Laboratory Test Results of Commercial Packaged HVAC Maintenance Faults

Prepared for the California Public Utilities Commission

February 25, 2016

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ABSTRACT

Laboratory tests were conducted to evaluate California ratepayer-funded commercial HVAC maintenance programs. Laboratory test results of HVAC maintenance faults were conducted on two single- and two dual-compressor R22 commercial packaged rooftop units equipped with economizers and non-TXV or thermostatic expansion valves (TXV). The average uncertainty for laboratory tests of sensible capacity and application sensible efficiency (EER*s) were 0.6% and 0.8% respectively. The non-TXV and TXV models tested in the laboratory represent 48% of total units in the programs. Laboratory tests were conducted on economizers representing 90% of all economizers receiving program services. The following tests have been completed: 1) out-of-box tests, 2) AHRI tests, 3) manufacturer refrigerant charge (RC) protocols, 4) economizer damper leakage and operation, 5) low airflow, 6) incorrect charge, 7) evaporator coil blockage, 8) condenser coil blockage, 9) refrigerant line restrictions, 10) non-condensables, 11) multiple faults, and 12) diagnostic field measurement instruments. Out-of-box tests were below AHRI ratings for dual compressor 7.5-ton units and within AHRI rating tolerances for single-compressor 3-ton units. Modifications were required to lower fan speeds and external static pressure (ESP) for each unit to establish AHRI test conditions and achieve published efficiency and capacity ratings within tolerances. The average AHRI EER rating for all units tested was 11.1. The average application efficiency for the same units tested at typical field conditions with no economizer was 20% less than the AHRI rating. The average efficiency with an economizer and closed dampers was 37% less, and with dampers 1-finger open the average efficiency was 49% less than the AHRI rating. Due to being tested under conditions and with faults they were not intended to diagnose, the manufacturer refrigerant charge (RC) protocol average accuracy was 45 +/- 3% based on 992 tests of faults on 4 units. For similar reasons, the CEC RC protocol average accuracy was 31 +/- 4% based on 445 tests. The CEC temperature split protocol average accuracy was 90 +/- 2% based on 736 tests of faults causing low airflow or low capacity. The tested protocols were less reliable with combined faults of low airflow and evaporator coil blockage indicating the importance of checking and correcting dirty filters or coil blockage before checking refrigerant charge and airflow. For comparison, studies of medical diagnostics indicate general accuracy of 31% with 55% accuracy for easier cases and 5.8% for more difficult cases. Laboratory tests indicate manufacturer troubleshooting procedures would be effective if used in a systematic manner to diagnose faults such as: overventilation, low cooling/heating capacity, blocked condenser/evaporator, refrigerant restrictions, non-condensables, and refrigerant charge faults. Troubleshooting procedures and protocols are less effective at diagnosing low airflow from undercharge, and RC protocols alone cannot diagnose other faults they were not designed to diagnose. Based on tests of five economizers installed on four units, the average closed damper outdoor air fraction (OAF) was 18 +/- 3% of total system airflow which meets or exceeds ASHRAE 62.1 minimum ventilation requirements for most buildings. Opening economizer dampers from 1-to-3-fingers provided 27 to 39% outdoor air fractions which exceed ASHRAE 62.1 minimum requirements and reduces EER*s by 20 +/- 3% compared to closed dampers.[1] The reduction in efficiency due to overventilation or outside air

[1] EER*s is defined as sensible cooling capacity divided by total electric power. EER*s is used to evaluate load impacts based on satisfying the thermostat setting which determines operational time and energy use.

leakage beyond minimum requirements represents an important energy efficiency opportunity for space cooling and heating. Sealing the gap between economizer perimeter and cabinet (under the hood) with UL-181 tape reduced OAF by 6 +/- 2% and improved EER*s by 5.4 +/- 2%. For all units tested, the average fully-open damper OAF was 68 +/- 5% which limits economizer free cooling. Laboratory tests indicate proper economizer operation improves efficiency by 6 to 120% versus damper closed and compressor-based cooling. For airflow tests where ESP was controlled by supply/return dampers, optimal efficiency was achieved with lower than rated airflow (i.e., 318 and 349 scfm/ton). Adjusting fan speed/airflow caused less impact on efficiency than increasing ESP with the code tester. Refrigerant undercharging by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% reduced EER*s by 0 to 3%. Evaporator coil blockage of 5 to 50% reduced airflow by 1 to 13% and EER*s by 1 to 11%. Condenser coil blockage of 5 to 80% increased discharge pressure and power by 1 to 33% and reduced EER*s by 2 to 36%. Liquid line restrictions (at the filter drier) reduced refrigerant temperatures by 15 to 20F and EER*s by 9 to 36%. Non-condensables of 0.25 to 1% increased discharge pressure by 13 to 29% and reduced EER*s by 9 to 22%. Efficiency impacts caused by multiple fault combinations were similar to the sum of individual fault impacts except for condenser blockage plus restrictions where the sum of individual fault impacts was less. Manufacturer troubleshooting protocols applied to test results of combined multiple faults through a logical progression reduced or eliminated “false alarms,” misdetections, and misdiagnoses compared to using refrigerant charge protocols only. Tests of eight different types of sensors on liquid and suction lines found accuracy ranging from of 1.1 +/- 0.6F for Type-K thermocouple clamps to 9.7 +/- 7.1F for insulated thermistors. Attachment and detachment tests of refrigerant hoses without EPA low-loss fittings found 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and EER*s per attachment/detachment. Tests of digital pressure measurement instruments found 0.6 +/- 0.2% accuracy based on measurements of ten pressures and 15 instruments from 6 manufacturers. Laboratory tests of a Pitot-tube array airflow grid from one manufacturer found 10.2 +/- 0.6% accuracy based on three measurements at 2,000, 2,500, and 3,000 cfm. Field data collection protocols and analytical methods have been tested in the field and the laboratory to evaluate energy efficiency impacts of condenser and evaporator coil cleaning, refrigerant charge adjustment, and economizer repair. Application sensible efficiency impacts are correlated to laboratory test results of EER*s versus compressor discharge pressure, evaporator airflow, refrigerant charge, and outdoor air fractions. The protocols can also be used by technicians to reduce unintended outdoor airflow, establish minimum outdoor airflow per AHSRAE 62.1, and improve cooling and heating efficiency.

CITATION

R. Mowris, E. Jones, R. Eshom, K. Carlson, J. Hill, P. Jacobs, J. Stoops. 2015. Laboratory Test Results of Commercial Packaged HVAC Maintenance Faults. Prepared for the California Public Utilities Commission. Prepared by Robert Mowris & Associates, Inc. (RMA).

ACKNOWLEDGEMENT

This work was prepared under a subcontract with KEMA, Inc., under prime contract number 09PS5863B for the California Public Utilities Commission. John Hill, Ph.D., provided technical guidance and review comments for J.J. Hirsch & Associates. Peter Jacobs provided technical guidance and review comments for Building Metrics. John Stoops, Ph.D., managed the study on behalf of KEMA, Inc.

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1 Executive Summary

Commercial and residential heating, ventilating, air conditioning (HVAC) energy consumption in the United States accounts for 30% of average summer peak-day electricity loads, 22% of total electricity use, and 44% of total natural gas use in commercial buildings.[2] A 2002 study published by the Hewlett Foundation indicates that improved HVAC installation and maintenance represents one of the largest achievable opportunities for energy efficiency savings.[3] This report provides laboratory test results of commercial packaged HVAC maintenance faults. Tests were performed to support the ratepayer-funded HVAC Maintenance and Installation program evaluations. Tests were performed to evaluate energy efficiency impacts of HVAC maintenance faults, fault detection diagnostic (FDD) and instrumentation accuracy, and improve the Database for Energy Efficiency Resources (DEER). Test planning began in 2012 and laboratory tests were conducted from 2013 through 2015 at Intertek in Plano, Texas, an independent Air-conditioning Heating and Refrigeration Institute (AHRI) certified laboratory. This is the same facility used by manufacturers to certify their equipment. Tests were conducted on four new packaged HVAC roof-top units with and without economizers installed.[4] The following four units were tested: 1) 7.5-ton unit equipped with multiple-fixed orifice expansion valves (i.e., non-TXV), 2) 7.5-ton units equipped with thermostatic expansion valve (TXV), 3) 3-ton unit equipped with non-TXV expansion valve, and 4) 3-ton unit equipped with TXV expansion valve.[5] The tested units used R22 as a refrigerant since they were manufactured prior to the required changeover to R410a. These R22 units were selected since the majority of installed HVAC systems still use R22. Tests results are provided to verify AHRI ratings, baseline performance, single faults, and multiple faults including combinations of faults on individual circuits and multiple circuits. The following tests have been completed on one or more units: 1) "out-of-box" tests, 2) AHRI tests, 3) manufacturer refrigerant charge protocols, 4) economizer damper leakage, 5) low airflow, 6) incorrect charge, 7) evaporator coil blockage, 8) condenser coil blockage, 9) refrigerant line restrictions, 10) non-condensables, 11) multiple faults, and 12) field measurement instruments.[6] **Appendix A** provides a data dictionary to define key

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2 United States Energy Information Agency (USEIA). 2003. Commercial Building Energy Consumption Survey. http://www.eia.gov/consumption/commercial/data/2003/pdf/c1arse-c38arse.pdf.

3 Rufo, M., Coito F. 2002. California's Secret Energy Surplus: The Potential for Energy Efficiency. Xenergy, Inc. http://www.p-2.com/PEERS/Hewlett-Foundation-Report-9-23-02.pdf.

4 Electro-mechanically controlled damper system attached to a packaged HVAC system designed to provide minimum outdoor airflow per ASHRAE 62.1 when outdoor air temperatures are greater than economizer changeover setting and maximum outdoor airflow to save energy and cool conditioned space instead of compressor-based cooling when outdoor air temperatures are lower than the changeover setting.

5 One ton of cooling is defined as the heat energy removed from one short ton of water (2,000 pounds) to produce one ton of ice at 32F (0°C) in 24 hours. The energy required for the phase change of liquid water at 32F (0C) into solid ice at 32F is referred to as the heat of fusion which is 144 Btu/lb multiplied by 2,000 lbs of water or 288,000 Btu of energy over a 24 hour period requires 12,000 Btu/hour to make one ton of ice in one day. The Btu is the energy required to raise one pound (lb) of water one degree Fahrenheit (F).

