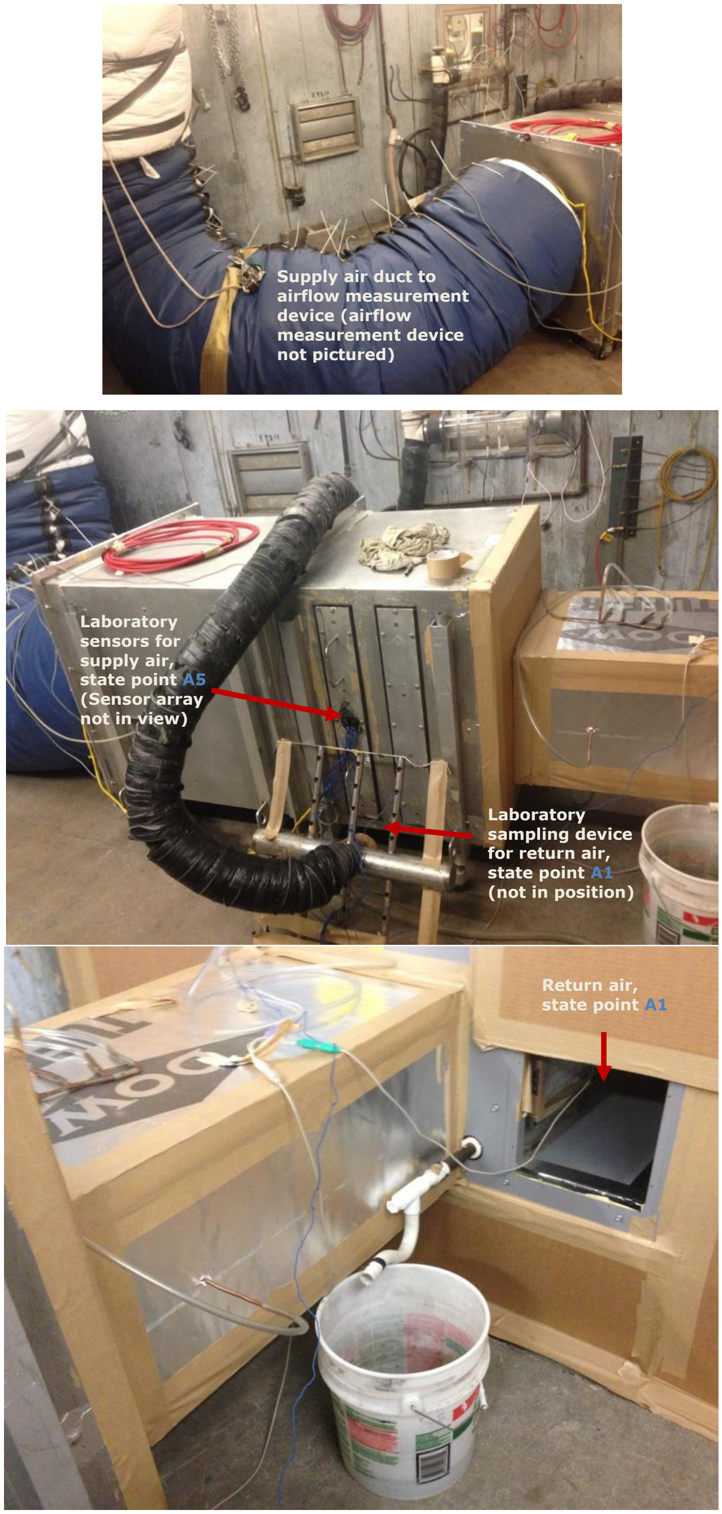


**FIGURE 10. TEST SETUP-INDOOR SECTION**

**TABLE 4. LIST OF STATE POINT MEASUREMENTS – REFRIGERANT-SIDE**

| STATE POINT | DESCRIPTION | LAB SENSORS | ONBOARD FDD SENSORS | IN-FIELD FDD SENSORS |
|---|---|---|---|---|
| R1 | Evaporator Outlet | Pressure (psig), temperature (F) | None | None |
| R2 | Compressor Inlet | Pressure (psig), temperature (F) | Pressure (psig) | Pressure (psig), temperature (F) |
| R3 | Condenser Inlet | Pressure (psig) | Pressure (psig) | None |
| R4 | Condenser Outlet | Pressure (psig), temperature (F) | None | None |
| R5 | Mass Flow Meter Outlet | Refrigerant mass flow (lb/min) | None | None |
| R6 | Expansion Device Inlet | Pressure (psig) | None | Pressure (psig), temperature (F) |
| R7 | Expansion Device Outlet* | None | None | None |

*Note: R7 is a theoretical point; it is assumed that R7 has enthalpy equal to that of R6, for laboratory refrigerant-side cooling capacity calculations. Physical measurements at R7 are problematic in nature; after passing through the expansion device, refrigerant passes through a distributor, which creates a multitude of parallel piping entries into the evaporator coil with limited space for instrumentation.

**TABLE 5. LIST OF STATE POINT MEASUREMENTS – AIR-SIDE**

| STATE POINT | DESCRIPTION | LAB SENSORS | ONBOARD FDD SENSORS | IN-FIELD FDD SENSORS |
|---|---|---|---|---|
| A1 | Return Air | Dry-bulb and wet-bulb temperature (F) | None | None |
| A2 | Outside Air Entering | None | Dry-bulb temperature (F) | None |
| A3 | Mixed Air | None | None | Dry-bulb and wet-bulb temperature (F) |
| A4 | Evaporator Leaving / Evaporator Fan Entering | None | None | Dry-bulb and wet-bulb temperature (F) |
| A5 | Supply Air | Dry-bulb and wet-bulb temperature (F) | Dry-bulb temperature (F) | None |
| A6 | Condenser Entering – Side #1 | Dry-bulb and wet-bulb temperature (F) | None | None |
| A7 | Condenser Entering – Side #2 | Dry-bulb and wet-bulb temperature (F) | None | Dry-bulb temperature (F) |
| A8 | Condenser Entering – Side #3 | Dry-bulb and wet-bulb temperature (F) | None | None |

# CONTROL PARAMETERS AND TEST INTERVALS

All testing was conducted similarly to the steady-state wet coil tests outlined in AHRI – 210/240-2008. All test scenarios encompass a 1-hour span of data. This hour comprises a 30-minute pre-test interval, followed by a 30-minute data collection interval. Reported parameters are straight averages across the 30-minute data collection interval. Table 6 lists the targeted control parameters used for testing.

**TABLE 6. CONTROL PARAMETERS**

| CONTROL PARAMETER | TEST OPERATING TOLERANCE | TEST CONDITION TOLERANCE | TARGET | UNITS |
|---|---|---|---|---|
| Outdoor Test Chamber DB: Cond inlet DB | 2.0 | 0.5 | 95, 80, or 115 | °F |
| Indoor Test Chamber DB: Evaporator fan inlet DB | 2.0 | 0.5 | 80 | °F |
| Evaporator outlet DB | 2.0 | N/A | N/A | °F |
| Indoor Test Chamber WB: Evaporator fan inlet WB | 1.0 | 0.3 | 67 | °F |
| Evaporator outlet DP (calc'd equivalent) | ~2.8 | N/A | N/A | °F |
| Supply duct RH (calc'd equivalent) | ~8 | N/A | N/A | % |
| Electrical Voltage | 2.0 | 1.5 | (230 V) | % of reading |
| Nozzle Press Drop | 2.0 | N/A | N/A | % of reading |

# CALCULATIONS

Various calculation methods are available for laboratory testing. Table 7 lists the calculation methods used in this project. A comprehensive summary of calculation methods applicable to a given test scenario may be found in the appendix, in Table 17.

**TABLE 7. CALCULATION METHODS**

| TEST # | CALCULATION METHODS | CALCULATED PARAMETERS |
|---|---|---|
| 1 | Refrigerant-side measurements and calculations | Enthalpies, saturated temperatures, gross cooling capacity, EER |
| 2 | Compressor regression -> refrigerant-side measurements and calculations | Gross cooling capacity, refrigerant mass flow, compressor power |
| 3 | Air-side measurements and calculations | Enthalpies, net cooling capacity, EER |

**Percent difference** is defined as the difference between two values, divided by the average of the data set. This data set may comprise the two values, or it may comprise several other values. For the purposes of this project, it is used when comparing different methods of calculations of a certain parameter. Percent difference is given by the following equation:

**EQUATION 1. CALCULATING PERCENT DIFFERENCE**

$$\%\,Difference = \frac{Value_1 - Value_2}{\frac{1}{2}(Value_1 + Value_2)} \times 100\%$$

**Percent change** is defined as the relative shift in a parameter, or the change of two values divided by one original value. Percent change is used when comparing a parameter from one fault test scenario, to its baseline scenario (shift in a parameter due to a fault). The following equation provides the percent change.

**EQUATION 2. CALCULATING PERCENT CHANGE**

$$\%\,Change = \frac{Value_1 - Value_2}{Value_1} \times 100\%$$

Energy Efficiency Ratio (EER) calculations were performed as follows:

**EQUATION 3. GROSS ENERGY EFFICIENCY RATIO**

$$EER_R = \frac{\dot{Q}_R}{P}$$

Or

$$EER_A = \frac{\dot{Q}_{A-G}}{P}$$

Where:

| | |
|---|---|
| $EER_R$ | = Energy Efficiency Ratio (refrigerant-side-based), Btu/hr/Watt (W) |
| $EER_A$ | = Energy efficiency ratio (air-side-based), Btu/hr/W |
| $\dot{Q}_R$ | = Refrigerant-side gross cooling capacity, Btu/hr |
| $\dot{Q}_{A-G}$ | = Air-side gross cooling capacity, Btu/hr |
| $P$ | = Total power (compressor + fans + misc.), W |

Refrigerant-side calculations for gross cooling capacity were performed as follows:

**EQUATION 4. REFRIGERANT-SIDE GROSS COOLING CAPACITY**

$$\dot{Q}_R = \dot{m}_R \times (h_{R1} - h_{R7})$$

$$h_{R7} = h_{R4}$$

Where

| | |
|---|---|
| $\dot{Q}_{R-C}$ | = Refrigerant-side gross cooling capacity, Btu/hr |
| $\dot{m}_R$ | = Refrigerant mass flow rate, lbs. /hr |
| $h_{R1}$ | = Enthalpy at refrigerant-side state point R1, Btu/lb |
| $h_{R7}$ | = Enthalpy at refrigerant-side state point R7, Btu/lb |
| $h_{R4}$ | = Enthalpy at refrigerant-side state point R4, Btu/lb |

In addition, the HVAC unit’s compressor manufacturer provided compressor regression curves, and is able to output cooling capacity, refrigerant mass flow rate, and compressor power. Saturated evaporating temperatures and condensing temperatures, based on pressures measured at state points R2 and R3, respectively, were used to generate data. This data was used as a reference point to establish confidence in existing measurements/calculations for baseline tests 1, 2, and 3.

Compressor regression outputs and test measurements/calculations, along with the associated percent differences (all rounded to the nearest one) are presented in Table 8, Table 9, and Table 10. Percent differences between refrigerant-side calculations and compressor regressions for gross cooling capacity ranged from -2% to -7%. Percent differences for compressor power ranged from 0% to 4%. Percent differences for refrigerant mass flow ranged from 2% to 4%.

**TABLE 8. BASELINE GROSS COOLING CAPACITY: REFRIGERANT-SIDE VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - GROSS COOLING CAPACITY: REFRIGERANT-SIDE (BTU/HR) | VALUE 2 - GROSS COOLING CAPACITY: COMPRESSOR REGRESSIONS (BTU/HR) |
|---|---|---|---|
| 1 | -7% | 53,858 | 57,965 |
| 2 | -4% | 59,602 | 61,825 |
| 3 | -2% | 63,242 | 64,745 |

**TABLE 9. BASELINE COMPRESSOR POWER: MEASURED VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - COMPRESSOR POWER: MEASURED (W) | VALUE 2 - COMPRESSOR POWER: COMPRESSOR REGRESSIONS (W) |
|---|---|---|---|
| 1 | 4% | 4,909 | 4,700 |
| 2 | 0% | 3,879 | 3,888 |
| 3 | 0% | 3,318 | 3,311 |

**TABLE 10. BASELINE REFRIGERANT MASS FLOW: REFRIGERANT-SIDE VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - REFRIGERANT MASS FLOW: MEASURED (LBS./MIN) | VALUE 2 - REFRIGERANT MASS FLOW: COMPRESSOR REGRESSIONS (LBS./MIN) |
|---|---|---|---|
| 1 | 2% | 880 | 859 |
| 2 | 3% | 864 | 837 |
| 3 | 4% | 851 | 818 |

It is important to note that refrigerant-side and compressor regression calculation issues exist for any tests featuring low refrigerant charge or non-condensables. Low charge tests yield mixed phase refrigerant flow in the liquid line. With mixed phase liquid line refrigerant flow, refrigerant properties look-ups become inaccurate and refrigerant mass flow measurements are compromised. In addition, while the regression model may still be suitable for predicting refrigerant mass flow and compressor power, any gross cooling capacity outputs are likely suspect.