6 Out-of-box tests are performed to evaluate the tested condition of the unit as received when taken out of the

performance metrics and Intertek test file information and a glossary to define acronyms and technical terms.

Laboratory tests were conducted on one- and two-compressor systems from the largest manufacturers representing 75% of systems that received HVAC maintenance program services. The specific 1- and 2-circuit non-TXV models tested in the laboratory represent 14% of total units that received incentives in one of the largest commercial HVAC maintenance programs. Including heat pumps with similar evaporator, compressor, expansion devices, economizer and manufacturer RCA protocols, the models tested in the laboratory represent 48% of total non-TXV and TXV models in the programs.[7] Laboratory tests were conducted on economizers from the largest manufacturers representing 90% of all economizers receiving HVAC energy efficiency program services. Laboratory tests were also performed on field measurement instruments from the largest manufacturers representing 80% of instruments used by technicians performing services in the HVAC maintenance programs based on observations of technicians. Tests were performed at standard rating conditions to verify published ratings at 95F outdoor air temperature (OAT) and 80F indoor drybulb and 67F indoor wetbulb. Tests were also performed at application rating conditions of 55F, 82F, 95F, and 115F OAT and 75F indoor drybulb and 62F indoor wetbulb.

Out-of-box tests were below AHRI ratings for dual compressor 7.5-ton units and within AHRI rating tolerances for single-compressor 3-ton units. Modifications were required to lower fan speeds and external static pressure (ESP) for each unit to establish AHRI test conditions and achieve published efficiency and capacity ratings within tolerances. Due to test conditions and faults they were not intended to diagnose, the manufacturer refrigerant charge (RC) protocol average accuracy was 45 +/- 3% based on 992 tests. For similar reasons, the CEC RC protocol average accuracy was 31 +/- 4% based on 445 tests. The CEC temperature split protocol average accuracy was 90 +/- 2% based on 736 tests of faults causing low airflow or low capacity. The protocols were less reliable with combined faults of low airflow and evaporator coil blockage indicating the importance of checking and correcting dirty filters or coil blockage before checking refrigerant charge and airflow. For comparison, studies of medical diagnostics indicate general accuracy of 31% with 55% accuracy for easier cases and 5.8% for more difficult cases. Laboratory tests indicate manufacturer troubleshooting procedures might be effective if used in a systematic manner to diagnose faults such as: overventilation, low cooling/heating capacity, blocked condenser/evaporator, refrigerant restrictions, non-condensables, and refrigerant charge faults. Troubleshooting procedures and protocols are less effective at diagnosing low airflow from undercharge, and RC protocols alone cannot diagnose other faults they were not designed to diagnose.

Based on tests of five economizers installed on four units, the average closed damper OAF was 18 +/- 3% of total system airflow which meets or exceeds ASHRAE 62.1 minimum ventilation requirements for most buildings. Opening economizer dampers from 1-to-3-fingers provided 27

shipping box with factory settings and no economizer. After out-of-box tests are completed, the refrigerant charge is recovered from each circuit and carefully weighed to determine if it matches the factory name plate.

[7] The IOU HVAC Maintenance program provides tracking data for 10478 tons of similar TXV units made by the same manufacturer out of a total population of 45820 tons.

to 39% outdoor air fractions which exceed ASHRAE 62.1 minimum requirements and reduced application sensible efficiency (EER*s) by 20 +/- 3% compared to closed dampers. Sealing the gap between economizer perimeter and cabinet with UL-181 tape (under the hood) reduced OAF by 6 +/- 2% and improved EER*s by 5.4 +/- 2%. For all units tested, the average fully-open damper OAF was 68 +/- 5% which limits economizer free cooling. Laboratory tests indicate proper economizer operation improves efficiency by 6 to 120% versus damper closed and compressor-based cooling. For airflow tests where ESP was controlled by supply/return dampers, optimal efficiency was achieved with lower than rated airflow (i.e., 318 and 349 scfm/ton). Adjusting fan speed/airflow caused less impact on efficiency than increasing ESP with the code tester. Refrigerant undercharging by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% reduced EER*s by 0 to 3%. Evaporator coil blockage of 5 to 50% reduced airflow by 1 to 13%, EER*s by 1 to 11%, and total efficiency by 1 to 4%. Condenser coil blockage of 5 to 80% increased discharge pressure and power by 1 to 33% and reduced EER*s by 2 to 26%, and total efficiency by 2 to 36%. Field measurements of 28 dirty evaporators found a 0.9% impact on airflow based on coils cleaned 12.6 months previously. This corresponds to an average EER*s impact of 0.6% equivalent to 5% evaporator coil blockage based on laboratory data. Field measurements of 28 dirty condensers found a 4.6% impact on discharge pressure based on coils cleaned 12.6 months previously. This corresponds to an average EER*s impact of 3.4% equivalent to 12% condenser coil blockage based on laboratory data. A larger sample of 60 to 80 units is required to evaluate evaporators and condensers that have not been cleaned for 36 +/- 6 months prior to field measurements. A liquid line restriction on circuit 1, emulating a filter drier restriction on the 7.5-ton non-TXV and TXV units, reduced refrigerant temperatures by 15 to 20F and EER*s by 9 to 36% and cooling capacity by 11 to 39%. Non-condensables of 0.3 to 1% increased discharge pressure by 6 to 29% and power by 6 to 20% and reduced sensible cooling efficiency by 9 to 22% and capacity by 4 to 14%. Application sensible efficiency impacts caused by multiple fault combinations were similar to the sum of individual fault impacts except for condenser blockage plus restrictions where the sum of individual fault impacts was less. Manufacturer troubleshooting protocols applied to test results of combined multiple faults through a logical progression reduced or eliminated "false alarms," misdetections, and misdiagnoses compared to using refrigerant charge protocols only.

Tests of eight different types of sensors on liquid and suction lines found accuracy ranging from of 1.1 +/- 0.6F for Type-K thermocouple clamps to 9.7 +/- 7.1F for insulated thermistors. Attachment and detachment tests of refrigerant hoses without EPA low-loss fittings found 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and efficiency per attachment/detachment. Tests of digital pressure measurement instruments found 0.6 +/- 0.2% accuracy based on measurements of ten pressures and 15 instruments from 6 manufacturers. Laboratory tests of a Pitot-tube array airflow grid from one manufacturer found 10.2 +/- 0.6% accuracy based on three measurements at 2,000, 2,500, and 3,000 cfm. Additional laboratory tests are planned to evaluate other HVAC units with R22 replacement refrigerants (NU-22, RS-44), field-measurement tools, heating impacts due to economizer outdoor airflow, overventilation, unintended leakage, and low-leakage economizers.

Field data collection protocols have been tested in the field and the laboratory to evaluate energy efficiency impacts of condenser and evaporator coil cleaning, refrigerant charge adjustment, and economizer repair. Efficiency impacts have been correlated to laboratory test results of application sensible efficiency versus compressor discharge pressure, evaporator airflow, refrigerant charge, and outdoor air fractions. The protocols can be used by technicians to reduce unintended outdoor airflow, establish minimum outdoor airflow per AHSRAE 62.1 and improve cooling and heating efficiency.

The laboratory test results are applicable to non-tested systems. The test results can be used provide improved estimates of cooling system performance by modifying simulation algorithms in eQuest and, as a result, better estimates of DEER energy savings for HVAC maintenance measures with cooling system impacts.[8] The test data will also assist in the development of non-DEER work papers. The laboratory findings provide information to stakeholders including IOU program implementers, the Western HVAC Performance Alliance (WHPA), and many industry and other participants who have been working cooperatively to improve HVAC equipment energy efficiency through improved maintenance standards, certifications, training, and FDD exploration.[9] Laboratory test results have been presented at professional conferences sponsored by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), American Council for an Energy-Efficient Economy (ACEEE), and International Energy Program Evaluation Conference (IEPEC). Data have been provided to CPUC Energy Division advisors and consultants working on the DEER ex ante savings. Testing methodologies and priorities from the last cycle (2010-12) have been discussed and comments were received from the IOUs and members of ASHRAE and the WHPA.

1.1 Completed Tests and Key Findings

Laboratory tests have been performed on two dual-compressor packaged rooftop units (RTUs) and two single-compressor RTUs in order to:

1) Understand the energy efficiency impacts of observed HVAC maintenance faults,
2) Diagnose information associated with faults, and
3) Evaluate single and multiple measures in order to understand energy efficiency and peak savings potential across a variety of common packaged rooftop HVAC unit designs.

As noted above, the CPUC initiated laboratory testing to evaluate savings being claimed by HVAC maintenance and installation programs, improve DEER updates, examine FDD reliability and field instrument accuracy, and research unexpected findings. The 2010-12 and 2013-15 HVAC Maintenance programs measure list includes:

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[8] Itron 2005. 2005-2007 Database for Energy Efficiency Resources (DEER) Update Study, Final Report. Itron, Inc., J.J. Hirsch & Associates, Synergy Consulting, and Quantum Consulting. Also see www.deeresources.com.

[9] ANSI/ASHRAE/ACCA 2008. American National Standards Institute (ANSI), American Society of Heating Refrigeration and Air-Conditioning Engineers (ASHRAE), Air Conditioning Contractors of America (ACCA). ANSI/ASHRAE/ACCA Standard 180: Standard Practice for Inspection and Maintenance of Commercial Building HVAC Systems. Hunt, M., Heinemeier, K., Hoeschele, M., Weitzel, E. 2010. HVAC Energy Efficiency Maintenance Study. CALMAC Study ID SCE0293.01.

1) Condenser coil cleaning,
2) Evaporator coil cleaning,
3) Adjust airflow,
4) Refrigerant test,
5) Refrigerant service,
6) Economizer functional test,
7) Adjust economizer change-over setting,
8) Check and re-position outside air dampers (SDGE3226),
9) Economizer repair,
10) Replace thermostat,
11) Adjust thermostat schedule,
12) Comb condenser fins,
13) Replace damaged refrigerant line insulation, and
14) Notched v-belt upgrade.