For tests with non-condensables, refrigerant mass flow measurements are compromised, and refrigerant properties look-ups for all refrigerant-side state points are no longer applicable. The relationships between system pressures and properties change when pure R-410a is not present, and the mixture of nitrogen vapor and liquid refrigerant likely yields inaccurate refrigerant liquid line mass flow measurements.

Air-side calculations are performed as follows:

**EQUATION 5. AIR-SIDE GROSS COOLING CAPACITY**

$$\dot{Q}_{A-G} = \dot{Q}_{A-N} + \dot{Q}_{EVAP\ FAN}$$

Where

| | |
|---|---|
| $\dot{Q}_{A-G}$ | = Air-side gross cooling capacity, Btu/hr |
| $\dot{Q}_{A-N}$ | = Air-side net cooling capacity, Btu/hr |
| $\dot{Q}_{EVAP\ FAN}$ | = Evaporator fan heat, Btu/hr |

**EQUATION 6. EVAPORATOR FAN HEAT**

$$\dot{Q}_{EVAP\ FAN} = P_{EVAP\ FAN} \times C_1$$

Where

| | |
|---|---|
| $P_{EVAP\ FAN}$ | = Measured Evaporator Fan Power, W |
| $C_1$ | = 3.41214163, Conversion Factor, Btu/hr/W |

**EQUATION 7. AIR-SIDE NET COOLING CAPACITY**

$$\dot{Q}_{A-N} = \dot{m}_A \times (h_{A1} - h_{A6})$$

Where

| | |
|---|---|
| $\dot{Q}_{A-N}$ | = Air-side net cooling capacity, Btu/hr |
| $\dot{m}_A$ | = Indoor air mass flow rate, lbs./hr |
| $h_{A1}$ | = Enthalpy at air-side state point A1, Btu/lb |
| $h_{A6}$ | = Enthalpy at air-side state point A6, Btu/lb |

**EQUATION 8. INDOOR AIR MASS FLOW RATE**

$$\dot{m}_A = \dot{V}_S \times \rho_S \times C_2$$

Where:

| | |
|---|---|
| $\dot{m}_A$ | = Indoor air mass flow rate, lbs./hr |
| $\dot{V}_S$ | = Indoor air volumetric flow rate, “standard” conditions, ft$^3$/min |
| $\rho_S$ | = Density of air = 0.074887, “standard” conditions, lbs./ft$^3$ |
| $C_2$ | = Conversion factor = 60, min/hr |

Table 11 presents the refrigerant-side and air-side gross cooling capacity calculations from baseline tests 1-3, along with percent differences (rounded to the nearest ones value) between the two methods. Percent differences range from 1% to 4%.

**TABLE 11. BASELINE GROSS COOLING CAPACITIES: REFRIGERANT-SIDE VS. AIR-SIDE**

| TEST # | % DIFFERENCE | VALUE 1 - GROSS COOLING CAPACITY: REFRIGERANT-SIDE | VALUE 2 - GROSS COOLING CAPACITY: AIR-SIDE |
|---|---|---|---|
| 1 | 4% | 53,858 | 51,957 |
| 2 | 1% | 59,602 | 59,233 |
| 3 | -4% | 63,242 | 65,607 |

## BASELINE TEST SCENARIOS

Table 12 lists all baseline tests performed for this project.

**TABLE 12. BASELINE TEST SCENARIOS**

| TEST # | DESCRIPTION | INDOOR CHAMBER AIR CONDITION | OUTDOOR CHAMBER AIR CONDITION |
|---|---|---|---|
| 1 | Baseline | 80°F /67°F (DB/WB) | 115°F DB |
| 2 | | | 95°F DB |
| 3 | | | 80°F DB |

Indoor chamber air conditions were maintained at 80°F/67°F (DB/WB), as per the AHRI 210/240 condition. Outdoor chamber air conditions were chosen from TAG input, and from consideration of SCE service territories.

As per the current 2008 Title-24 standards, outdoor design conditions are selected from reference Joint Appendix JA2. For general comfort cooling applications, outdoor conditions are based on the 0.5% cooling DB and Mean Coincident WB values. SCE service territories contain climate zones (CZs) 6, 8, 9, 10, 13, 14, 15, and 16. CZ 15, El Centro, had the most extreme 0.5% cooling DB design condition of 111°F. In addition, California research evaluating the performance of air conditioners optimized for "hot and dry" climates, defines the "hot and dry" DB at 115°F.[xi] Insight on the lower bound of outdoor test chamber design conditions was drawn from the 2009 ASHRAE Fundamentals. CZ6 had the lowest cooling design DB value of the SCE CZs; Los Angeles International Airport's (CZ6) 2% cooling design DB condition was 77.8°F.

Ultimately, 115°F was chosen as the upper limit, 95°F was chosen as the intermediate, and 80°F was chosen as the lower limit for outdoor chamber test conditions. The current standard condition in AHRI 210/240 is 95°F. These tests were conducted with no directly imposed faults, maintaining control parameters at several selections of return and outdoor chamber air conditions. Testing was conducted with onboard FDD technology A and in-field FDD technology B installed and functional.

Baseline testing required that the installed liquid line restriction valve be set to the wide open position. Diagnostic outputs were recorded for Test 2.

Comment

*One might assume that ideally, baseline testing should be done without installing any aftermarket FDD technologies, so as to quantify unbiased, independent HVAC performance. FDD technology B is considered "non-invasive" in that its installation was not anticipated to have significant impacts on HVAC operation. FDD technology B consists of air sensors, clamp-on thermocouples, and refrigerant pressure hoses. This is not significantly different from what is used in typical laboratory instrumentation. More exploration may be done to quantify impacts, if any, of FDD technology instrumentation. This work can lead to more clarity regarding what can be considered "invasive." This is not currently considered a priority issue.*

Packaged unit air conditioners are typically designed to operate from 350 – 400 SCFM per rated ton[9]; 350 SCFM per ton was chosen for the baseline airflow rate of 1,750 SCFM. The AHRI 210/240 cooling full-load air volume rate specification of 37.5 SCFM/1000 Btu/h (450 CFM/ton) was not exceeded.

The factory-shipped refrigerant nameplate charge for the RTU was 20 lbs. Manufacturer charging charts were also available for guidance in achieving proper charge levels. These charts indicate correct charge by providing a proper outdoor coil leaving refrigerant temperature (state point R4) that must be maintained for any given compressor refrigerant discharge pressure. These charging charts were used during baseline Test 2, to ensure proper charge was maintained after various laboratory modifications were performed for subsequent testing. These modifications included additional liquid line runs, a refrigerant mass flow meter, various refrigerant pressure/temperature sensors, and a liquid line restriction valve. The proper adjusted total refrigerant charge was **23.3 lbs.**

Figure 11 maps the available fully-defined state points R1, R3, R4, and R7 on the refrigeration side on a pressure-enthalpy diagram, using measurements/calculations from baseline Tests 1-3; R7 is defined using the pressure at R1 and the enthalpy at R4. The state points are plotted around the saturation dome for R-410a. Sufficient sensors were not available to give a comprehensive view of the refrigeration cycle at all possible refrigerant-side state points. However, the available state points indicate that the baseline test results follow key anticipated trends such as:

- For Tests 1 – 3, as outdoor test chamber conditions decrease from 115°F to 80°F, high-side pressures also drop.
- For Tests 1 – 3, the R4 state points do not lie on the saturation dome; there is sub-cooled liquid in the liquid line (no mixed phase flow).
- Slightly superheated vapor is maintained at the evaporator outlet, R1, throughout Tests 1 – 3; the evaporator is successfully boiling away liquid refrigerant to provide cooling.

[9] http://www.energy.ca.gov/2008publications/CEC-400-2008-017/rev1_chapters/NRCM_Chapter_10_Acceptance_Requirements.pdf

- The condenser inlet R3 contains high pressure superheated vapor; the slightly superheated vapor from the evaporator outlet has been compressed to a higher pressure/temperature.
- The pressure-enthalpy plot of the cycle that contains R1, R3, R4, and R7, exhibits the typical shape of a vapor-compression cycle.

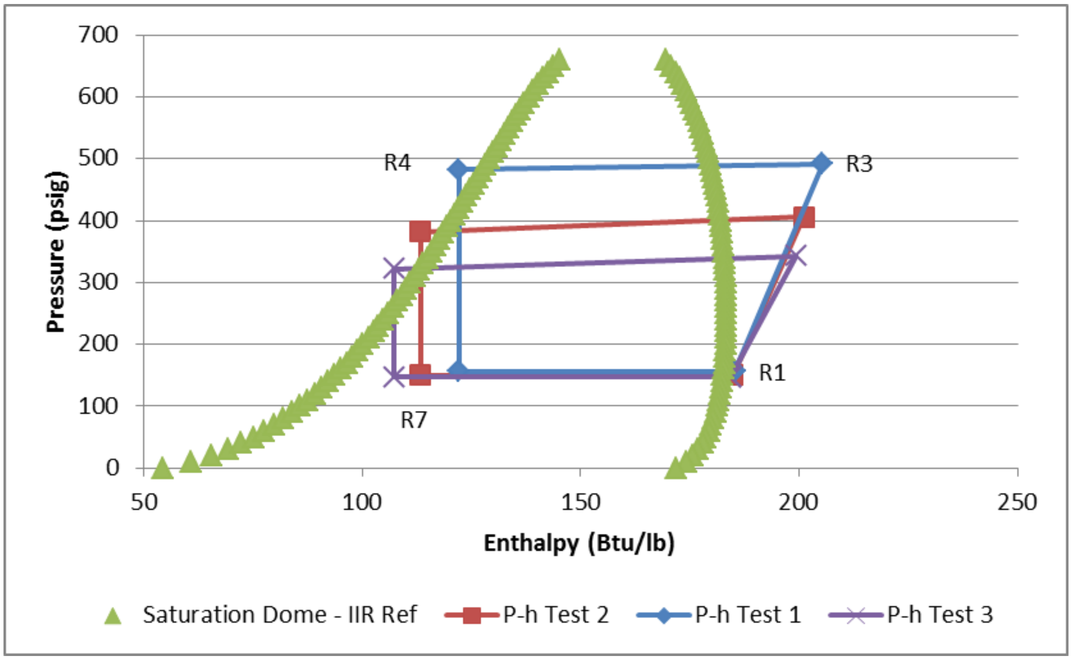


FIGURE 11. P-H DIAGRAM: BASELINE REFRIGERATION PROCESSES AT VARYING OPERATING CONDITIONS[10]

## ECONOMIZER FAULT TEST SCENARIOS

The onboard FDD has two distinct alarms that point to a variety of potential economizer faults:

- **Alarm 1:** Loss of communication with the economizer actuator - Communication wiring problem with actuator
- **Alarm 2:** Economizer Damper Fault
  - Economizer damper actuator out of calibration
  - Economizer damper actuator torque above load limit
  - Economizer damper actuator hunting excessively
  - Economizer damper stuck or jammed
  - Economizer damper actuator mechanical failure
  - Economizer damper actuator direction switch wrong

These two codes indicate two general types of economizer fault scenarios that can be tested. Table 13 lists the two economizer fault tests that were established based on

[10] IIR standard reference point for enthalpy

the economizer FDD capabilities of the onboard FDD. Testing of these economizer faults did not require control of indoor chamber and outdoor chamber conditions or any steady-state testing analogous to AHRI 210/240.

**TABLE 13. SINGLE-FAULT TEST SCENARIOS**

| TEST # | DESCRIPTION | INDOOR CHAMBER AIR CONDITION | OUTDOOR CHAMBER AIR CONDITION |
|---|---|---|---|
| 4 | Economizer Mechanical Fault | N/A | N/A |
| 5 | Economizer Communications Fault | | |

The economizer mechanical fault was imposed by physically obstructing the movement of the damper at its gears. Once obstructed, the RTU's service mode economizer test was activated. This service test mode attempted to actuate the obstructed dampers of the economizer, resulting in a fault. Removal of the damper blades was also employed as an alternative method. Both methods triggered the same damper fault alarm.

The economizer communications fault was imposed by disconnecting the wiring harness at the damper motor. Once the RTU was powered on, the onboard FDD immediately recognized an economizer communication fault.

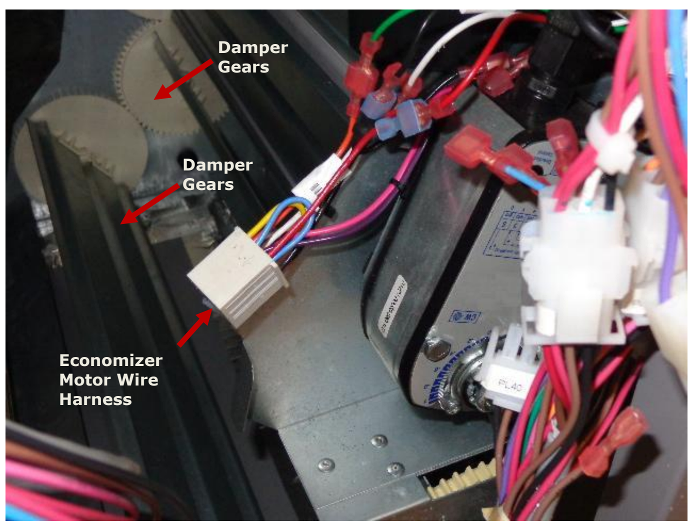


**FIGURE 12. ECONOMIZER WIRING HARNESS DISCONNECT**

# SINGLE FAULT TEST SCENARIOS

Table 14 details all single-fault tests that underwent steady-state testing.

**TABLE 14. SINGLE-FAULT TEST SCENARIOS**

| ID/OD Test Chamber Conditions -> | | | AHRI 80F/67F ID 95F OD | | | Hot & Dry 80F/67F ID 115F OD | Mild Ambient 80F/67F ID 80F OD |
|---|---|---|---|---|---|---|---|
| | | Test ID -> | Test 6 | Test 7 | Test 8 | Test 9 | Test 10 |
| Low Charge | Severity -> | None | Low | Med | Hi | Hi | Hi |
| | | 0% | -10% | -20% | -30% | -30% | -30% |
| | | 23.3 lbs | 20.9 lbs | 18.6 lbs | 16.3 lbs | 16.3 lbs | 16.3 lbs |
| | | Test ID -> | Test 11 | Test 12 | Test 13 | Test 14 | Test 15 |
| High Charge | Severity -> | None | Low | Med | Hi | Hi | Hi |
| | | 0% | 10% | 20% | 30% | 30% | 30% |
| | | 23.3 lbs | 25.6 lbs | 27.9 lbs | 30.2 lbs | 30.2 lbs | 30.2 lbs |
| | | Test ID -> | Test 16 | Test 17 | Test 18 | Test 19 | Test 20 |
| Liquid Line Restriction | Severity -> | None | Low | Med | Hi | Hi | Hi |
| | | 0% | 71% | 90% | 100% | 100% | 100% |
| | | 0 psi | 87 psi | 111 psi | 123 psi | 188 psi | 130 psi |
| | | Test ID -> | Test 21 | Test 22 | Test 23 | Test 24 | Test 25 |
| Non-Condensables | Severity -> | None | Low | Low | Hi | Hi | Hi |
| | | (0%) 1 atm $N_2$ | (36%) 1 atm $N_2$ | (64%) 1 atm $N_2$ | (100%) 1 atm $N_2$ | (100%) 1 atm $N_2$ | (100%) 1 atm $N_2$ |
| | | 0 oz. | 0.5 oz $N_2$ | 0.9 oz $N_2$ | 1.4 oz $N_2$ | 1.4 oz $N_2$ | 1.4 oz $N_2$ |
| | | Test ID -> | Test 26 | Test 27 | Test 28 | Test 29 | Test 30 |
| Evaporator Airflow Reduction | Severity -> | None | Low | Med | Hi | Hi | Hi |
| | | 0% | -23% | -46% | -67% | -67% | -67% |
| | | 1750 SCFM | 1350 SCFM | 950 SCFM | 550 SCFM | 550 SCFM | 550 SCFM |
| | | Test ID -> | Test 31 | Test 32 | Test 33 | Test 34 | Test 35 |
| Condenser Airflow Reduction | Severity -> | None | Low | Med | Hi | Hi | Hi |
| | | 0% | -16% | -37% | -58% | -36% | -65% |
| | | 3240 SCFM | 2720 SCFM | 2040 SCFM | 1350 SCFM | 2050 SCFM | 1140 SCFM |

For these single-fault testing scenarios, the strategy was to:

- Capture the effects of three incremental fault levels at a standardized condition of 80°F/67°F (DB/WB) indoor chamber, 95°F outdoor chamber.
- Capture the effects of the most pronounced fault level, at two extra outdoor test chamber air conditions.

When determining the increments of faults the following questions are asked:

- Is the fault increment reasonably representative of what has been encountered in-field?
- Does the fault increment induce a failure mode or otherwise prohibit the HVAC system from operating in a steady state fashion? Examples of an HVAC unit not working in a steady fashion include:
  - A condenser airflow reduction may be severe enough to cause HVAC system shutdown by tripping the high-pressure switch.

- A liquid line restriction can be severe enough to drop low-side pressures to a point that will cause HVAC system shutdown by tripping the low-pressure switch.
- A liquid line restriction can be severe enough to drop the evaporator temperature low enough to cause coil frosting.

When moving between test scenarios, proper measures were taken to reverse the effects of each fault to bring the HVAC system back to baseline operating conditions, as needed. Bringing the system back to baseline was not necessary after performing incremental faults in the same family/category. For example, after testing 10% charge reduction, it was not necessary to bring the system back to baseline before proceeding with a 20% charge reduction test.

Procedures for specific faults are detailed in the sections below. Tests were conducted in a manner analogous to that of the cooling mode steady-state wet-coil tests in AHRI 210/240.

### A. (Tests 6-10) Low Refrigerant Charge

The low refrigerant charge fault describes a state where an HVAC system contains refrigerant charge levels significantly below that which was intended by the manufacturer. Low charge levels may occur because of improper charging or servicing practices, or general system leakage. The HVAC system will have less working fluid available to remove heat from the conditioned space(s) and may operate with significant performance degradation.

Increments of low refrigerant charge are defined on a percent of nominal charge reduced basis (by mass). The 10% reduction scenario refers to an HVAC system that contains 90% of its nominal charge. Defined, nominal charge is the amount of refrigerant required to achieve compliance with manufacturer specifications.

The extreme test scenario may be considered to be the state right before:

- Evaporator frost forms (saturated evaporator temperatures below 32°F); or
- The HVAC system shuts down on low suction pressure. (The low-pressure cutout limit for the HVAC system’s compressor was determined to be 54 psig.)

During this test, refrigerant was recovered from the RTU as it was running. The baseline charge was **23.3 lbs.** Low charge levels were tested at **10%**, **20%**, and **30%** under nominal charge. This corresponds to total charge levels of **20.9 lbs.**, **18.6 lbs.**, and **16.3 lbs.**, for Tests 6, 7 and 8, respectively. Tests 9 and 10 capture different combinations of indoor and outdoor chamber air conditions with a total charge of **16.3 lbs.**

Comment

> *For future reference, it is suggested that anyone performing tests of this nature should always comprehensively verify that the HVAC setup is at the original manufacturer specification and should verify that correct evacuation and charging procedures have been followed such as maintaining cleanliness of evaporator and condenser surfaces, etc.*

*Mixed-phase refrigerant flow was encountered in the liquid line at all tested levels of low charge. Refrigerant-side calculations of gross cooling capacity were compromised through errors with refrigerant mass flow measurement and enthalpy calculations. The refrigerant regression model can still be used to check refrigerant mass flow and compressor power.*

### B. (Tests 11-15) High Refrigerant Charge

The high refrigerant charge fault describes a state where an HVAC system contains refrigerant charge levels significantly above the original manufacturer specifications. High levels of refrigerant charge may occur from improper charging/servicing. The HVAC system will have excessive working fluid available to remove heat from the conditioned space. As a result, the system may operate with increased high-side pressures, significant performance degradation, and may run the risk of introducing liquid refrigerant into the compressor.

Increments of high refrigerant charge are defined on a percent of nominal charge basis. For example, the 20% high charge scenario refers to an HVAC system that contains 120% of its nominal charge. Nominal charge is defined as the amount of refrigerant required to achieve manufacturer specifications.

The extreme test scenario may be considered to be the state right before:

- The liquid refrigerant is introduced into the compressor; or
- The HVAC system shuts down on high head pressure. (The high pressure cutout limit for the HVAC system's compressor was determined to be 650 psig.)

High charge faults were imposed by weighing in additional refrigerant into the RTU as it was running. The RTU had a baseline charge of **23.3 lbs.** Only new R-410a refrigerant was added to eliminate the possibility of contaminants. Since R-410a is a blend, it was essential to ensure liquid was pulled from the supply tank. If vapor was pulled from the supply tank, the constituents of the blend would have boiled away from the supply tank at different rates, thereby changing the ratio of the blend added to the RTU. R-410a was pulled from the supply tank as a liquid to avoid impacts to the blend's ratio. Testing was performed with **10%**, **20%**, and **30%** above nominal charge levels. This corresponds to total charge levels of **25.6 lbs.**, **27.9 lbs.**, and **30.2 lbs.**, respectively.

### C. (Tests 16-20) Refrigerant Liquid Line Restrictions

The refrigerant liquid line restrictions fault describes a state in which refrigerant flow is unintentionally restricted in a certain part of the liquid line. These restrictions cause unwanted pressure drops at certain points in the system. These restrictions include sources such as bent refrigerant lines, dirty liquid line filter-driers, or solder blockages at pipe joints. Restricted/clogged expansion devices may also exhibit similar impacts to the HVAC system. High levels of line restriction may result in system failure on low suction pressure or evaporator frosting.

During testing, a refrigerant liquid line restriction was simulated with a ball valve on the liquid line of the RTU, downstream of the mass flow meter, and upstream of the thermostatic expansion valve (TxV). The refrigerant pressure differential across the restriction valve was measured, using state points R4 (condenser outlet) and R6 (expansion device inlet). Increments of line restrictions were defined in set pressure drops and measured in pounds per square inch.

The extreme test scenario may be considered to be:

- The state right before the evaporator frost forms (saturated evaporator temperatures below 32°F); or
- The HVAC system shuts down on low suction pressure. (The low-pressure cutout limit for the HVAC system's compressor was determined to be 54 psig.)

Refrigerant liquid line restriction faults were initiated while the RTU was activated and running. Before restrictions were imposed, baseline pressure differential was on the order of **1 psi** to **3 psi**. Restrictions were initiated with the installed liquid line ball valve while monitoring the compressor suction pressure (state point R2) and Saturated Evaporator Temperature (SET) (state point R2). Initially, a restriction was imposed on the RTU until a SET near 35°F was achieved; 35°F was chosen as the target temperature to prevent frost formation on the evaporator. However, fine-tuned control of SET proved difficult with the ball valve; a SET of 37°F was achieved for Test 18. This restriction was deemed the extreme test scenario and subsequent restriction increments of psi drops were established based on the extreme scenario. Testing was performed with **87 psi**, **111 psi**, and **123 psi** liquid line restrictions for Tests 16, 17, and 18, respectively. These test points provided adequate resolution for mapping the range of fault impacts on the RTU. SETs of 37°F and 36°F were achieved for Test 19 and Test 20, respectively; pressure drops of **188 psi** and **130 psi** were achieved for Test 19 and Test 20, respectively.

Comment

> *Use of a ball valve is not recommended as fine-tuned adjustments proved difficult. Valve selection for liquid line restriction testing should consider those that feature more precise control and maintain low losses in the fully open position, such as a needle valve.*

### D. (Test 21-25) Non-Condensables

The refrigerant line non-condensables fault describes a state in which contaminants such as air, water vapor, or nitrogen mix with the refrigerant in an HVAC system. The physical properties of these contaminants and their subjection to the HVAC system's working pressures mean they exist as gases throughout the system. These contaminants impose their own properties on the overall working fluid, which typically results in performance degradation. Non-condensables may be introduced through faulty equipment servicing.

Refrigerant non-condensables were simulated with nitrogen gas. Increments were defined on a mass-of-nitrogen basis. A mass of nitrogen exists at a specific pressure, when introduced into an empty non-running HVAC system by itself. This pressure is directly proportional to the mass of nitrogen introduced into an empty system.

The extreme scenario may be considered as the mass of nitrogen that pressurizes to one atm, when introduced into an HVAC system by itself (no refrigerant). This represents a scenario where an empty HVAC system, exposed to atmospheric pressure, is not subjected to evacuation, and is charged with refrigerant.

Appropriate amounts of nitrogen faults were determined through pretesting. All refrigerant was recovered from the RTU. Then, nitrogen was weighed into the RTU until one atm of measured pressure was achieved. This corresponded to **1.4 ounces** (oz) of nitrogen for this specific test setup.

There are two types of methods in which non-condensables may be introduced into this RTU for testing; they are described below:

- **Method 1:** The correct nominal charge has been weighed in with an additional specified mass of nitrogen. That is to say, the mass of correct nominal charge of the system is known, and simply weighed into a system that contains non-condensables.
- **Method 2:** A specified mass of nitrogen has been added, and a non-nominal charge of refrigerant has been added. That is to say, refrigerant is incrementally added to the RTU that contains non-condensables, until the design sub-cooling value has been achieved (10°F).

Method 1 is considered representative of a singular, non-condensables fault. Method 2 is considered representative of a multiple simultaneous fault comprised of non-condensables and low refrigerant charge. As such, Method 1 was used for singular-fault testing.