Some of these measures have been discontinued and some are receiving little or no incentives (i.e. replace damaged refrigerant line insulation, comb condenser fins). Replace thermostats or adjust thermostat schedules are best verified in the field and simulated using the eQuest or DOE-2 computer programs.[10] The following measures have been tested both in the field and the laboratory to develop field measurement protocols and laboratory setup procedures.

- Condenser coil cleaning (i.e., blockage) was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts. Condenser discharge pressure was measured in the field before and after cleaning coils and the ratio of dirty-to-clean condenser coil discharge pressure measurements (at constant OAT) are used to evaluate the impacts of condenser coil cleaning. Coil blockage was emulated in the laboratory by installing plastic-corregated cardboard to cover 5 to 80% of the upstream side of the condenser. Regression equations are based on laboratory test results of application sensible efficiency versus discharge pressure ratio due to coil blockage at constant OAT.
- Evaporator coil cleaning (i.e., blockage) was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts. Evaporator airflow was measured in the field before and after cleaning coils and the ratios of dirty-to-clean airflow measurements are used to evaluate the impacts of evaporator coil cleaning. Coil blockage was emulated in the laboratory by installing plastic-corregated cardboard to cover 5 to 50% of the upstream

[10] J.J. Hirsch & Associates. 2014. eQuest. Quick Energy Simulation Tool. http://www.doe2.com/equest/ DOE-2.2 Building Energy Use and Cost Analysis Program: Volume 6: New Features, Version 41-48. February 2014.

side of the evaporator. Regression equations are based on laboratory test results of EER*s versus evaporator airflow due to coil blockage.

- Adjust airflow was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts based on the ratios of as-found airflow per manufacturer recommended airflow. Regression equations are based on laboratory tests of EER*s versus percent airflow from 60 to 110% of 400 cfm/ton.
- Refrigerant charge testing and service (i.e., incorrect charge) was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts based on the ratios of as-found refrigerant charge per factory charge (recovery and weigh-out of refrigerant). Regression equations are based on laboratory tests of EER*s versus percent factory charge from -40% to +40%.
- Economizer test and repair was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts based on measurements of economizer performance. Laboratory tests were performed at outdoor drybulb temperatures ranging from 50 to 70 degrees Fahrenheit with different economizer controls, sensors, actuators, and damper positions (i.e., outdoor air leakage). Non-steady-state transient tests of economizers need to be performed to evaluate economizer damper controls and functionality to understand economizer, sensor, and thermostat control integration.
- Economizer repair including check and re-position outside air dampers (SDGE3226) was measured in the field and the laboratory to develop methods to evaluate energy efficiency impacts based on the as-found outdoor air fraction. Regression equations are based on laboratory tests of EER*s versus OAF. Outdoor airflow fractions were measured using average outdoor, return, and mixed air temperatures when the outdoor drybulb temperature is at least 20 degrees Fahrenheit greater than the return air temperature. These protocols can also be used by technicians to reduce unintended outdoor airflow, establish minimum outdoor airflow per AHSRAE 62.1, and improve cooling and heating efficiency.

The following HVAC maintenance measures were tested in the laboratory.

- Refrigerant charge was tested with 60 to 140% of factory charge at 82, 95, and 115F OAT.
- Condenser coil blockage was tested with plastic corrugated cardboard to cover the upstream side of the condenser by 5 to 80% to increase discharge pressure by 2 to 50%.
- Evaporator coil blockage was tested with plastic corrugated cardboard to cover the upstream side of the evaporator by 5 to 50% to decrease evaporator airflow by 1 to 18%.
- Airflow were tested by adjusting fan speed (rpm) or external static pressure (ESP) to emulate duct system installation faults to vary airflow by 60 to 110% of 400 cfm/ton.
- Economizer outdoor air leakage and efficiency was tested with closed, 1-finger, 2-finger, 3-finger and fully open dampers with and without tape to seal the gap between perimeter frame and cabinet at 55F, 95F and 115F OAT.[11]

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[11] Some economizer manufacturers have a default of 3.2V (20% open) for low speed fan (heating mode) and 2.8V

- Liquid line restrictions were tested by closing a "service" valve installed upstream of liquid-line driers to produce a 14 to 20F liquid refrigerant temperature decreases.
- Non-condensables air and water vapor faults were tested by adding 0.4 to 1 ounce of Nitrogen per circuit to produce 10 to 30% discharge pressure increases.
- Multiple faults were tested with a combination of low airflow, low charge, economizer damper outdoor airflow, condenser coil blockage, evaporator coil blockage, or restrictions.
- Cycling tests were performed to evaluate cycling loss assumptions used in IEER rating calculations and DEER simulations.
- Manufacturer and generic CEC refrigerant charge and airflow FDD protocols were tested for accuracy, false alarms, misdetection, or misdiagnoses under non-faulted and faulted test conditions they were not intended to diagnose.

Field observations and data from previously published studies were used to establish baseline laboratory test conditions with review comments and suggestions from CPUC consultants, HVAC PCG stakeholders, and WHPA members. Laboratory test results have been presented to the CPUC, IOUs, WHPA, HVAC industry, and energy efficiency professionals at numerous venues within months of the completion of the first laboratory tests at Intertek. Peer-reviewed papers were presented and published by IEPEC and ACEEE. Laboratory tests conducted in 2012-15 have provided the following overall key findings.

- Out-of-box tests were below the AHRI ratings for dual compressor 7.5-ton units and within AHRI rating tolerances for single-compressor 3-ton units.
- Modifications were required to lower fan speeds and ESP for each unit to establish required AHRI test conditions and achieve published efficiency and capacity ratings within AHRI tolerances.
- The roof top units (RTU) perform as rated when tested under AHRI-specific conditions without an economizer installed, 0.15 to 0.25 inches of total static pressure, low fan speed, 95F outdoor conditions, and 80F drybulb (DB) and 67F wetbulb (WB) indoor conditions.[12]
- Almost all RTUs are installed with a vertical conditioned air discharge while all AHRI rating tests are performed with a horizontal conditioned air discharge. While Intertek is in the process of building a vertical test chamber, most AHRI-certified laboratories are unable to easily test units in the vertical configuration (i.e., vertical air inlet and outlet). The impact of this testing constraint is currently unknown but adds an additional level of uncertainty to AHRI ratings as compared to actual field performance.
- Manufacturer refrigerant charge (RC) protocol average accuracy was 48 +/- 3% based on 992 tests of maintenance faults on four units due do being tested under conditions and with faults

(10% open) for high speed (cooling mode).

[12] AHRI 2014/240 minimum external resistance for 3-ton units is 0.15 inches of water column (IWC) and AHRI 340/360 minimum external resistance for 7.5-ton units is 0.25 IWC. One IWC is equivalent to 0.03612 per square inch (psi) or 249.088 Pascal at 0C.

they were not intended to diagnose. For the same reasons, the CEC RC protocol average accuracy was 31 +/- 4% based on 445 tests of faults on three units. Resolving obvious maintenance faults such as cleaning coils, installing clean air filters, and reducing outdoor airflow by temporarily sealing the economizer prior to initial FDD testing will improve accuracy. For comparison, studies of medical diagnostics indicate general accuracy of 31% with 55% accuracy for easier cases and 5.8% for more difficult cases.[13]

- The CEC temperature split (ΔTS) protocol average accuracy was 90 +/- 2% based on 736 tests of faults causing low airflow or low sensible cooling capacity due to overventilation, evaporator/condenser blockage, refrigerant over/undercharge, refrigerant restrictions, and non-condensables. The CEC ΔTS protocol properly identified faults associated with low airflow, low cooling capacity and excess outdoor airflow causing lower cooling capacity by 7% or more. The CEC ΔTS protocol was unreliable with combined faults of low airflow and evaporator blockage indicating the importance of correcting obvious maintenance faults such as dirty filters or coil blockage before checking refrigerant charge and airflow.
- Laboratory tests indicate manufacturer troubleshooting procedures will be effective if used in a systematic manner to diagnose faults such as: excessive outdoor air, low cooling/heating capacity, blocked condenser/evaporator, refrigerant restrictions, non-condensables, and refrigerant overcharge or undercharge.
- Total and sensible EER* of the five units tested were 21 to 37% lower than AHRI ratings when tested under emulated field conditions with an economizer installed, typical ESP, and factory fan speeds at 95F OAT and 75F DB and 62F WB indoor conditions.
- Codes and standards programs and Title 24 assume rated performance to establish expected energy impacts. Laboratory tests of emulated field performance under optimal installation and maintenance with economizers installed and closed dampers can be 18 to 53% less efficient than rated performance.
- Units tested with open economizer dampers were 25 to 40% less efficient than AHRI ratings.
- Based on tests of five economizers installed on four units, the average closed economizer damper outdoor airflow (OA) was 18 +/- 6% of total system airflow which meets or exceeds ASHRAE 62.1 minimum ventilation requirements for most buildings. Opening dampers from 1-to-3-fingers provided 27 to 39% OAF which exceeds ASHRAE 62.1 minimum requirements and reduced EER*s by 10 to 32% compared to closed dampers.[14] Tests with tape sealing the gap between economizer perimeter and cabinet (under the hood) found an

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[13] Ashley N. Meyer. D. Payne V. Meeks. D. Rao. R. Singh. H. 2013. Physicians' Diagnostic Accuracy, Confidence, and Resource Requests: A Vignette Study. Journal of the American Medical Association (JAMA) Internal Medicine. http://archinte.jamanetwork.com/article.aspx?articleid=1731967. Ross. R. 2014. Expert Opinion Software for Medical Diagnosis and Treatment. JAMA Internal Medicine. 2014; 174(4):638-639. doi:10.1001/jamainternmed.2013.13794.

[14] HVAC technicians establish minimum outdoor damper openings using 1, 2, or 3 of their fingers. Tests were performed with the following finger diameters: 1-finger is 0.74 inch (1.88 cm), 2-fingers is 1.289 inches (3.27 cm), and 3-fingers is 1.972 inches (5.01 cm).

OAF difference of 6 +/- 2% and improved EER*s of 5 +/- 2%. For all units tested, the average fully-open damper OAF was 68 +/- 4.6% which limits economizer free cooling.