For Test 21 through Test 23, the RTU underwent steady state testing with mixture iterations consisting of 23.3 lbs. of refrigerant and nitrogen levels of **0.5 oz**, **0.9 oz**, and **1.4 oz**. These nitrogen levels correspond to approximately one third, two thirds, and the entire one atm pretest amount. The mixture containing 23.3 lbs. of refrigerant and 1.4 oz of nitrogen was tested at two additional outdoor chamber air conditions (Tests 24 and 25).

Comment

> *Non-condensables tests were conducted incorrectly at the third-party laboratory and subsequently re-done at SCE's TTC. See Appendix B for details regarding TTC setup.*

## E. <u>(Tests 26-30) Evaporator Airflow Reduction</u>

The evaporator airflow reduction fault describes a state in which the HVAC system's evaporator is reduced due to factors such as airflow obstructions, dirty/fouled evaporator, dirty filters, or evaporator fan problems. Evaporator airflow reductions result in lower evaporator temperatures/pressures and significant performance degradation may result. High levels of evaporator airflow reduction may result in system failure on low suction pressure or evaporator frosting.

Evaporator airflow was measured downstream of the supply air. The evaporator airflow reduction fault was simulated by restricting airflow downstream, at the supply air outlet. Baseline evaporator airflow is set equal to the manufacturer-rated value of 1,750 SCFM. Fault increments were based on the percent reduction in evaporator airflow. For example, an HVAC unit running with 1,349 SCFM will represent a 23% evaporator airflow reduction fault scenario (where 1-[1,349/1,750] = 23%).

The extreme test scenario may be considered as the state right before:

- The evaporator frost forms (SET under 32°F); or

- The RTU shuts down on low suction pressure. (The low-pressure cutout limit for the RTU's compressor is determined to be 54 psig.)

To establish the fault increments, pretesting for the extreme condition was needed. Airflow restriction was performed while monitoring the compressor suction pressure (state point R2) and SET (state point R2). A SET near 35°F was chosen as a point before evaporator frost will occur; a SET of 36°F was achieved for Test 28. Evaporator frosting and compressor low-pressure compressor cutout did not occur for Test 28. Evaporator airflow rates of 1,349 SCFM, 951 SCFM, and 585 SCFM were used for Tests 26, 27, and 28, respectively; evaporator airflow reductions of 23%, 46%, and 67% were used for Tests 26, 27, and 28, respectively.

Test 29 and 30 captured extreme fault conditions at two additional outdoor chamber air conditions. In these scenarios, restrictions were initiated up to a point below SET of 35°F. The SET was allowed to drift down to 23°F in Test 29, and down to 29°F in Test 30. Evaporator airflow rates of 293 SCFM (83% reduction) and 573 SCFM (67% reduction) were used for Tests 29 and 30, respectively. Test 29 exhibited signs of evaporator frost. Test 30 did not exhibit signs of evaporator frost.

Comment

> *Evaporator airflow reduction tests were repeated at SCE's TTC to explore repeatability of the test method. See Appendix B for details regarding TTC setup.*

### F. <u>(Tests 31-35) Condenser Airflow Reduction</u>

The condenser airflow reduction fault describes a state in which the HVAC system's condenser encounters reduced airflow because of factors such as condenser air inlet/outlet obstructions or condenser fouling. Condenser airflow reductions result in higher refrigerant condensing temperatures/pressures and significant performance degradation may result. High levels of condenser airflow reduction may result in system failure on high head pressure.

The condenser airflow reduction faults were simulated by restricting airflow at the condenser air outlet. A duct was setup over the condenser outlet to allow for restrictions to be done via dampers downstream (see Figure 13).

The extreme scenario may be considered as the state before the RTU shuts down on high head pressure. An airflow measurement device (not shown) was located downstream to establish increments of condenser airflow reduction.

FIGURE 13. CONDENSER AIRFLOW REDUCTION

The RTU's high-pressure switch is set to trip when compressor discharge pressures reach 650 psig. Compressor discharge/condenser inlet pressures (state point R3) were monitored. The extreme fault condition was established at a condenser inlet pressure of 599 psig, where condenser airflow was measured at 1,350 SCFM, or a 58% airflow reduction (baseline condenser airflow was 3,243 SCFM). Tests 31, 32, and 33 ran at condenser airflow rates of 2,723 SCFM (16% reduction), 2,043 SCFM (37% reduction), and 1,350 SCFM (58% reduction), respectively.

Test 34 and 35 captured the extreme fault at two additional outdoor chamber air conditions. In these scenarios, condenser airflow was again restricted to a point before the high-pressure switch would trip. In Test 34, the condenser refrigerant inlet pressure was established at 601 psig. In Test 35, the condenser refrigerant inlet pressure was established at 599 psig. Tests 34 and 35 ran at condenser airflow rates of 2,049 SCFM (36% reduction) and 1,142 SCFM (65% reduction), respectively.

## MULTIPLE-FAULT TEST SCENARIOS

Multiple-fault scenarios explored three levels of imposed fault combinations, and focused on operation at standard AHRI indoor and outdoor test chamber air conditions. For all multiple-fault scenarios tested, the strategy focused on capturing the effects of three incremental fault levels at the standardized condition of 80°F/67°F (DB/WB) indoor, and 95°F outdoor. When determining increments of faults the following questions are generally asked:

- Is the fault increment representative of what happens in the field?

- Does the fault increment induce a failure mode, or otherwise prohibit the HVAC system from operating in a steady state fashion? Examples of an HVAC system not operating in a steady state fashion include:
    - A condenser airflow reduction may be severe enough to cause HVAC system shutdown by tripping the high-pressure switch.
    - A liquid line restriction may be severe enough to drop low-side pressures to a point that will cause HVAC system shutdown by tripping the low-pressure switch.
    - A liquid line restriction may be severe enough to drop the evaporator temperature low enough to cause coil frosting (transient impacts, will build up).

**Combinations of Low Refrigerant Charge, Evaporator Airflow, and Condenser Airflow Reduction**

The families of multiple-fault tests were categorized in the following manner:

- **Tests 36 - 38 (2-fault):** Evaporator airflow and condenser airflow reduction increments
- **Tests 39, 42, and 45 (2-fault):** Low refrigerant charge and evaporator airflow reductions
- **Tests 40, 43, and 46 (2-fault):** Low refrigerant charge and condenser airflow reductions
- **Tests 41, 44, and 47 (3-fault):** Low refrigerant charge, evaporator airflow, and condenser airflow reductions

Airflow was measured at the evaporator and the condenser. Evaporator and condenser airflow reductions were imposed at airflow increments equivalent to those used in the previous single-fault scenarios of evaporator and condenser airflow reduction faults. Low charge increments were imposed in the same 10% increments conducted in the single low charge fault tests: **10%**, **20%**, and **30%** low charge. Table 15 details all of the multiple-fault tests that underwent steady-state testing.

## TABLE 15. MULTIPLE-FAULT TEST SCENARIOS

| | | | | | |
|---|---|---|---|---|---|
| | | ID/OD Test Chamber Conditions -> | AHRI 80F/67F ID 95F OD | | |
| | | Test ID -> | Test 36 | Test 37 | Test 38 |
| Evaporator and Condenser Airflow Reduction | Overall Severity -> | None | Low | Med | Hi |
| | Evaporator Airflow Reduction Severity -> | 0% | -21% | -46% | -67% |
| | | 1750 SCFM | 1377 SCFM | 950 SCFM | 586 SCFM |
| | Condenser Airflow Reduction Severity -> | 0% | -16% | -37% | -58% |
| | | 3240 SCFM | 2732 SCFM | 2042 SCFM | 1361 SCFM |
| | | Test ID -> | Test 39 | Test 42 | Test 45 |
| Low Charge and Evaporator Airflow Reduction | Overall Severity -> | None | Low | Med | Hi |
| | Low Charge Severity -> | 0% | -10% | -20% | -30% |
| | | 23.3 lbs | 20.9 lbs | 18.6 lbs | 16.3 lbs |
| | Evaporator Airflow Reduction Severity -> | 0% | -23% | -45% | -67% |
| | | 1750 SCFM | 1350 SCFM | 955 SCFM | 580 SCFM |
| | | Test ID -> | Test 40 | Test 43 | Test 46 |
| Low Charge and Condenser Airflow Reduction | Overall Severity -> | None | Low | Med | Hi |
| | Low Charge Severity -> | 0% | -10% | -20% | -30% |
| | | 23.3 lbs | 20.9 lbs | 18.6 lbs | 16.3 lbs |
| | Condenser Airflow Reduction Severity -> | 0% | -17% | -37% | -58% |
| | | 3240 SCFM | 2699 SCFM | 2042 SCFM | 1350 SCFM |
| | | Test ID -> | Test 41 | Test 44 | Test 47 |
| Low Charge, Evaporator and Condenser Airflow Reduction | Overall Severity -> | None | Low | Med | Hi |
| | Low Charge Severity -> | 0% | -10% | -20% | -30% |
| | | 23.3 lbs | 20.9 lbs | 18.6 lbs | 16.3 lbs |
| | Evaporator Airflow Reduction Severity -> | 0% | -23% | -46% | -67% |
| | | 1750 SCFM | 1351 SCFM | 950 SCFM | 583 SCFM |
| | Condenser Airflow Reduction Severity -> | 0% | -17% | -37% | -58% |
| | | 3240 SCFM | 2700 SCFM | 2048 SCFM | 1347 SCFM |

# Testing Order and Procedures

Table 16 summarizes the order that all test scenarios were conducted. Care was taken to conduct test scenarios in an order that minimized test burden as much as possible. The following was considered in the test order selection:

- Baseline tests were conducted before conducting fault tests; the AHRI rating condition test was conducted first and an effort was made to match to manufacturer ratings as much as possible.
- Steps were taken to avoid excessive iterations of refrigerant charging/recovery.
- Efforts were made to minimize transitioning between outdoor chamber conditions.

**Table 16. Test Scenario Order**

| Test# | Fault | Description | Indoor Chamber Air Condition | Outdoor Chamber Air Condition |
|---|---|---|---|---|
| 2 | Baseline | | 80°F/67°F (DB/WB) | 95°F DB |
| 1 | | | | 115°F DB |
| 3 | | | | 80°F DB |
| 28 | Evaporator Airflow Reduction | High Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 27 | | Medium Severity | | |
| 26 | | Low Severity | | |
| 33 | Condenser Airflow Reduction | High Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 32 | | Medium Severity | | |
| 31 | | Low Severity | | |
| 38 | Two Multiple Faults – Evaporator and Condenser Airflow Reduction | High Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 37 | | Medium Severity | | |
| 36 | | Low Severity | | |
| 29 | Evaporator Airflow Reduction | High Severity | 80°F/67°F (DB/WB) | 115°F DB |
| 34 | Condenser Airflow Reduction | High Severity | | 115°F DB |
| 35 | Condenser Airflow Reduction | High Severity | 80°F/67°F (DB/WB) | 80°F DB |
| 30 | Evaporator Airflow Reduction | High Severity | | 80°F DB |
| 18 | Refrigerant Liquid Line Restrictions | Low Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 17 | | Medium Severity | | |
| 16 | | High Severity | | |
| 19 | | High Severity | | 115°F DB |
| 20 | | High Severity | | 80°F DB |
| 11 | High Refrigerant Charge | Low Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 12 | | Medium Severity | | |
| 13 | | High Severity | | |
| 14 | | High Severity | | 115°F DB |
| 15 | | High Severity | | 80°F DB |
| 6 | Low Refrigerant Charge | Low Severity | | |

| Test# | Fault | Description | Indoor Chamber Air Condition | Outdoor Chamber Air Condition |
|---|---|---|---|---|
| 39 | Two Multiple Faults - Low Charge and Evaporator Airflow Reduction | Low Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 41 | Three Multiple Faults - Low Charge, Evaporator and Condenser Airflow Reduction | Low Severity | | |
| 40 | Two Multiple Faults - Low Charge and Condenser Airflow Reduction | Low Severity | | |
| 7 | Low Refrigerant Charge | Medium Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 42 | Two Multiple Faults - Low Charge and Evaporator Airflow Reduction | Medium Severity | | |
| 44 | Three Multiple Faults - Low Charge, Evaporator and Condenser Airflow Reduction | Medium Severity | | |
| 43 | Two Multiple Faults - Low Charge and Condenser Airflow Reduction | Medium Severity | | |
| 8 | Low Refrigerant Charge | High Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 45 | Two Multiple Faults - Low Charge and Evaporator Airflow Reduction | High Severity | | |
| 47 | Three Multiple Faults - Low Charge, Evaporator and Condenser Airflow Reduction | High Severity | | |
| 46 | Two Multiple Faults - Low Charge and Condenser Airflow Reduction | High Severity | | |
| 9 | Low Refrigerant Charge | High Severity | 80°F/67°F (DB/WB) | 115°F DB |
| 10 | | High Severity | | 80°F DB |
| 2* | Repeat baseline | | 80°F/67°F (DB/WB) | 95°F DB |
| 21 | Non-Condensables | Low Severity | 80°F/67°F (DB/WB) | 95°F DB |
| 22 | | Medium Severity | | |
| 23 | | High Severity | | |
| 24 | | High Severity | | 115°F DB |
| 25 | | High Severity | | 80°F DB |
| 4 | Economizer Mechanical Fault | | NA | NA |
| 5 | Economizer Communications Fault | | | |

# TEST METHOD CONCLUSIONS AND RECOMMENDATIONS

This project successfully developed a steady-state test method suitable for simulating HVAC faults in a laboratory environment, but is not intended to be the final and universal solution to fully understanding FDD and HVAC maintenance. This lab test method does not capture transient impacts of faults, and cannot inform of the actual severity, incidence, and prevalence of faults experienced by equipment in the field. The overwhelming permutations of fault severities, fault combinations, indoor/outdoor conditions, and HVAC equipment characteristics make laboratory testing a potentially large burden for directly exploring FDD technologies via lab testing alone.