- Laboratory tests indicate that proper economizer operation improves EER*s by 6 to 120% compared to non-functional economizer with damper closed and compressor-based cooling.
- For airflow tests where ESP was controlled by supply/return dampers, the optimal airflow/efficiency was achieved with lower than rated airflow (i.e., 318 and 349 scfm/ton). Adjusting fan speed/airflow at unit caused less impact on efficiency than increasing ESP with the code tester.
- Refrigerant undercharging by 5 to 40% reduced EER* by 4 to 47% and overcharging by 5 to 40% reduced EER*s by 0 to 3%.
- Evaporator coil blockage of 5 to 50% reduced airflow by 1 to 13%, reduced EER*s by 1 to 11%, and reduced total efficiency by 1 to 4%.
- Condenser coil blockage tests of 5 to 80% increased discharge pressure by 2 to 33% and total power by 1% to 24% and reduced EER*s by 2 to 26% and total efficiency by 2 to 36%.
- Restrictions on circuit 1 of the 7.5-ton non-TXV and TXV units caused a 15 to 20F temperature drop and reduced EER*s by 8 to 36% and capacity by 11 to 39%.
- Non-condensables of 0.25 to 1% on the 7.5-ton non-TXV and TXV units reduced EER*s by 9 to 22% and capacity by 4 to 14% and increased power by 6 to 26% depending on OAT.
- Multiple fault impacts on the 7.5-ton non-TXV, 7.5-ton TXV, and 3-ton non-TXV units were similar to the sum of individual fault impacts (except for condenser blockage plus C1 restriction where sum was 8 to 13% less). Manufacturer troubleshooting protocols applied to test results of combined multiple faults through a logical progression reduced or eliminated misdetections, and misdiagnoses compared to only using refrigerant charge protocols.
- Tests of eight types of temperature sensors on liquid and suction lines found accuracy of 1.1 +/- 0.6F for Type-K thermocouple clamps to 9.7 +/- 7.1F for insulated thermistors.
- Refrigerant pressure sensor attachment and detachment tests without EPA low-loss fittings found 0.4 to 0.5% loss of charge and 0.2% reduced EER*s per test based on 60 tests.
- Digital pressure measurement instruments tests found 0.6 +/- 0.2% accuracy based on measurements of ten different pressures and 15 instruments from 6 manufacturers.
- Pitot-tube array airflow tests for one manufacturer found 10.2 +/- 0.6% accuracy based on three measurements at 2,000, 2,500, and 3,000 cfm.

The laboratory tests indicate manufacturer factory charge provided optimal efficiency under most conditions. However, depending on pre-existing faults, technicians using currently available tools and protocols may incorrectly diagnose units with the correct factory charge as undercharged or overcharged. Incorrect measurements and/or diagnosis might cause technicians to add or remove refrigerant charge which can cause efficiency degradation. Pre-existing faults can include: condenser/evaporator coil fouling, overventilation, cabinet/unintended outdoor air leakage, economizer failure, low airflow, improper refrigerant charge, refrigerant-line restrictions, non-condensables, or refrigerant contamination.

Laboratory testing discovered all tested units have unintended outdoor air leakage around economizer frames and cabinet panels reducing EER*s by 3 to 15%. Economizer repair savings are based on the assumption that economizers actually provide 5 to 15% outdoor air at the minimum position and 85 to 95% outdoor air when fully open. Laboratory tests indicate economizers actually provide 18 +/- 3% outdoor airflow when fully closed and 68 +/- 5% outdoor airflow when fully open.

2 BACKGROUND

Laboratory tests of HVAC maintenance faults are intended to assist with evaluation studies of California ratepayer-funded commercial HVAC maintenance programs. Measuring the impacts of HVAC maintenance faults in the field is impossible due to constantly changing indoor and outdoor temperature and pressure conditions. Laboratory tests can accurately control and measure return, supply, and outdoor air drybulb and wetbulb temperatures, airflow, static pressure, power, and refrigerant pressures and temperatures. The synergistic relationship between field and laboratory measurements of HVAC measures is well established.[15] Laboratory tests are performed in an AHRI-certified facility under carefully controlled conditions in order to accurately measure performance parameters with and without single and multiple maintenance and installation faults. The laboratory tests provide scientific engineering information to assist with evaluation studies of HVAC maintenance measures on the CPUC Efficiency Savings and Performance Incentives (ESPI) list.[16] Laboratory testing also provides data for updating ex-ante estimates developed for the DEER analyses and work papers.[17]

Laboratory tests are performed of single- and multiple-faults under various operating conditions typical of field installed systems. For example, condenser coil cleaning saves energy by reducing discharge pressure and compressor power and increasing cooling capacity and efficiency. Tests of incremental discharge pressure increases caused by varying the amount of condenser coil blockage under different operating conditions provides performance curves that can be mapped to field measurements of discharge pressure before and after cleaning condenser coils. Similar performance curves are developed based on tests of evaporator coil blockage, refrigerant charge

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[15] The California Evaluation Framework, Chapter 13: Sampling, prepared for the CPUC, prepared by Hall, N., Barata, S., Chernick, P., Jacobs, P., Keating, K., Kushler, M., Migdal, L., Nadel, S., Prahl, R., Reed, J., Vine, E., Waterbury, S., Wright, R. February 2004. "For example, the sensitivity of the efficiency of an air conditioner to refrigerant charge and air flow variation may be studied in a laboratory, using instrumentation and test protocols that cannot be easily duplicated in field. Once this relationship is established, it can be applied to field measurements of refrigerant charge and air flow to estimate the impacts of correcting these problems."

[16] ESPI measures requiring ex-post evaluation are listed in D.13-09-023, Attachment 3.

[17] DEER contains information on selected energy-efficient technologies and measures. DEER provides estimates of the energy-savings potential for these technologies in residential and nonresidential applications. The database contains information on typical measures – those commonly installed in the marketplace – and data on the costs and benefits of more energy-efficient measures. Energy-efficient measures provide the same services using less energy, but they usually cost slightly more. DEER updates have been developed by the California Public Utilities Commission (CPUC) with funding provided by California ratepayers. http://www.deeresources.com/

and airflow faults, economizer damper positions, cabinet leakage, refrigerant restrictions, and non-condensables.

Laboratory and field tests are intended to be integrated into the overall HVAC maintenance and installation impact evaluation. The combination of laboratory and field testing provides the experimental scientific basis for evaluating the impacts of HVAC maintenance services that are not possible in field-only or laboratory-only studies. For example, the energy efficiency impacts of condenser coil cleaning can only be measured in the field by measuring the compressor discharge pressure associated with pre-existing dirty or blocked condensers, thoroughly cleaning the coils, and measuring the discharge pressure afterwards at constant outdoor temperature. Evaporator coil cleaning must be evaluated in the same way measuring the evaporator airflow associated with pre-existing dirty or blocked air filters and evaporator coils, installing clean air filters and thoroughly cleaning the coils, and measuring the evaporator airflow afterwards. These measurements must be made on a statistically significant population of at least 60 to 80 units with dirty condenser or evaporator coils that have not been cleaned for 12 to 36 months prior to measurements to correlate EER*s impacts associated with changes in discharge pressure or airflow based on the time for coils to become dirty or blocked. Maintenance degradation occurs over time. If the field evaluation does not include a large enough sample of faulted units over time, it will be impossible to evaluate energy efficiency impacts.

Refrigerant charge measures can be evaluated in the field by recovering charge on a sample of units and correlating these measurements to laboratory tests of refrigerant charge impacts on the application sensible energy efficiency. Recovering and weighing out charge cannot be performed by typical air conditioning technicians since it requires specialized equipment and expertise to accurately recover the charge, Measuring economizer outdoor air leakage also requires specialized equipment, procedures and expertise to accurately measure and correlate outdoor airflow measurements to the application sensible energy efficiency and develop accurate building energy simulations to evaluate space cooling and heating energy savings estimates.

Energy savings benefits provided by DEER for weather-dependent measures are derived from eQuest computer simulations of prototypical buildings. The energy savings are based on simulated HVAC systems performance driven by simulated space heating and cooling loads. Laboratory measurements of HVAC maintenance faults help to improve the ability of simulation models to predict the performance of HVAC systems with faults under typical field conditions, thus improving the DEER estimates, as well as IOU work paper estimates that rely on DEER methods. Laboratory results to date have been applied to the DEER modeling process for economizer operation, supply fan power and part-load effects on unit efficiency. This affects all buildings served by packaged HVAC systems, as described in **Section 2.1**.

Laboratory tests focused on the cooling performance of 3-ton and 7.5-ton unitary packaged HVAC systems covered under ANSI/AHRI Standards 210/240 and 340/360. Considering the 33% impact of HVAC end uses on peak electricity loads, any improvement in actual system performance revealed by laboratory testing is important. Laboratory tests of HVAC maintenance faults performed under application field conditions with an economizer installed will produce significantly different results than tests performed at the ANSI/AHRI 210/240 or 340/360 Standard Rating Conditions.

For buildings served by unitary cooling systems, the energy simulation of the cooling system in the DEER process uses a set of certified efficiency values and a number of performance maps that adjust rated values to non-rated conditions.[18] Rated conditions include steady-state total and sensible cooling capacities and cooling efficiency and fan power values that occur at the AHRI "A" ratings point. The AHRI 210/240 and 340/360 "A" ratings point is defined as steady-state operation with an ambient temperature of 95ºF and return air conditions to the cooling coil of 80ºF dry bulb temperature and 67ºF wet bulb temperature. The DEER team derives performance maps from manufacturers' expanded engineering tables and supply fan performance tables, which are normalized to the performance at the AHRI "A" rating point. These data are typically obtained from heating and cooling system engineering literature and are likely based on computer simulations. The quality, completeness and usefulness of these data sets vary across manufacturers and unit types. In almost all cases, some performance estimates at conditions not included in the manufacturer's expanded engineering data are required to complete the performance maps. Laboratory tests help to clarify and develop those estimates not provided in manufacturers' engineering literature and provide guidance on how systems perform under conditions that occur in the field.

Laboratory tests have been performed on two 3-ton single-compressor and three 7.5-ton two-compressor systems from the largest manufacturers representative of 75% of systems that received HVAC maintenance program services in the program.[19] The 1- and 2-circuit non-TXV models tested in the laboratory represent 14% of total units that received incentives in one of the largest commercial HVAC maintenance programs. Testing of a larger sample is required, including heat pumps with similar evaporator, compressor, non-TXV expansion devices, economizer, and manufacturer RCA protocols. The non-TXV models tested in the laboratory represent 25% of total units (i.e., tons of cooling). The TXV models tested represent 22.9% of total units (i.e., tons of cooling) that participated in the same program.

Laboratory tests have been performed on economizers from the largest manufacturers representing 90% of all economizers receiving maintenance services under the programs. Tests have been performed on field measurement instruments from the largest manufacturers representing 80% of instruments used by technicians performing services in the maintenance programs. Laboratory test data can be applied to non-tested systems since the fundamental operational characteristics do not differ widely between manufacturers. The test data should provide improved estimates of cooling and heating system performance by modifying simulation algorithms in eQuest and, as a result, better estimates of DEER energy impact for all measures with cooling system impacts.

[18] Performance maps are bi-quadratic equations based on manufacturer performance data for makes and models of units sold in California.

[19] Program tracking data are used to determine quantity of units receiving HVAC maintenance program incentives by manufacturer (see EEGA 2267 RCA Data_SDGE3161.xls).

2.1 Laboratory Tests Improve Program Evaluation, Design, and Delivery

Laboratory test results improve understanding about HVAC maintenance faults and their impact on HVAC system performance. The test results are used to perform ex-post evaluation of program performance, improve ex-ante savings estimates used in program design, evaluate fault detection diagnostic test procedures, and investigate new measures to improve HVAC maintenance program delivery.

2.1.1 Laboratory Tests Inform Program Evaluation

One of the primary purposes of laboratory tests is to inform impact evaluations of HVAC maintenance programs and programs that provide incentives for the installation of new high efficiency HVAC equipment. There are large uncertainties associated with HVAC maintenance measures and the in-situ efficiency of new high efficiency equipment. Laboratory tests help reduce uncertainties by providing accurate energy efficiency performance information under typical operating conditions. Laboratory test data can also be used to develop building energy simulation models or support other analytical methods to evaluate energy savings from HVAC maintenance and installation programs. Tests of single- and multiple-faults provide improved information and methods to evaluate load impacts. For example, laboratory tests of sensible energy efficiency impacts versus fault conditions can be correlated to field measurements of fault conditions to evaluate energy efficiency impacts of refrigerant charge, coil blockage, or economizer faults at various airflow rates, damper positions, and indoor/outdoor temperature conditions. Recovering and weighing refrigerant charge and comparing under or overcharge amounts to factory charge can be used to estimate load impacts of refrigerant charge measures. Similarly measurements of discharge pressure at constant OAT, or evaporator airflow before and after installing clean air filters and cleaning dirty coils not cleaned for a known time period can be used to estimate load impacts of coil cleaning measures.

2.1.2 Laboratory Tests Improve Program Design and DEER Updates

Laboratory tests have been used to improve program design by improving DEER updates of ex-ante savings estimates for HVAC maintenance program energy efficiency measures. Lab tests are used to improve building energy simulation models used to update DEER ex ante energy savings estimates. The following is a partial list of DEER updates based on lab test results.

- DEER update no longer "forces" packaged unit performance maps to match AHRI EER ratings. Results from out-of-box laboratory tests indicating 20 to 30% lower EER levels than previously assumed at field conditions with cabinet/economizer damper leakage, and factory fan speeds are used to establish unit efficiency more representative of equipment as delivered from the factory.[20]

[20] Fan speed is generally established though fan-motor tap selection for direct drive fans or drive-sheave positions

- DEER update includes improved part-load performance maps for larger EER-rated units (>65,000 Btuh cooling capacity) based on laboratory test results. Cycling-loss tests of two 7.5-ton 2-stage TXV units found cyclic degradation coefficients (C_d) 2 or 3 times greater than previously assumed. Measured cycling losses were approximately twice that assumed in IEER rating formulae. New DEER part-load performance maps based on higher lab-measured cycling losses have been incorporated into most recent code-update. The revised part-load maps affect both code-compliant and Tier level units equally.
- Updated DEER economizer model includes higher minimum damper position outdoor air leakage and lower outdoor airflow at maximum (fully open) damper position. Additional revisions to the DEER models are necessary to accurately model interactions between the thermostat and the economizer controller to include part-load economizer performance with fan-only (no compressors) versus economizer performance with compressor operation.
- Updated DEER refrigerant charge measure will include lower cooling capacity and efficiency impacts due to refrigerant under/over charge.

2.1.3 Laboratory Tests Evaluate FDD Protocols and Procedures

Laboratory test results are used to evaluate FDD protocols and procedures used by technicians participating in energy efficiency programs and complying with California building energy efficiency standards. Laboratory test results are also used to evaluate manufacturer FDD protocols and procedures. **Section 4** provides unit-specific manufacturer refrigerant charge diagnostic protocols for each unit tested. **Section 4** also provides the California Energy Commission (CEC) refrigerant charge and airflow (RCA) protocols for each unit tested.[21] The protocols are evaluated and compared for accuracy in terms of diagnosing refrigerant charge and airflow faults. The CEC RCA protocols have been used in California in commercial HVAC maintenance programs since 2004.[22] The CEC protocols provide methods to evaluate superheat

and pulley diameters for belt-driven fans.

[21] CEC 2001. Energy Efficiency Standards for Residential and Nonresidential Buildings Energy Commission Publication No. P 400-01-024. Appendix L – Procedures for Determining Required Refrigerant Charge and Adequate Airflow for Split System Space Cooling Systems without Thermostatic Expansion Valves. http://www.energy.ca.gov/title24/archive/2001standards/2001-10-04_400-01-024.PDF. CEC 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. RA3.2 Procedures for Determining Refrigerant Charge for Split System Space Cooling Systems Without a Charge Indicator Display. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

[22] PG&E. 2004. PY2004-2005 Verified Charge and Airflow Upstream Incentive Program Policies and Procedures Manual Residential and Nonresidential. PG&E. 2006. PY2006 Verified Charge and Airflow Contractor Incentive Program Field Policies and Procedures Manual. PG&E. 2007-08. Verified Charge and Airflow Contractor Incentive Program Field Policies and Procedures Manual. Conservation Services Group (CSG). 2007. CPACS Commercial Verified Charge Adjustment Technical Specifications. CSG. 2011-2015. Contractor Manual Commercial HVAC Tune-ups, Quality Maintenance and Related Measures. SDG&E Premium Efficiency Cooling Program. Version 1.0 – August 25, 2011 through Version 1.4 - March 1, 2015.

(SH) for air conditioners equipped with non-TXV fixed orifice or capillary tube expansion devices and subcooling (SC) for air conditioners equipped with a thermostatic expansion valve (TXV). For non-TXV units, the actual SH must be within +/-5F of the target SH for the non-TXV system to pass the refrigerant charge test. The actual superheat is the refrigerant suction line temperature minus the evaporator saturation temperature (based on refrigerant suction pressure). The CEC target SH values are published in two tables and require measurements of condenser entering outdoor air DB temperature and return air WB temperature to determine the target SH. For TXV units, the actual SC must be within +/-3F of the target SC provided by the manufacturer for the TXV system to pass the refrigerant charge test. The difference between actual and target SC is referred to as ΔSC.

The CEC protocol provides a method to evaluate airflow based on the actual temperature split (TS) measurement equal to the return air DB minus supply air DB temperature. The CEC target TS values are published in a table and require measurements of return air DB and WB temperatures to determine the target TS. According to the CEC Reference Appendices, if the actual TS minus target TS (ΔTS) "is between plus 3F and minus 3F, then the system passes the adequate airflow criterion" or if ΔTS is "between minus 3F and minus 100F, the system passes, but it is likely that the (sensible) capacity is low."[23] The CEC TS protocol provides an FDD method to evaluate low airflow or low capacity.

The CEC superheat, subcooling, and temperature split protocols are based on HVAC Servicing Procedures which provide the following information.[24]

> "The focus of this manual is placed on the field servicing of residential and light commercial HVAC equipment. Unlike many other manuals written about servicing HVAC equipment, this manual encompasses all areas of servicing, including mechanical refrigeration system, electrical system, and air distribution system. This manual has been designed as a field companion to be carried with you in your truck and on the job."

The CEC target superheat table (RA3.2-2) is taken from Superheat Calculator (GT24-01) and the CEC target temperature split table (RA3.2-3) is taken from the Proper Airflow Range Calculator (GT24-01).[25] Other manufacturers provide similar superheat, subcooling, and airflow

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[23] CEC. 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. Effective January 1 2010.

[24] Carrier. 1995. HVAC Servicing Procedures. SK29-01A, 020-040. Carrier Service Procedure SP-14, page 164-167, provides the temperature split (TS) method to check airflow. Carrier Service Procedure SP-4, page 78-89, provides the superheat (SH) method for non-TXV units or subcooling (SC) for TXV units. CEC Target Superheat Table RA3.2-2 is taken from the Carrier Superheat Calculator (Carrier Corporation. 1986. GT24-01). CEC Target Temperature Split Table RA3.2-3 is taken from the Carrier Proper Airflow Range Calculator (Carrier Corporation. 1986. GT24-01).

[25] Carrier 1986. R22 Superheat, Subcooling, and Airflow Calculator. GT24-01 020-434. Carrier 1998. R410A Superheat, Subcooling, and Airflow Calculator. GT58-01A 020-517.

calculators.[26] The CEC RCA protocol is based on manufacturer RCA protocols that have been used by field technicians for more than 30 years.

The applicability of FDD protocols based on generic superheat or subcooling target values can introduce additional uncertainty especially if the generic protocol is inconsistent with unit-specific manufacturer protocols that involve multiple and different parameters such as direct measurements of suction pressure, discharge pressure, suction temperature, or liquid temperature rather than indirect measurements of superheat or subcooling. The applicability of generic airflow FDD protocols such as the CEC temperature split (TS) protocol was found to have significantly less uncertainty since it is applicable to all systems irrespective of refrigerant expansion valves and number of circuits.