Industry acceptance of an FDD laboratory test method should continue to be a priority, with a clear understanding of how it fits into a combination of other diverse efforts. Ideally, field efforts, lab efforts, and simulation efforts will be cohesively orchestrated and leveraged to best understand and enhance FDD and HVAC maintenance. In this scheme, a larger variety of scenarios can be explored, in an informed, effective manner. An ideal scheme of efforts should include the following:

- Well-trained and experienced field specialists who use best practices and technologies to implement/promote quality HVAC maintenance, and inform laboratory testers and simulation experts.
- Laboratory testers who adhere to a standardized lab test method to generate and compile key data across a variety of important scenarios, and work with field specialists and simulation experts to develop, explore, and enhance technologies and best practices.
- Simulation experts who work with laboratory testers and field specialists, and leverage validated simulation/modeling techniques to explore mathematical, field, and lab-generated data to develop, explore, and enhance technologies and best practices.

An enhanced understanding of FDD and a standardization of terms and practices allows for broader adoption of reliable, accurate, cost-effective FDD methods and technologies and ultimately widespread enhancement and persistence of HVAC performance. The following activities are recommended with regards to an FDD lab test method:

- Coordinate with industry leaders through venues such as the WHPA FDD committee in a manner that is in alignment with the committee's research roadmap.
- Continue to disseminate findings and engage industry through venues such as ASHRAE, ACEEE, and WHPA FDD Committee.
- Support the efforts of ASHRAE SPC207P to ensure a lab test method is developed that generates data that is reliable, repeatable, and reasonably representative of field conditions, and helps to enhance the understanding of FDD performance and the objective distinctions of various FDD technologies.
- Use data generated by an industry-accepted lab test method to evaluate FDD technologies that are considered for adoption into utility programs, or California Statewide or Federal Codes and Standards.

- Conduct studies to characterize faults encountered in the field to inform a prioritization of lab test scenarios that should be investigated; characteristics include fault type, severity, prevalence, and incidence.
- Investigate the transient impacts of faults associated with cyclic laboratory testing; consider adoption into the lab test method based on the merits of the results.
- Investigate and enhance current mechanisms to run simulations for FDD and fault impact evaluations, based on reliable lab data generated by an industry-accepted FDD lab test method.
- Investigate the troubleshooting performance of manual diagnostics, by both certified and non-certified technicians, with and without the assistance of FDD technologies
- Investigate the variances in fault impacts and FDD performance across key equipment characteristics/configurations, such as (not limited to) refrigerant types, heat exchanger types, expansion device types.

# APPENDIX A: CALCULATION METHODS MATRIX

TABLE 17. SUMMARY: APPLICABLE CALCULATION METHODS PER TEST SCENARIO

| | Parameter -> | Gross Cooling Capacity | | | Refrigerant Mass Flow | |
|---|---|---|---|---|---|---|
| | Measurement Type -> | Refrigerant | Refrigerant | Air | Refrigerant | Refrigerant |
| | Calculation Method -> | Enthalpy Method | Compressor Regression | Enthalpy Method | Measurement | Compressor Regression |
| 1 | Base | Y | Y | Y | Y | Y |
| 2 | | Y | Y | Y | Y | Y |
| 3 | | Y | Y | Y | Y | Y |
| 4 | Economizer Mechanical Fault | N/A | N/A | N/A | N/A | N/A |
| 5 | Economizer Communications Fault | N/A | N/A | N/A | N/A | N/A |
| 6 | Low Charge | N | N | Y | N | Y |
| 7 | | N | N | Y | N | Y |
| 8 | | N | N | Y | N | Y |
| 9 | | N | N | Y | N | Y |
| 10 | | N | N | Y | N | Y |
| 11 | High Charge | Y | Y | Y | Y | Y |
| 12 | | Y | Y | Y | Y | Y |
| 13 | | Y | Y | Y | Y | Y |
| 14 | | Y | Y | Y | Y | Y |
| 15 | | Y | Y | Y | Y | Y |
| 16 | Line Restrictions | Y | Y | Y | Y | Y |
| 17 | | Y | Y | Y | Y | Y |
| 18 | | Y | Y | Y | Y | Y |
| 19 | | Y | Y | Y | Y | Y |
| 20 | | Y | Y | Y | Y | Y |
| 21 | Non-condensables | N | N | Y | N | N |
| 22 | | N | N | Y | N | N |
| 23 | | N | N | Y | N | N |
| 24 | | N | N | Y | N | N |
| 25 | | N | N | Y | N | N |
| 26 | Evaporator Airflow Reduction | Y | Y | Y | Y | Y |
| 27 | | Y | Y | Y | Y | Y |
| 28 | | Y | Y | Y | Y | Y |
| 29 | | Y | Y | Y | Y | Y |
| 30 | | Y | Y | Y | Y | Y |

| | Parameter -> | Gross Cooling Capacity | | | Refrigerant Mass Flow | |
|---|---|---|---|---|---|---|
| | Measurement Type -> | Refrigerant | Refrigerant | Air | Refrigerant | Refrigerant |
| | Calculation Method -> | Enthalpy Method | Compressor Regression | Enthalpy Method | Measurement | Compressor Regression |
| 31 | Condenser Airflow Reduction | Y | Y | Y | Y | Y |
| 32 | | Y | Y | Y | Y | Y |
| 33 | | Y | Y | Y | Y | Y |
| 34 | | Y | Y | Y | Y | Y |
| 35 | | Y | Y | Y | Y | Y |
| 36 | Evaporator and Condenser Airflow Reduction | Y | Y | Y | Y | Y |
| 37 | | Y | Y | Y | Y | Y |
| 38 | | Y | Y | Y | Y | Y |
| 39 | Low Charge & Evaporator Airflow Reduction | N | N | Y | N | Y |
| 42 | | N | N | Y | N | Y |
| 45 | | N | N | Y | N | Y |
| 40 | Low Charge & Condenser Airflow Reduction | N | N | Y | N | Y |
| 43 | | N | N | Y | N | Y |
| 46 | | N | N | Y | N | Y |
| 41 | Low Charge, Evaporator and Condenser Airflow Reduction | N | N | Y | N | Y |
| 44 | | N | N | Y | N | Y |
| 47 | | N | N | Y | N | Y |

# APPENDIX B: TTC SUPPLEMENTAL TESTING

Several test scenarios were chosen to be conducted at the TTC to ensure the validity of test data. Table 18 and Table 19 summarize the test scenarios that were chosen for TTC supplemental testing. Single fault evaporator airflow reduction tests (Tests 26 – 30) were chosen to show repeatability.

The notable differences in the test setups are summarized as follows:

- The third-party lab uses ductwork on the supply of the RTU, whereas TTC implements ductwork on both return and supply. (This is optional as per Federal test method/AHRI 210/240.)
- The third-party lab measures airflow downstream of the RTU, on the supply air whereas, TTC measures airflow upstream of the RTU, on the return air.
- The third-party lab uses tube sampling devices for average DB and WB air temperature measurement, whereas TTC uses the average measurements of thermocouples placed directly in the airstream for DB, and tube sampling devices for humidity measurement.
- Both the TTC and third-party test setups use a two-chamber configuration, but the third-party lab setup uses a single room with a partition, whereas the TTC uses two physical test chambers.
- The third-party lab setup uses a longer liquid line extension for its instrumentation than the TTC setup, which impacts the adjusted nominal charge; the nominal charge of the TTC setup is 21.3 lbs., the nominal charge of the third-party lab setup is 23.3 lbs.

**TABLE 18. TTC SUPPLEMENTAL TESTING: BASELINE TEST SCENARIOS**

| TEST # | DESCRIPTION | INDOOR CHAMBER AIR CONDITION | OUTDOOR CHAMBER AIR CONDITION |
|---|---|---|---|
| 1 | Baseline | 80°F/67°F (DB/WB) | 115°F DB |
| 2 | | | 95°F DB |
| 3 | | | 80°F DB |

**TABLE 19. TTC SUPPLEMENTAL TESTING: SINGLE-FAULT TEST SCENARIOS**

| TEST # | Fault | DESCRIPTION | INDOOR CHAMBER AIR CONDITION | OUTDOOR CHAMBER AIR CONDITION |
|---|---|---|---|---|
| 21 | Non-Condensables | Low Severity – 0.5 oz $N_2$ | 80°F/67°F (DB/WB) | 95°F DB |
| 22 | | Medium Severity – 0.9 oz $N_2$ | | |
| 23 | | High Severity – 1.4 oz $N_2$ | | |
| 24 | | High Severity – 1.4 oz $N_2$ | | 115°F DB |
| 25 | | High Severity – 1.4 oz N2 | | 80°F DB |

| Test # | Fault | Description | Indoor Chamber Air Condition | Outdoor Chamber Air Condition |
|---|---|---|---|---|
| 26 | Evaporator Airflow Reduction | Low Severity – 23% under nominal evaporator airflow | 80°F/67°F (DB/WB) | 95°F DB |
| 27 | | Medium Severity – 46% under nominal evaporator airflow | | |
| 28 | | High Severity – 67% under nominal evaporator airflow | | |
| 29 | | High Severity – 83% under nominal evaporator airflow | | 115°F DB |
| 30 | | High Severity – 67% under nominal evaporator airflow | | 80°F DB |

Table 20 summarizes the measurements and calculations for the TTC supplemental test setup. Figure 14 through Figure 20 illustrate the instrumentation setup for air-side and refrigerant-side state points. Air-side state points are indicated in blue and refrigerant-side state points are indicated in red; Air-side and refrigerant-side measurements are identified by their numbering per Table 20. State points, test procedures, and calculation methods of the TTC setup are held consistent with those of the third-party lab setup.

**Table 20. TTC Supplemental Test Setup: Measurements and Calculation List**

| Test # | Description | Units | Side? | State Point | Meas/Calc? |
|---|---|---|---|---|---|
| 1 | Refrigerant-side gross cooling capacity | Btu/h | Refrigerant -side | N/A | Calculation |
| 2 | Evaporator outlet - refrigerant enthalpy | Btu/lb. | | R1 | |
| 3 | Evaporator outlet - refrigerant superheat | F | | R1 | |
| 4 | Compressor suction - refrigerant saturated evaporator temperature (SET) | F | | R2 | |
| 5 | Compressor discharge - refrigerant saturated condensing temperature (SCT) | F | | R3 | |
| 6 | Condenser outlet - refrigerant sub-cooling | F | | R4 | |
| 7 | TXV inlet - refrigerant enthalpy | Btu/lb. | | R6 | |
| 8 | Evaporator outlet - refrigerant temperature | F | Refrigerant -side | R1 | Measurement |
| 9 | Evaporator outlet - refrigerant pressure | psig | | | |
| 10 | Compressor inlet - refrigerant temperature | F | | R2 | |
| 11 | Compressor inlet - refrigerant pressure | psig | | | |
| 12 | Compressor Discharge/Condenser inlet - refrigerant temperature | F | | R3 | |
| 13 | Compressor Discharge/Condenser inlet - refrigerant pressure | psig | | | |