Laboratory test data and other research studies indicate that Air Conditioner Maintenance (ACM) FDD protocols can provide “false alarms,” misdetections, or misdiagnoses of refrigerant charge and airflow faults. Test results presented in **Section 4** indicate manufacturer unit-specific refrigerant charge protocol average accuracy was 49% and the CEC refrigerant charge protocol average accuracy was 44% over the range of charge faults and protocols tested. The CEC temperature split protocol average accuracy was 90% when diagnosing low airflow or low sensible cooling capacity due to overventilation or other maintenance faults. For comparison, studies of medical diagnostics indicate general accuracy of 31% with 55% accuracy for easier cases and 5.8% for more difficult cases.[27]

Yuill and Braun evaluated the CEC Refrigerant Charge Analysis (RCA) protocol and reported 41% correct diagnosis for non-TXV and 64% correct diagnosis for TXV equipped systems. Yuill and Braun evaluated five FDD protocols and reported “false alarm” rates of 37 to 85% based on experimental data.[28] FDD protocols are further confounded by the presence of economizers or ventilation air dampers that lead to incorrect measurement of evaporator coil entering air conditions upon which manufacturer or generic refrigerant charge charts are based. Additional FDD issues are caused by improper airflow (< or >400 cfm/ton); coil blockage, non-condensables, refrigerant restrictions, or measurement instrument errors which the ACM FDD methods assume are not present.

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26 York. 1991. Required Superheat Calculator (non-TXV), Subcooling Calculator (TXV), and Proper Airflow Range. Form 501.00-PM5Y (5/91). Trane. 1996. Air Conditioning Charging Calculator Required Superheat, TXV Refrigerant Charging Curve. Pub. No. 22-8065-07.

27 Yuill, D, Braun, J. 2012. Evaluating Fault Detection and Diagnostics Protocols Applied to Air-Cooled Vapor Compression Air-Conditioners, International Refrigeration and Air Conditioning Conference. http://docs.lib.purdue.edu/iracc/1307.

28 Braun, J. Yuill, D. 2014. Evaluation of the effectiveness of currently utilized diagnostic protocols. Ray W. Herrick Laboratories Purdue University. Prepared for Portland Energy Conservation, Inc. “False alarm” is defined as diagnosis of a fault with the following: 1) fault intensity ratio (FIR) for capacity and COP are above 95% threshold, 2) refrigerant charge is less than 105% of “nominal,” and 3) superheat is between 1 and 36F. “Nominal” is defined as charge yielding maximum COP at 95F outdoor and 80/67F indoor. Laboratory tests do not adhere to this relative definition and does not adopt the “fault intensity ratio” which is not feasible in the field where technicians do not have access to accurate measurements of capacity or efficiency.

Laboratory and field measurements of unit-specific manufacturer FDD protocols indicate fewer problems diagnosing refrigerant charge faults when no other faults are present due to wider tolerances and multi-step procedures.[29] Nevertheless, both types of protocols have limitations and neither can distinguish non-condensables and restrictions from refrigerant charge faults, condenser or evaporator heat transfer faults, low airflow, or expansion valve failure. Manufacturer troubleshooting protocols applied to test results of combined multiple faults through a logical progression reduced or eliminated "false alarms," misdetections, and misdiagnoses compared to using refrigerant charge protocols only. Laboratory tests indicate manufacturer troubleshooting procedures might be effective if used in a systematic manner to diagnose faults such as: overventilation, low cooling/heating capacity, blocked condenser/evaporator, refrigerant restrictions, non-condensables, and refrigerant overcharge or undercharge. However, laboratory tests indicate that manufacturer troubleshooting procedures and refrigerant charge protocols will likely be less effective at diagnosing low airflow from undercharge, and refrigerant charge protocols in general cannot diagnose other faults.

These findings are significant and require additional testing. Laboratory tests have not been performed on systems with micro-channel heat exchangers (MCHE). Manufacturers claim MCHE systems may be more sensitive to incorrect charge, non-condensables, restrictions, or coil blockage than systems with conventional heat exchangers. MCHE systems use require 20 to 40% less refrigerant and are about 10% more efficient than conventional tube and fin condensers.[30]

2.1.4 Laboratory Tests Improve Field Measurement Accuracy

Evaluation of generic and unit-specific manufacturer protocols used for diagnosing refrigerant charge and airflow faults require accurate measurement of the "correctness" of the diagnosis and applicability of the protocol in the field. Measurement errors inherent to even the most careful field measurements can result in the requirement that a fault must have an efficiency impact that is large enough to rise above the uncertainty of the measurement (typically a 5% negative impact on cooling capacity or efficiency from the rated performance). If a FDD protocol identifies a fault, but the fault causes less than 5% degradation in performance (either capacity or efficiency), it is considered a False Alarm.[31] Another consideration is that instruments commonly used by

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[29] Mowris, R., Eshom, R., Jones, E. 2013. Lessons Learned from Field Observations of Commercial Sector HVAC Technician Behavior and Laboratory Testing. IEPEC. http://www.iepec.org/conf-docs/conf-by-year/2013-Chicago/129.pdf#page=1. Mowris, R., Eshom, R., Jones, E. 2011. Laboratory Measurements of HVAC Installation and Maintenance Faults. ASHRAE. June 2011. Mowris, R., Eshom, R., Jones, E. 2011a. Procedures to Diagnose and Correct Refrigerant Restrictions and Non Condensables for Residential HVAC Split Systems, Prepared for the California Energy Commission, September 6, 2011.

[30] Carrier, 2007, Commercial documentation on micro-channel heat exchangers, www.carrier.com. Cremaschi, 2007, HPC, 2007, Heat Pump Center, Newsletter #3, 2007. Yanik, M. Jianlong, J. 2012. Application of MCHE in Commercial Air Conditioners. Danfoss. 2013. How to Cut Costs and Impacts of Your AC and Refrigeration Systems.

[31] J. Braun, D, Yuill. 2014. Evaluation of the Effectiveness of Currently Utilized Diagnostic Protocols. Ray W. Herrick Laboratories, Purdue University.

technicians can cause uncertainty. Tests of eight different types of sensors on liquid and suction lines found accuracy ranging from of 1.1 +/- 0.6F for Type-K thermocouple clamps to 9.7 +/- 7.1F for insulated thermistors. The installation and removal of pressure measurement instruments can introduce faults. Tests of attachment and detachment of refrigerant hoses without EPA low-loss fittings, found 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and efficiency per attachment/detachment. Tests of economizer outdoor air leakage and coil blockage indicate the importance of checking and correcting obvious faults prior to installing measurement instruments to check refrigerant charge and airflow.

2.1.5 Laboratory Tests Identify New Measures

Laboratory tests have identified new measures such as reducing overventilation by optimizing damper position and reducing untended outdoor airflow by sealing around the economizer perimeter frame (under the hood). Based on tests of five economizers installed on four units, the average closed economizer damper outdoor airflow is 18 +/- 3% of total system airflow which meets or exceeds ASHRAE 62.1 minimum ventilation requirements for most buildings. Opening economizer dampers from 1-to-3-fingers provided outdoor airflow of 27 to 39% which far exceeds ASHRAE 62.1 minimum requirements and reduced EER*s by 20 +/- 3% compared to closed dampers. Tests with tape sealing the gap between economizer perimeter frame and cabinet (under the hood) found an average outdoor airflow reduction of 6 +/- 2% and improved EER*s of 5 +/- 2%.[32]

2.2 Laboratory Tests Improve Understanding of HVAC System Performance

Laboratory tests on commercial packaged units have improved the understanding of HVAC system performance under typical and part-load conditions as well as faulted conditions including low airflow, improper refrigerant charge, coil blockage, restrictions, non-condensables, economizer functionality, overventilation, and unintended outdoor air leakage. Examples from some of the laboratory tests are summarized below. Other examples are provided in **Section 4**.

[32] Average difference between unsealed versus sealed tests of 7.5-ton and 3-ton non-TXV and TXV units.

2.2.1 Sensible Cooling Capacity Performance

The purpose of commercial air conditioning systems is to control the temperature and humidity of conditioned building spaces and provide adequate outdoor air ventilation and filtration to maintain thermal comfort, health, and safety. While air conditioners provide dehumidification, thermostats on most buildings only control sensible temperatures. Sensible cooling capacity indicates whether or not the equipment can meet the cooling load and satisfy the thermostat setpoint which determines how long the cooling system operates and how much energy is used. Engineers designing buildings require accurate information regarding HVAC equipment sensible cooling capacity at various airflows (cfm) and indoor/outdoor temperature conditions. This information allows the engineer to correctly specify the size of HVAC equipment to meet design-day cooling loads based on building energy simulation models or ACCA Manual N calculations.[33] The California Energy Commission Title 24 standards do not set limits on the size of cooling or heating equipment, but they recommend equipment sizing of 120% for airflow (cfm) and cooling capacity and 125% for heating capacity based on the calculated load.[34] Manufacturers of packaged HVAC equipment provide tables of gross sensible and total cooling capacity ratings at various airflows (cfm) and indoor/outdoor temperature conditions. The manufacturer gross capacity ratings do not include correction factors of indoor drybulb temperatures or fan heat. **Table 1** provides the ACCA Manual N commercial cooling load calculations for spaces likely to be served by the 7.5-ton and 3-ton units tested in the laboratory. The design sensible cooling loads include ASHRAE 62.1 minimum outdoor air ventilation requirements for typical building occupancies. The manufacturer (Mfr) rated sensible cooling capacities are net (i.e., including correction factors indoor drybulb temperature and fan heat). At 95F OAT the equipment is 118 to 126% oversized. The average oversizing is 123 +/- 6% and comparable to the CEC recommended 120% oversizing. Green highlighting indicates capacity is greater than 105% of ACCA Manual N including ventilation loads and no less than 10% of non-faulted capacity. Yellow highlighting is used to indicate sensible capacity is between 100 and 105% of ACCA Manual N including ventilation loads and no less than 10% of non-faulted capacity. Red highlighting indicates sensible capacity is less than the ACCA Manual N including ventilation loads or less than 10% of non-faulted capacity.