| | | | | | |
|---|---|---|---|---|---|
| 14 | Condenser outlet - refrigerant temperature | F | | R4 | |
| 15 | Condensing unit outlet - refrigerant pressure | psig | | | |
| 16 | Mass flow meter outlet - refrigerant temperature | F | | | |
| 17 | Mass flow meter outlet - refrigerant pressure | psig | | R5 | |
| 18 | Refrigerant mass flow | lb./min | | | |
| 19 | TXV inlet - refrigerant temperature | F | | R6 | |
| 20 | TXV inlet - refrigerant pressure | psig | | | |
| 21 | Air-side net cooling capacity | SCFM | | | |
| 22 | Air mass flow | lb./min | | N/A | |
| 23 | Evaporator volumetric airflow | SCFM | | | |
| 24 | Return air - enthalpy | Btu/lb. | | A1 | |
| 25 | Return air - grid average DB temperature | F | | | |
| 26 | Supply air - enthalpy | F | Air-side | A5 | Calculation |
| 27 | Supply air - grid average DB temperature | F | | | |
| 28 | Condenser inlet air - side #1 average temperature | F | | A6 | |
| 29 | Condenser inlet air - side #2 average temperature | F | | A7 | |
| 30 | Condenser inlet air - side #3 average temperature | F | | A8 | |
| 31 | Condenser inlet air - **total average temperature** | F | | A6, A7, & A8 | |
| 32 | Return air - DB temperature #1 (upstream of return pressure taps) | F | | | |
| 33 | Return air - DB temperature #2 (upstream of return pressure taps) | F | | | |
| 34 | Return air - DB temperature #3 (upstream of return pressure taps) | F | | | |
| 35 | Return air - DB temperature #4 (upstream of return pressure taps) | F | | | |
| 36 | Return air - DB temperature #5 (upstream of return pressure taps) | F | | | |
| 37 | Return air - DB temperature #6 (upstream of return pressure taps) | F | | A1 | |
| 38 | Return air - DB temperature #7 (upstream of return pressure taps) | F | Air-side | | Measurement |
| 39 | Return air - DB temperature #8 (upstream of return pressure taps) | F | | | |
| 40 | Return air - DB temperature #9 (upstream of return pressure taps) | F | | | |
| 41 | Return air - Tw temperature (upstream of DB grid) | F | | | |
| | Return air - Td temperature (upstream of DB grid) | F | | | |
| 42 | Air pressure taps across RTU**(downstream of DB grids) #1** | in $H_2O$ | | | |

| | | | | | |
|---|---|---|---|---|---|
| 43 | Air pressure taps across RTU**(downstream of DB grids) #2** | in $H_2O$ | | A1 - A5 | |
| 44 | Air pressure taps across RTU**(downstream of DB grids) #3** | in $H_2O$ | | | |
| 45 | Air pressure taps across RTU**(downstream of DB grids) #4** | in $H_2O$ | | | |
| 46 | Outside air inlet - DB temperature | F | | A2 | |
| 47 | Mixed air - DB temperature | F | | A3 | |
| 48 | Mixed air - **Rh** temperature (upstream of DB) | F | | | |
| 49 | Evaporator fan inlet - DB temperature | F | | A4 | |
| 50 | Evaporator fan inlet - **Rh** temperature (upstream of DB) | F | | | |
| 51 | Supply air - DB temperature #1 (upstream of supply pressure taps) | F | | A5 | |
| 52 | Supply air - DB temperature #2 (upstream of supply pressure taps) | F | | | |
| 53 | Supply air - DB temperature #3 (upstream of supply pressure taps) | F | | | |
| 54 | Supply air - DB temperature #4 (upstream of supply pressure taps) | F | | | |
| 55 | Supply air - DB temperature #5 (upstream of supply pressure taps) | F | | | |
| 56 | Supply air - DB temperature #6 (upstream of supply pressure taps) | F | | | |
| 57 | Supply air - DB temperature #7 (upstream of supply pressure taps) | F | | | |
| 58 | Supply air - DB temperature #8 (upstream of supply pressure taps) | F | | | |
| 59 | Supply air - DB temperature #9 (upstream of supply pressure taps) | F | | | |
| 60 | Supply air - Td temperature (upstream of DB) | F | | | |
| 61 | Condenser inlet air - side #1-W, DB temperature #1 | F | | A6 | |
| 62 | Condenser inlet air - side #1=W, DB temperature #2 | F | | | |
| 63 | Condenser inlet air - side #1-W, DB temperature #3 | F | | | |
| 64 | Condenser inlet air - side #1-W, DB temperature #4 | F | | | |
| 65 | Condenser inlet air - side #2-S, DB temperature #1 | F | | A7 | |
| 66 | Condenser inlet air - side #2-S, DB temperature #2 | F | | | |
| 67 | Condenser inlet air - side #2-S, DB temperature #3 | F | | | |
| 68 | Condenser inlet air - side #2-S, DB temperature #4 | F | | | |
| 69 | Condenser inlet air - side #3-E, DB temperature #1 | F | | A8 | |
| 70 | Condenser inlet air - side #3-E, DB temperature #2 | F | | | |
| 71 | Condenser inlet air - side #3-E, DB temperature #3 | F | | | |
| 72 | Condenser inlet air - side #3-E, DB temperature #4 | F | | | |
| 73 | ASHRAE airflow measurement device nozzle Delta-P #1 | psig | | N/A | |

| | | | | | |
|---|---|---|---|---|---|
| 74 | ASHRAE airflow measurement device nozzle Delta-P #2 | psig | | N/A | |
| 75 | Total RTU power | Watts | Electrical | N/A | Measurement |
| 76 | Total RTU voltage | Volts | | N/A | |
| 77 | Total RTU amperage | Amps | | N/A | |
| 78 | Total RTU Frequency | Hz | | N/A | |
| 79 | Compressor power | Watts | | N/A | |
| 80 | Evaporator fan power | Watts | | N/A | |
| 81 | Condenser fan power | Watts | | N/A | |
| 82 | Condensate collected | lbs. | Other | N/A | Measurement |

**TABLE 21. ACCURACY, CALIBRATION DATES AND LOCATIONS, AND CORRESPONDING KEY MONITORING POINTS FOR SENSORS USED**

| SENSOR TYPE | MAKE/MODEL | ACCURACY (NIST TRACEABLE) | CALIBRATION DATE (LOCATION) | MONITORING POINTS DESCRIPTION |
|---|---|---|---|---|
| Temperature (type-T thermocouples) | Masy Systems, Ultra-Premium Probe | ± 0.18°C [at 0°C] (± 0.32°F) | 5-4-2011 (In-house) | Evap fan inlet DB<br>Evap coil inlet DB<br>Evap coil outlet DB<br>Outdoor chamber DB<br>Cond inlet DB<br>Cond outlet DB<br>All refrigerant temps |
| Relative Humidity (RH) | Vaisala, HMP 233 | ± 1% (0-90% RH)<br>± 2% (90-100% RH) | 5-5-2011 (SCE's Metrology Lab) | Evap outlet |
| Wet Bulb | Vaisala, HMP 247 | ± 0.013% of reading | 5-9-2011 (SCE's Metrology Lab) | Evap fan inlet |
| Relative Humidity (RH) | Vaisala, HMP 247 | ± (0.5 + 2.5% of reading)% RH | 5-9-2011 (SCE's Metrology Lab) | Supply duct |
| Dew Point | Edgetech, Dew Prime DF Dew Point Hygrometer | ± 0.2°C (± 0.36°F) | 5-5-2011 (SCE's Metrology Lab) | Evap inlet<br>Evap outlet |
| Pressure (0-1000 psig) | Setra, C207 | ± 0.13% of full scale | 4-14-2011 (In-house) | Comp discharge<br>Cond outlet<br>MFM inlet<br>TXV inlet |

| Sensor Type | Make/Model | Accuracy (NIST Traceable) | Calibration Date (location) | Monitoring Points Description |
|---|---|---|---|---|
| Pressure (0-500 psig) | Setra, C207 | ± 0.13% of full scale | 4-14-2011 (In-house) | Comp suction<br>Evap outlet |
| Pressure (0-10 inches of water, in-wg) | Ashcroft, AQS-28304 | ± 0.06% of full scale | 4-14-2011 (Tektronix Calibration Lab) | Across indoor unit |
| Power | Ohio Semitronics, GW5-002C | ± 0.2% of reading<br>± 0.04% of full scale<br>(cond: 1,000W FS)<br>(comp: 5,000W FS) | 5-11-2011 (In-house) | Condensing unit<br>Compressor<br>Condenser fan |
| Power | HIOKI 3169-21 | ± 0.5% of reading | 5-10-11 (In-house) | Indoor unit<br>Evap fan |
| Refrigerant Mass Flow Meter | Endress-Hauser, (Coriolis meter) 80F08-AFTSAAACB4AA | For liquids, ± 0.15% of reading<br>For gases, ± 0.35% of reading | 7-22-2010 (Homer R. Dulin Co.) | Refrigerant flow rate |
| Scale | HP-30K | ± 0.1 gram (± 0.0035 ounces) | 11-29-2010 (In-house) | Mass of condensate |