[33] ACCA 2008. Manual N - Commercial Load Calculations for Small Commercial Buildings. Fifth Edition. ACCA http://www.acca.org/technical-manual/manual-n/

[34] CEC 2013. Nonresidential Alternative Calculation Method Reference Manual for the Building Energy Efficiency Standards. http://www.energy.ca.gov/2013publications/CEC-400-2013-004/CEC-400-2013-004-CMF.pdf

Table 1: ACCA Manual N Commercial Cooling Load Calculations at 82F, 95F, and 115F OAT for Spaces Served by 7.5-ton and 3-ton Packaged Air Conditioners and Manufacturer Rated Sensible Capacity and Size

OAT (F)	7.5-ton ACCA Manual N Sensible Cooling Load Btuh	3-ton ACCA Manual N Sensible Cooling Load Btuh	7.5-ton non-TXV Mfr Rated Sensible Capacity Btuh	7.5-ton TXV Mfr Rated Sensible Capacity Btuh	3-ton non-TXV Mfr Rated Sensible Capacity Btuh	3-ton TXV Mfr Rated Sensible Capacity Btuh	7.5-ton non-TXV Size per ACCA Manual N %	7.5-ton TXV Size per ACCA Manual N %	3-ton non-TXV Size per ACCA Manual N %	3-ton TXV Size per ACCA Manual N %
82	37,156	15,463	59,673	65,203	25,907	26,237	161	175	168	170
95	48,390	20,593	57,182	61,173	25,009	25,936	118	126	121	126
115	67,864	30,232	52,997	54,823	23,384	24,858	78	81	77	82

The laboratory tests provide sensible cooling minus ventilation loads which are included in the measurements. The ACCA Manual N sensible cooling load design values minus ventilation loads are provided in **Table 2**. These values are used to evaluate sensible cooling capacity for test results provided in **Section 4**.

Table 2: ACCA Manual N Commercial Cooling Load Calculations Minus Ventilation Loads at 82F, 95F, and 115F OAT for Spaces Served by 7.5-ton and 3-ton Packaged Air Conditioners

OAT (F)	7.5-ton ACCA Manual N Sensible Cooling Load Minus Ventilation Load (Btuh)	3-ton ACCA Manual N Sensible Cooling Load Minus Ventilation Load (Btuh)
82	35,978	14,939
95	45,024	19,097
115	61,132	27,240

2.2.2 Performance under Typical Conditions

Building energy simulation models require realistic steady-state cooling capacity and efficiency information under typical operating conditions. Steady-state laboratory tests of units immediately after taken "out of the box" prior to AHRI benchmark tests indicate 5 to 24% lower performance than AHRI ratings.[35] The AHRI testing process does not represent typical installed conditions (such as required economizers). The most notable deviation of the AHRI test protocol from field conditions is the external static pressure (ESP) on the system's supply air fan and tests with the economizer installed. A number of changes are made to systems to perform the AHRI test procedure. These typically include changes to the supply fan pulleys to reduce fan power, no economizer installed, sealing of the test unit cabinet to control leakage (increasing rated capacity), adding insulation to the cabinet base and, on occasion, modifying refrigerant charge to achieve manufacturer specifications regarding discharge pressure, suction pressure, suction

[35] AHRI benchmark tests attempt to replicate the manufacturer's published ratings within 95% of the cooling capacity and efficiency under the AHRI rating conditions.

temperature, liquid temperature, superheat, subcooling, or approach temperature. The as-delivered “out-of-box” test setup provided consistently lower steady-state application efficiency (EER*) results compared to results for the same system when tested under the AHRI setup and indoor/outdoor temperature conditions.[36] These findings were provided to the DEER team for use with the direct expansion (DX) HVAC system models or other building simulation models used to estimate or evaluate ex ante savings values.

Table 3 provides published AHRI EER ratings and laboratory test application EER* for the 7.5-ton and 3-ton non-TXV and TXV units with no economizer and economizer with closed dampers and 1-finger open. Tests were performed under typical field conditions and external static pressures which are different than ANSI/AHRI 340/360 or 210/240 conditions.[37] The average AHRI EER rating is 11.1 for all units. The average EER* is 8.8 with no economizer or 20% less than the average AHRI rating. The average EER* is 7.0 with economizer dampers closed or 37% less than the average AHRI rating. The average EER* is 5.6 with economizer dampers 1-finger open or 49% less than the average AHRI rating. These test results indicate that typical field application efficiencies of new units are much less than the AHRI EER ratings.

Table 3: AHRI EER Ratings and Laboratory Test Application EER* for 7.5-ton and 3-ton non-TXV and TXV Units with No Economizer and Economizer with Closed and 1-Finger Open Dampers at Typical Field Conditions

Unit	AHRI EER Rating	Test	No Econo EER*	Test	Econo Closed EER*	Test	Econo 1-Finger Open EER*
7.5-ton non-TXV	11.0	Run-3-20N95	8.1	3-295	6.2	3-2951	5.3
7.5-ton TXV	11.0	3T-75629575-NE3-Retest	8.8	T2-TRN-95-CE-DM	7.6	T2-TRN-95-1FER-DM	6.1
3-ton non-TXV	11.0	C-MF-75629575-NE3J	9.2	C-MF-75629575-E3J	6.4	C-MF-75629575-1E3J	5.0
3-ton TXV	11.2	L-75629575-NE3-SS	9.1	L-75629575-E3	7.7	L-75629575-1E3	6.1
Average	**11.1**		**8.8**		**7.0**		**5.6**
Average Impact			-20%		-37%		-49%

2.2.3 Part-load Performance

Building energy simulation models require part-load performance data to model HVAC packaged system energy efficiency performance. Part-load performance data for larger units (greater than 65,000 Btuh rated net cooling capacity) cannot be determined from manufacturers’

[36] Ibid.

[37] ANSI/AHRI 340/360 and 210/240 test conditions are 80F DB and 67F WB indoor and 95F OAT and 0.25 IWC ESP for 7.5-ton units and 0.15 IWC for 3-ton units. Tests in the table were performed at typical field conditions of 75F DB and 62F WB indoor and 95F OAT and ESP of 1.1 to 1.2 IWC for 7.5-ton non-TXV, 0.6 to 0.9 IWC for 7.5-ton TXV, 0.5 to 0.6 IWC for 3-ton non-TXV, and 0.7 IWC for 3-ton TXV.

data.[38] Correct assessment of part-load performance of larger systems requires cycling tests, as is done for smaller systems via the prescribed "C" and "D" tests in AHRI Standard 210/240. According to Federal minimum efficiency standards, as of January 2010, all commercial unitary HVAC units rated above 65,000 Btu per hour are required to be rated with the Integrated Energy Efficiency Ratio (IEER) test. IEER is a weighted average of steady-state efficiency values for various test conditions. This rating is not based on measured cycling losses, but rather uses an assumed loss curve that may or may not represent actual system performance.[39] Current part-load performance maps are based on typical cycling losses associated with smaller systems (since they are required to include the "C" and "D" tests as part of their seasonal efficiency rating, or SEER). The "C" and "D" laboratory tests on the 7.5-ton non-TXV packaged air conditioner found that the relative cycling losses for these large systems are more than double their smaller counterparts. Features added to smaller systems to control cycling losses that increase their SEER rating are not used on larger commercial systems (greater than 65,000 Btu/hr) since cycling losses do not impact efficiency ratings. Additional tests might be needed to obtain more representative estimates of typical large system cycling losses, the results of which will be used for building energy simulation software to develop DEER ex ante energy savings estimates for HVAC maintenance and installation measures.

2.2.4 Refrigerant Charge Performance

In order to model the impact of refrigerant charge faults, building energy simulation programs require data from tests performed with varying levels of refrigerant below or above the manufacturer recommended factory charge. Tests were performed on systems with refrigerant charge varying from -40 to +40% of factory charge on five (5) new obsolete stock R22 commercial packaged cooling systems (one 7.5-ton non-TXV, two 7.5-ton TXV, one 3-ton non-TXV and one 3-ton non-TXV).[40] These tests showed that cooling capacity and efficiency performance are less affected by system overcharge than assumed in DEER refrigerant charge measures. Refrigerant undercharging by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% reduced EER*s by 0 to 3%.

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[38] Manufacturers do not provide part-load performance data in terms of cycling loss curves for units greater than or equal to 65,000 Btu/hr. Expanded performance data on SEER-rated units (with cooling capacities less than 65,000 Btu/hr) can be used to estimate cycling-loss coefficients and, thus, project part-load operation that includes cycling losses.

[39] AHRI Standard 340/360-2007 uses indoor conditions of 80F drybulb and 67F wetbulb and the following outdoor drybulb conditions to calculate IEER ratings: 95F (100%), 71F (75%), 68F (50%), and 65F (25%). In January 2010 IEER replaced the Integrated Part Load Value (IPLV) as the part load energy efficiency descriptor for all commercial unitary products rated above 65,000 Btu/h. See http://www1.eere.energy.gov/buildings/appliance_standards/pdfs/ac_hp_rfi_noda.pdf. Also see ASHRAE Standard 90.1-2007, Energy Standard for Buildings Except Low-Rise Residential Buildings, October 2007.

[40] These systems were chosen based on what is most commonly seen in the market, availability of these discontinued but new off the shelf units, and on the fact that most maintenance activities are still performed on R-22 systems. Tests were performed at typical indoor conditions of 75F DB and 62F WB and various outdoor conditions, evaporator airflow rates, economizer damper positions.

2.2.5 Economizer Performance

Building energy simulation models have historically assumed 0 to 5% outdoor air fractions (OAF) with closed dampers and 95 to 100% OAF with open dampers in order to model economizer performance and energy savings. Laboratory tests of economizer damper outdoor airflow found fully closed OAF ranged from 12 to 30% and fully open OAF ranged from 30 to 78% of total evaporator airflow. For all tests, the average closed damper OAF was 18 +/- 3% and the fully-open damper OAF was 68 +/- 5%. Tests were conducted at a variety of indoor and outdoor conditions to cover the range of conditions imposed on systems in the field.[41] There are no standard industry performance tests of HVAC systems with economizers installed. There are HVAC system ratings and economizer ratings but no combined system ratings. From 1978 to 1988, economizers were required in California on units with rated cooling capacity greater than 134,000 Btuh.[42] From 1992 to 2012, economizers were required on units in California with rated cooling capacity greater than 75,000 Btuh.[43] ASHRAE 90.1 and California Energy Commission (CEC) Building Energy Efficiency Standards currently require economizers on units with rated cooling capacity greater than or equal to 54,000 Btu/hr.[44] Test data provide accurate minimum and maximum economizer outdoor airflow rates to improve DEER ex ante energy savings estimates for economizers.