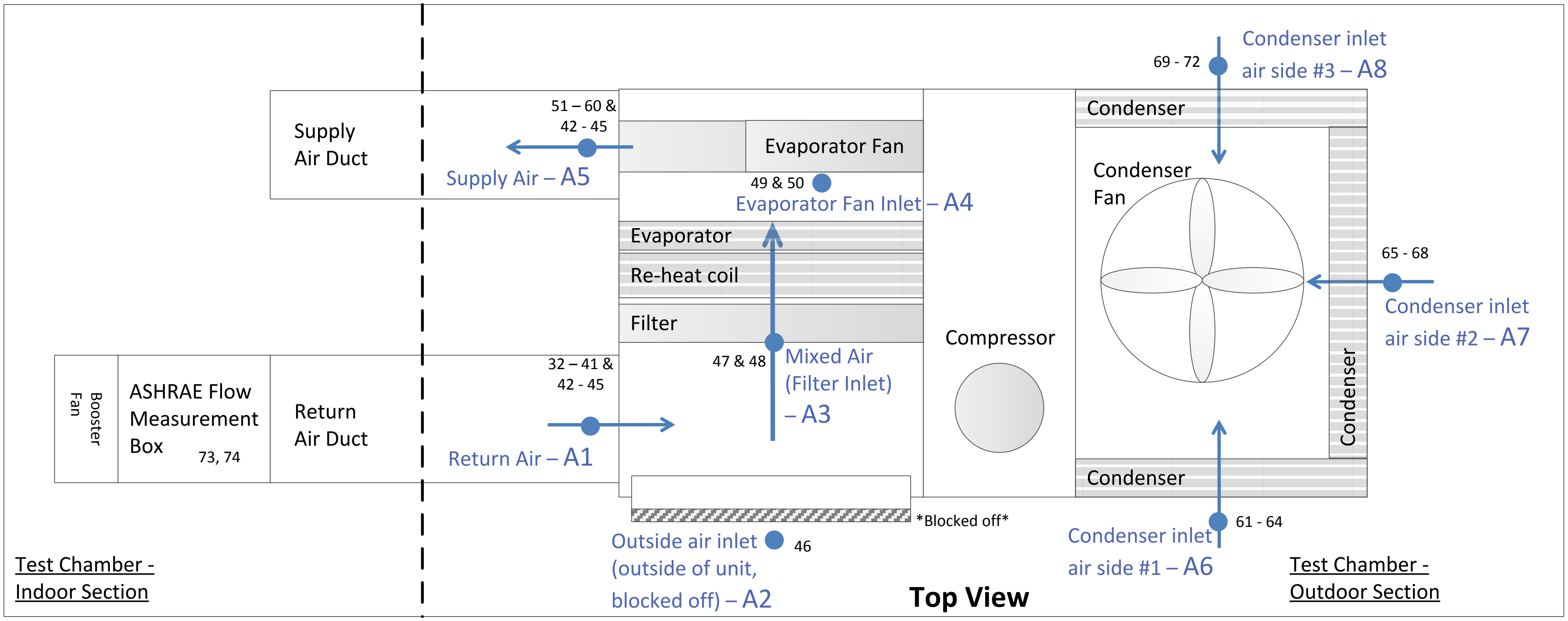


FIGURE 14. TTC SUPPLEMENTAL TEST SETUP: AIR-SIDE SENSOR DIAGRAM--TOP VIEW

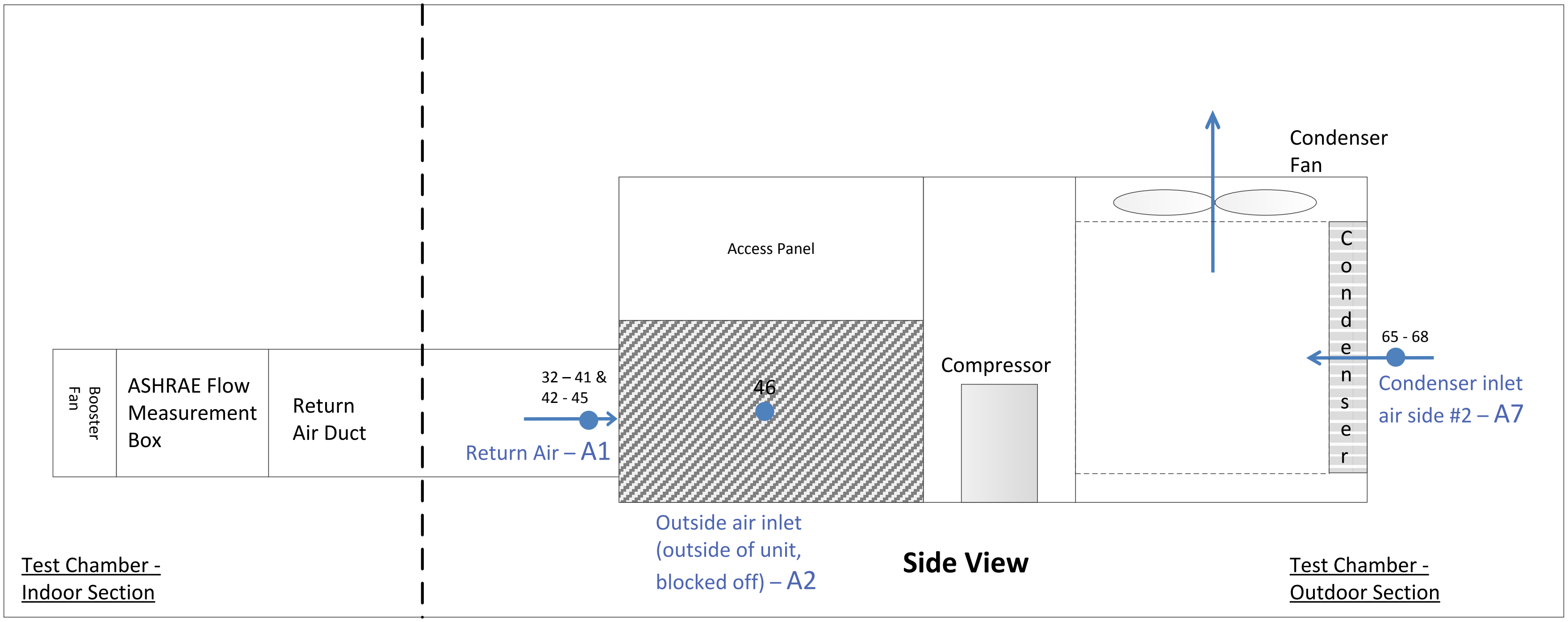


**FIGURE 15. TTC SUPPLEMENTAL TEST SETUP: AIR-SIDE SENSOR DIAGRAM--SIDE VIEW**

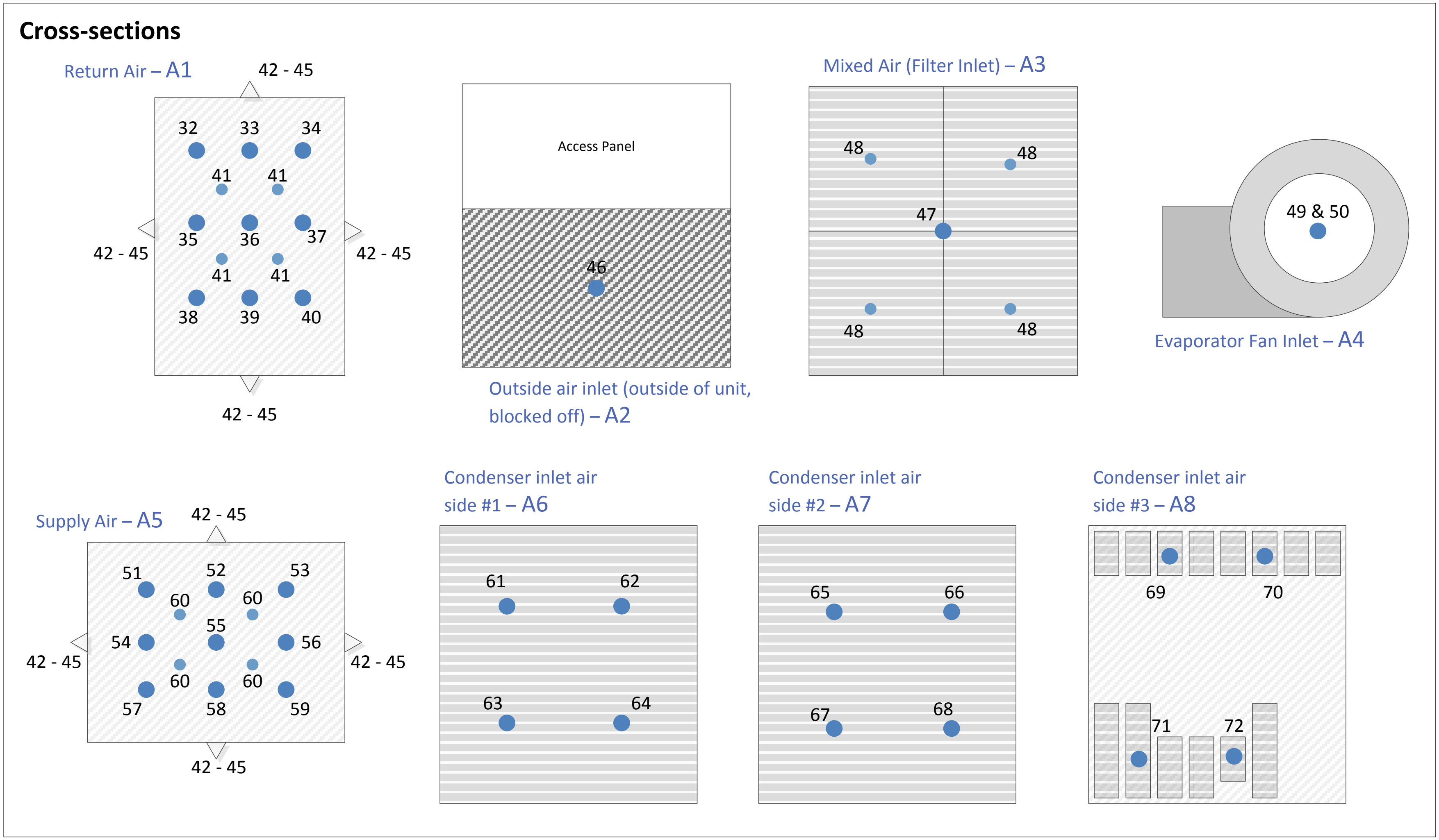


**FIGURE 16. TTC SUPPLEMENTAL TEST SETUP AIR-SIDE SENSOR DIAGRAM--CROSS-SECTIONAL VIEW**

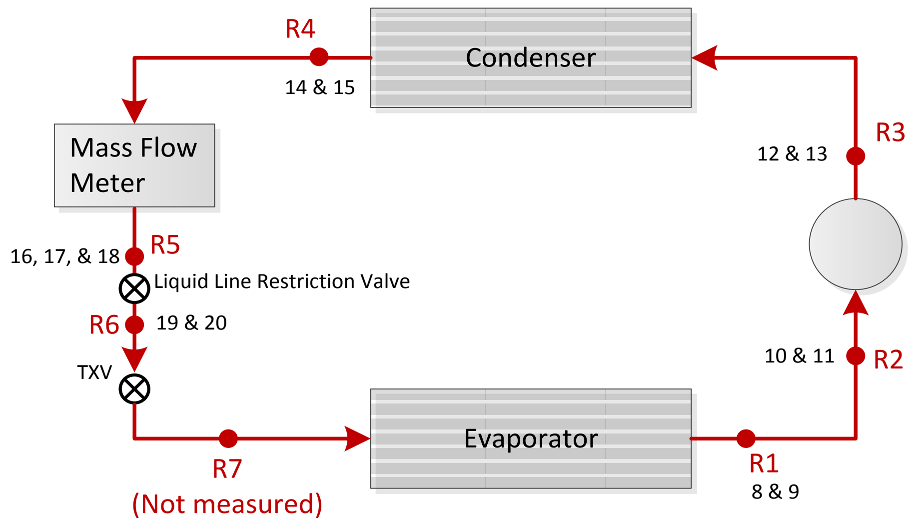


**FIGURE 17. TTC SUPPLEMENTAL TEST SETUP: REFRIGERANT-SIDE SENSOR DIAGRAM--BASIC VIEW**

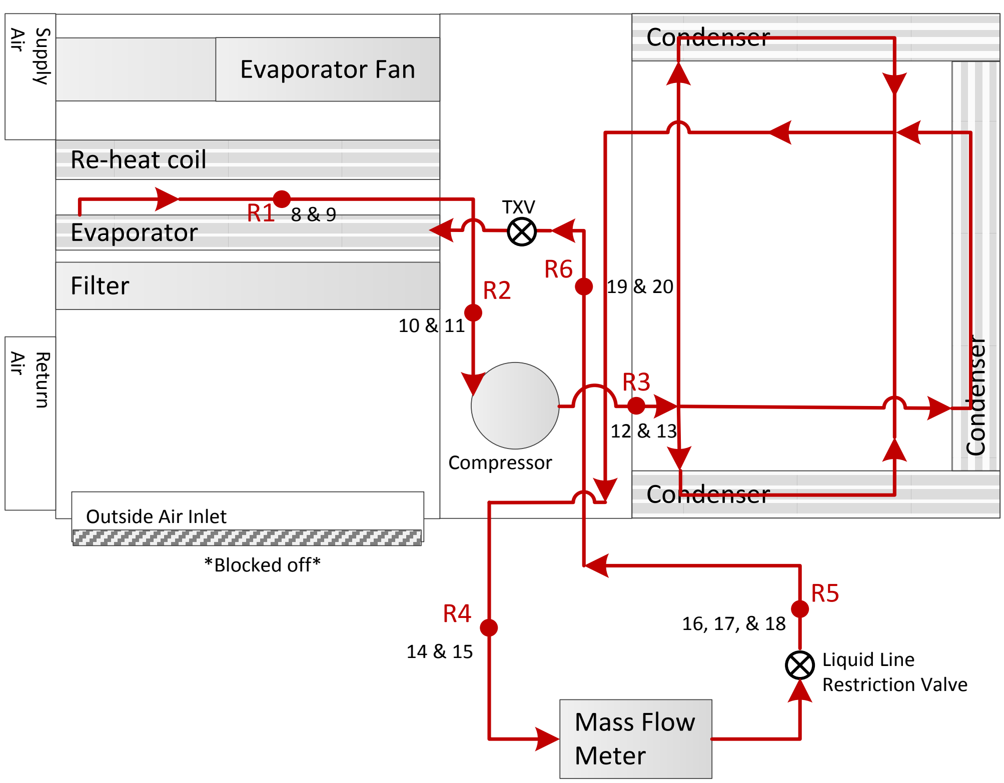


**FIGURE 18. TTC SUPPLEMENTAL TEST SETUP: REFRIGERANT-SIDE SENSOR DIAGRAM--DETAILED VIEW**

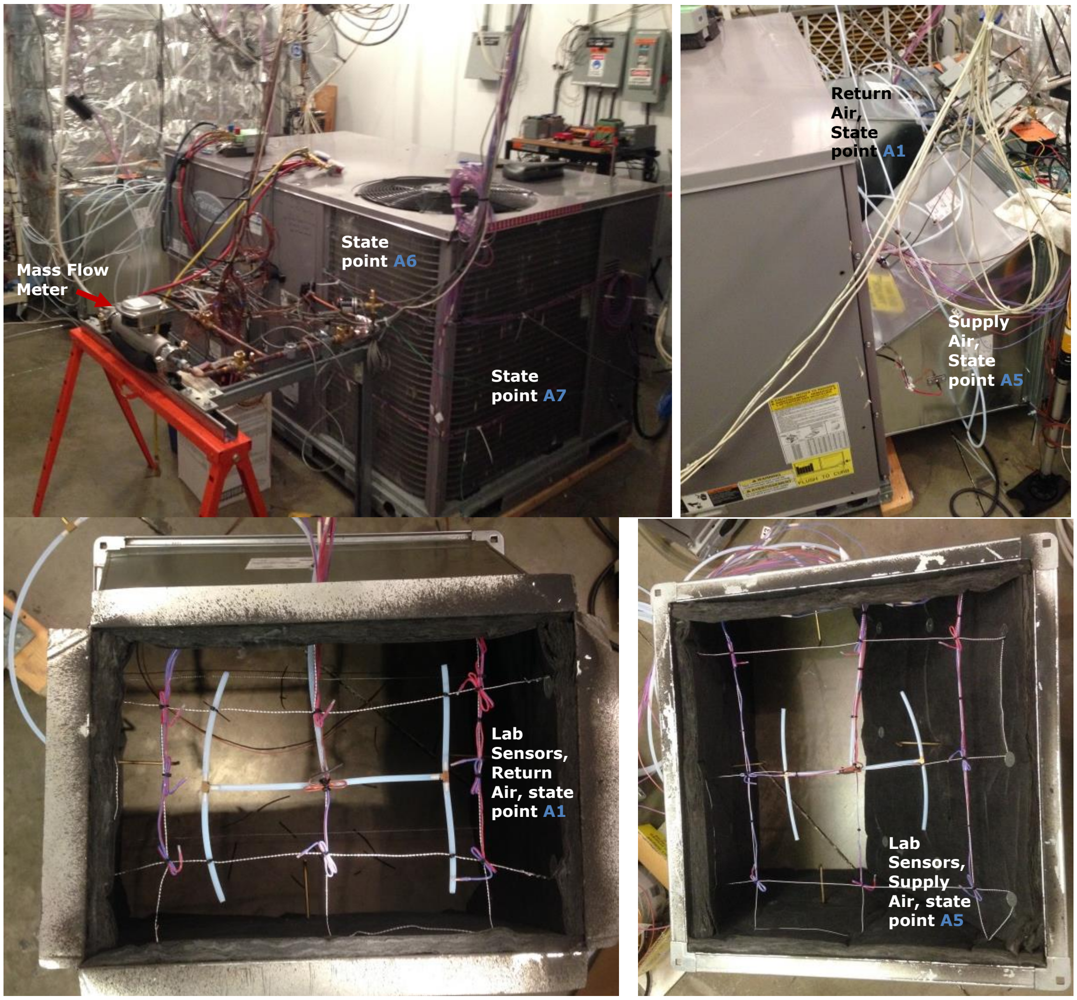


FIGURE 19. TTC SUPPLEMENTAL TEST SETUP

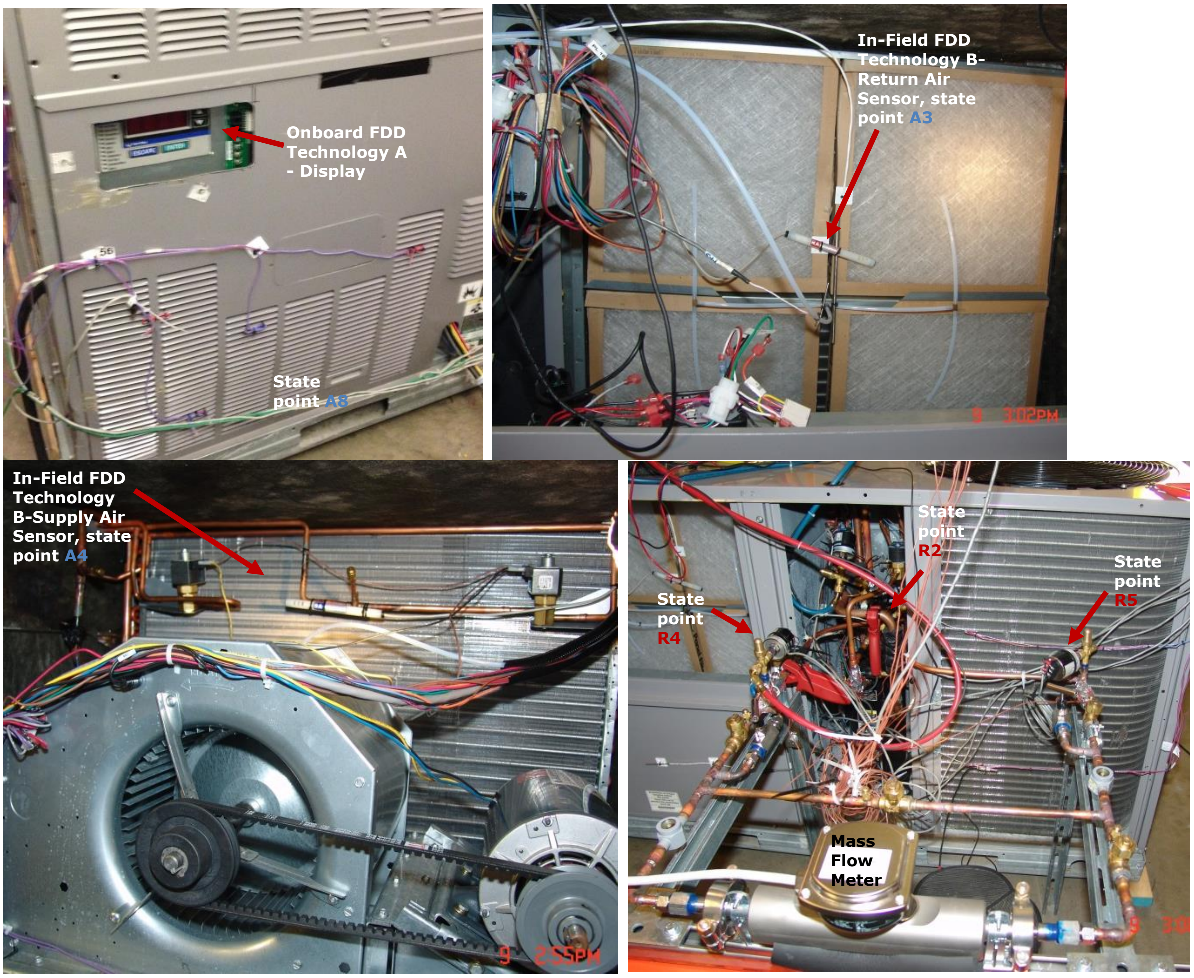


FIGURE 20. TTC SUPPLEMENTAL TEST SETUP 2

# CALCULATIONS

Various calculation methods are available for TTC laboratory testing. Table 22 lists the calculation methods used in this project. A comprehensive summary of calculation methods applicable to a given test scenario may be found in Appendix A, in Table 17.

TABLE 22. CALCULATION METHODS

| # | CALCULATION METHODS | CALCULATED PARAMETERS |
|---|---|---|
| 1 | Refrigerant-side measurements and calculations | Enthalpies, saturated temperatures, gross cooling capacity, EER |
| 2 | Compressor regression -> refrigerant-side measurements and calculations | Gross cooling capacity, refrigerant mass flow, compressor power |
| 3 | Air-side measurements and calculations | Enthalpies, net cooling capacity, EER |

**Percent difference** is defined as the difference between two values, divided by the average of the data set. This data set may comprise the two values, or it may comprise several other values. For the purposes of this project, percent difference is used when comparing different methods of calculations of a certain parameter. Percent difference is given by the following equation:

**EQUATION 9. CALCULATING PERCENT DIFFERENCE**

$$\%\,Difference = \frac{Value_1 - Value_2}{\frac{1}{2}(Value_1 + Value_2)} \times 100\%$$

**Percent change** is defined as the relative shift in a parameter, or the change of two values divided by one original value. Percent change is used when comparing a parameter from one fault test scenario, to its baseline scenario (shift in a parameter due to a fault). The following equation provides the percent change.

**EQUATION 10. CALCULATING PERCENT CHANGE**

$$\%\,Change = \frac{Value_1 - Value_2}{Value_1} \times 100\%$$

Energy Efficiency Ratio (EER) calculations are performed as follows:

**EQUATION 11. GROSS ENERGY EFFICIENCY RATIO**

$$EER_R = \frac{\dot{Q}_R}{P}$$

Or

$$EER_A = \frac{\dot{Q}_{A-G}}{P}$$

Where:

| | |
|---|---|
| $EER_R$ | = Energy Efficiency Ratio (refrigerant-side-based), Btu/hr/Watt (W) |
| $EER_A$ | = Energy efficiency ratio (air-side-based), Btu/hr/W |
| $\dot{Q}_R$ | = Refrigerant-side gross cooling capacity, Btu/hr |
| $\dot{Q}_{A-G}$ | = Air-side gross cooling capacity, Btu/hr |
| $P$ | = Total power (compressor + fans + misc.), W |

Refrigerant-side calculations for gross cooling capacity are performed as follows:

**EQUATION 12. REFRIGERANT-SIDE GROSS COOLING CAPACITY**

$$\dot{Q}_R = \dot{m}_R \times (h_{R1} - h_{R7})$$

$$h_{R7} = h_{R4}$$

Where

$\dot{Q}_{R-C}$ = Refrigerant-side gross cooling capacity, Btu/hr

$\dot{m}_R$ = Refrigerant mass flow rate, lbs. /hr

$h_{R1}$ = Enthalpy at refrigerant-side state point R1, Btu/lb

$h_{R7}$ = Enthalpy at refrigerant-side state point R7, Btu/lb

$h_{R4}$ = Enthalpy at refrigerant-side state point R4, Btu/lb

In addition, the HVAC unit's compressor manufacturer provided compressor regression curves, able to output cooling capacity, refrigerant mass flow rate, and compressor power. Saturated evaporating temperatures and condensing temperatures, based on pressures measured at state points R2 and R3, respectively, were used to generate data. This data was used as a reference point to establish confidence in existing measurements/calculations for baseline tests 1, 2, and 3.

Compressor regression outputs and test measurements/calculations, along with the associated percent differences (all rounded to the nearest one) are presented in Table 8, Table 9, and Table 10. Percent differences between refrigerant-side calculations and compressor regressions for gross cooling capacity ranged from -2% to -7%. Percent differences for compressor power ranged from 0% to 4%. Percent differences for refrigerant mass flow ranged from 2% to 4%.

**TABLE 23. BASELINE GROSS COOLING CAPACITY: REFRIGERANT-SIDE VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - GROSS COOLING CAPACITY: REFRIGERANT-SIDE (BTU/HR) | VALUE 2 - GROSS COOLING CAPACITY: COMPRESSOR REGRESSIONS (BTU/HR) |
|---|---|---|---|
| 1 | -4% | 56,087 | 59,811 |
| 2 | -3% | 62,659 | 64,573 |
| 3 | 0% | 67,153 | 67,251 |

**TABLE 24. BASELINE COMPRESSOR POWER: MEASURED VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - COMPRESSOR POWER: MEASURED (W) | VALUE 2 - COMPRESSOR POWER: COMPRESSOR REGRESSIONS (W) |
|---|---|---|---|
| 1 | 3% | 4,885 | 4,753 |
| 2 | 1% | 3,823 | 3,785 |
| 3 | 0% | 3,141 | 3,149 |

**TABLE 25. BASELINE REFRIGERANT MASS FLOW: REFRIGERANT-SIDE VS. COMPRESSOR REGRESSIONS**

| TEST # | % DIFFERENCE | VALUE 1 - REFRIGERANT MASS FLOW: MEASURED (LBS./MIN) | VALUE 2 - REFRIGERANT MASS FLOW: COMPRESSOR REGRESSIONS (LBS./MIN) |
|---|---|---|---|
| 1 | -2% | 873 | 892 |
| 2 | -1% | 859 | 864 |
| 3 | 1% | 838 | 833 |

It is important to note that refrigerant-side and compressor regression calculation issues exist for any tests featuring low refrigerant charge or non-condensables. Low charge tests yield mixed-phase refrigerant flow in the liquid line. With mixed-phase liquid line refrigerant flow, refrigerant properties look-ups become inaccurate and refrigerant mass flow measurements are compromised. In addition, while the regression model may still be suitable for predicting refrigerant mass flow and compressor power, any gross cooling capacity outputs are likely suspect.

For tests with non-condensables, refrigerant mass flow measurements are compromised, and refrigerant properties look-ups for all refrigerant-side state points are no longer applicable. The relationships between system pressures and properties changes when pure R-410a is not present, and the mixture of nitrogen vapor and liquid refrigerant likely yields inaccurate refrigerant liquid line mass flow measurements.

Air-side calculations are performed as follows:

**EQUATION 13. AIR-SIDE GROSS COOLING CAPACITY**

$$\dot{Q}_{A-G} = \dot{Q}_{A-N} + \dot{Q}_{EVAP\ FAN}$$

Where

$\dot{Q}_{A-G}$ = Air-side gross cooling capacity, Btu/hr

$\dot{Q}_{A-N}$ = Air-side net cooling capacity, Btu/hr

$\dot{Q}_{EVAP\ FAN}$ = Evaporator fan heat, Btu/hr

**EQUATION 14. EVAPORATOR FAN HEAT**

$$\dot{Q}_{EVAP\ FAN} = P_{EVAP\ FAN} \times C_1$$

Where

$P_{EVAP\ FAN}$ = Measured Evaporator Fan Power, W

$C_1$ = 3.41214163, Conversion Factor, Btu/hr/W

**EQUATION 15. AIR-SIDE NET COOLING CAPACITY**

$$\dot{Q}_{A-N} = \dot{m}_A \times (h_{A1} - h_{A6})$$

Where

$\dot{Q}_{A-N}$ = Air-side net cooling capacity, Btu/hr

$\dot{m}_A$ = Indoor air mass flow rate, lbs./hr

$h_{A1}$ = Enthalpy at air-side state point A1, Btu/lb

$h_{A6}$ = Enthalpy at air-side state point A6, Btu/lb

**EQUATION 16. INDOOR AIR MASS FLOW RATE**

$$\dot{m}_A = \dot{V}_S \times \rho_S \times C_2$$

Where:

$\dot{m}_A$ = Indoor air mass flow rate, lbs./hr

$\dot{V}_S$ = Indoor air volumetric flow rate, "standard" conditions, $ft^3/min$

$\rho_S$ = Density of air = 0.074887, "standard" conditions, $lbs./ft^3$

$C_2$ = Conversion factor = 60, min/hr

Table 26 presents the refrigerant-side and air-side gross cooling capacity calculations from baseline Tests 1-3, along with percent differences (rounded to the nearest ones value) between the two methods. Percent differences range from 1% to 4%.

**TABLE 26. BASELINE GROSS COOLING CAPACITIES: REFRIGERANT-SIDE VS. AIR-SIDE**

| TEST # | % DIFFERENCE | VALUE 1 - GROSS COOLING CAPACITY: REFRIGERANT-SIDE | VALUE 2 - GROSS COOLING CAPACITY: AIR-SIDE |
|---|---|---|---|
| 1 | 20% | 56,087 | 45,932 |
| 2 | 11% | 62,659 | 55,962 |
| 3 | 6% | 67,153 | 63,483 |

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