Laboratory tests were performed on commercial packaged HVAC systems with and without economizers installed since economizers are code-required for systems with rated cooling capacity exceeding 54,000 Btuh. AHRI efficiency and cooling capacity ratings of packaged commercial HVAC system ratings are determined without economizers installed and economizer leakage classification ratings are based on tests with the perimeter frame sealed.[45] HVAC plus economizer combined system tests are required to evaluate energy efficiency impacts of maintenance faults including overventilation due to damper leakage and unintended leakage around the perimeter frame where the economizer is attached to the cabinet. Two important findings were discovered from HVAC/economizer combined system tests. The first important

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[41] Units were tested at indoor conditions of 75F DB and 62 F WB and outdoor conditions of 82F DB and 68F WB, 95F DB and 75F WB, and 115F DB and 80F WB.

[42] California Energy Commission. 1978. California Energy Commission Conservation Division Regulations Establishing Energy Conservation Standards for New Residential and New Nonresidential Buildings. http://www.energy.ca.gov/title24/standards_archive/CEC-400-1978-001.PDF. California Energy Commission. 1988. Building Energy Efficiency Standards. 1988 Edition. http://www.energy.ca.gov/title24/standards_archive/1988_standards/CEC-400-1988-001.PDF.

[43] California Energy Commission. 1992. Energy Efficiency Standards for Residential and Nonresidential Buildings http://www.energy.ca.gov/title24/standards_archive/CEC-400-1992-001.PDF, CEC-400-2012-004-CMF-REV2.pdf

[44] ANSI/ASHRAE/IES Standard 90.1-2010, Energy Standard for Buildings Except Low-Rise Residential Buildings, Section 6.5.1 Economizers. http://www.energy.ca.gov/title24/. CEC. 2013. Building Energy Efficiency Standards for Residential and Nonresidential Buildings currently require economizers on units with cooling capacities greater than or equal to 54,000 Btuh. CEC-400-2012-004-CMF-REV2. 2012.

[45] Perimeter frame leakage refers to leakage between the economizer frame assembly and the HVAC unit cabinet.

finding is economizer dampers at typical minimum positions provide 27 to 39% outdoor airflow when set at 1- to 3-fingers open which exceeds ASHRAE 62.1 minimum requirements and reduced EER*s by 20 +/- 3% compared to closed dampers. The second important finding is fully-open economizer dampers only provide 30 to 78% OAF (68 +/- 5% average for all tests) which is significantly less than fully-open OAF of 95 to 100% assumed by the HVAC industry or previous DEER evaluations.[46]

The Air Movement and Control Association (AMCA) define four classes of damper leakage at static pressure of 1 inch water column (IWC).[47]

- Class 1A) 3 cfm/ft^2,
- Class 1) 4 cfm/ft^2,
- Class 2A) 10 cfm/ft^2 (ASHRAE 90.1), and
- Class 3) 40 cfm/ft^2.

Laboratory tests of Class 2 dampers on economizers from three different manufacturers of ASHRAE 90.1 compliant economizers indicate leakage rates about 6.5 times higher than the AMCA Class 1A standard (i.e., 60 to 80 cfm/ft^2) due to perimeter frame leakage not included in AMCA testing. Tests consistently measured closed economizer damper 12 to 30% outdoor air fractions with evaporator airflow varying from 267 to 450 scfm/ton. Laboratory tests measured maximum outside airflow rates during economizer operation between 30 and 78% of total system airflow. As noted above, this is significantly different from the 0 to 5% closed damper airflow and 95 to 100% fully open economizer airflow assumed by the HVAC industry or past DEER analyses. These findings are significant and will impact energy efficiency measures both positively and negatively when included in DEER update analyses. Measures with a cooling impact (such as a lighting retrofit), would likely see the related cooling benefits increase as economizer cooling would decrease with reduced maximum airflow rates. Direct economizer repair or control measures might have predicted reductions in cooling benefits as revised outdoor airflow values are implemented. However, the predicted reductions in cooling benefits can be overcome by optimizing economizer minimum damper positions and sealing the perimeter frame to reduce untended outdoor leakage (under the hood) which will save both cooling and heating energy. Test data are provided for five different economizers on four packaged units from three manufacturers (two 3-ton and two 7.5-ton units). Preliminary laboratory tests conducted on 4 RTUs with 5 economizers indicate that approximately 13 to 41% of closed damper flow is from the perimeter or economizer/unit connection joint. This junction can be cost effectively sealed

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[46] Past DEER/EQuest economizer outdoor air (OA) leakage assumptions were 5% for closed dampers and 100% for fully open dampers. Laboratory tests of class 2 dampers on economizers from three different manufacturers of ASHRAE 90.1 compliant economizers indicate leakage rates of 13 to 30% or about 4.8 to 11.1 times higher than the AMCA Class 1A standard. Laboratory tests of Class 1A dampers indicate 17% OAF or 6.3 times greater than Class 1A. Higher OAF might be due to perimeter frame leakage not included in standard ACMA tests.

[47] AMCA. 2010. AMCA 511-10 (Rev. 8/12) Certified Ratings Program–Product Rating Manual for Air Control Devices. Table 3. pp. 13. www.amca.org. ANSI/AMCA 2012. Standard 500-D-12 Laboratory Methods of Testing Dampers for Rating. Air Movement and Control Association International, Inc. www.amca.org/store/item.aspx?ItemId=55

with UL-approved waterproof metal tape to improve space cooling and heating efficiency. Taping around the economizer frame reduced unintended outdoor airflow by 6 +/- 2% and improved EER*s by 5.4 +/- 2% when the damper is closed or open from 10 to 30%. Additional tests are critical to expand and confirm the range of savings opportunities from commercial HVAC maintenance measures related to economizer operation and installation.

The ASHRAE 90.1 mechanical subcommittee investigated economizer damper leakage described as follows:[48]

"The damper leakage for outside air dampers is only an issue on units when they are running in the unoccupied mode for heating or cooling. That means it is not an issue on a 24/7 operation and is only an issue in the buildings that have unoccupied heating and cooling. In the occupied mode the dampers are open for minimum ventilation air so leakage is a non-issue. In the unoccupied mode the leakage is only an issue when the fan is on for heating or cooling, but the fan is cycled in most applications so when the fan is off there is no leakage."

This statement is correct if dampers meet the AMCA 511 standard, no other leakage exists except damper edge and jamb leakage and minimum damper position meets ASHRAE 62.1 outdoor air requirements. As noted above, for units tested in the laboratory, HVAC/economizer system outdoor air leakage appears to be much higher than previously assumed economizer leakage when dampers are closed or partially open in the minimum position. Preliminary laboratory tests have also shown that economizers only provide approximately 60 to 65% of outdoor air when fully open. Industry publications identified two economizer leakage areas:[49]

1) "Jamb leakage" between damper blade ends and frame, and
2) "Edge leakage" between damper-blade edges.

Laboratory tests indicate a third economizer leakage area:

3) "Perimeter and Gap Leakage" between economizer perimeter frame and HVAC cabinet and holes or gaps in the economizer or damper assembly.

Low-leakage dampers are supplied with blade and jamb seals. The type of seal supplied causes significant differences in leakage rates. There can be a 10-to-1 difference in a damper supplied with mechanically locked seals and flexible metal jamb seals versus a damper supplied with no seals at all. Economizers with no perimeter seals can increase leakage by 50% or more when dampers are closed for either Class 1 or Class 1A dampers.[50] Preliminary laboratory tests indicate that HVAC/economizer system outside air leakage is much greater than previously assumed, and fully open economizer outdoor airflow is 25 to 35% less than previously assumed.

[48] D. Lord. 2010. Simplified Damper Leakage. ASHRAE 90.1 Mechanical Subcommittee.

[49] J. Knapp. 2007. Damper Leakage Rates–More Important than Ever. AMCA International Inmotion. pp. 19-21 (Fall 2007). www.amca.org.

[50] Ibid.

Economizer outdoor airflow and ventilation can have significant impacts on indoor air quality, thermal comfort, energy efficiency and energy use. The following measures can save cooling energy and heating energy by reducing overventilation and unintended outdoor air leakage.

- Adjust and optimize economizer minimum damper positions to meet ASHRAE 62.1 ventilation requirements when buildings are occupied to reduce unintended outdoor air leakage.
- Seal unintended outdoor air leakage (i.e., gaps) around the perimeter of the economizer where it attaches to the cabinet (requires removal of economizer hood prior to sealing perimeter gaps with UL-181 metal tape under the hood).
- Reducing cabinet leakage by replacing stripped or missing screws used to secure cabinet panels to the unit.

Laboratory tests of these measures indicate energy efficiency improvements of 5 to 25% or more. These measures should be piloted in HVAC maintenance programs with proper training and pre/post measurements to ensure successful implementaton and verification of savings.

2.2.6 Field Measurement Instruments

Laboratory tests of field measurement instruments was performed to evaluate instrument accuracy, methods of attaching instruments to units, and procedures used to make measurements (i.e., time for unit or instrument to reach equilibirum in order to obtain accurate measurements). Field instrument testing was performed to evaluate how FDD might be impacted by field instrument accuracy, methods, and procedures.

- Evaporator airflow field measurement instruments were tested using four pitot-tube arrays and digital pressure gauges compared to the Code Tester measurements.[51] The measured accuracy was 10 +/- 1% of laboratory Code-Tester airflow measurements.[52]
- Refrigerant tube temperature measurement instruments were tested by soaking temperature sensors in the outdoor chamber to reach equilibrium before attaching 8 sensors to suction and liquid refrigerant tubes at different locations in the system (with and without insulation as applicable) to emulate best and worst measurement methods. Tests were conducted with eight sensors on liquid and suction lines with the following results. Suction line measurements tube temperatures were 25 to 40F less than OAT and liquid line temperatures

[51] Banks, E. Sills, C. Graves, C. 2002. Airflow Traverse Comparisons Using the Equal-Area Method, Log-Tchebycheff Method, and the Log-Linear Method, and Including Traverse Location Qualification. http://www.orau.org/ptp/PTP%20Library/library/Subject/stack%20sampling/airflow_traverse.pdf. The "code tester" is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92).

[52] The "code tester" is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92). Pitot-tube array measurements had a 200 cfm offset which could be corrected with additional testing. Additional tests of the pitot-tube array on other RTUs need to be performed to determine if measurements are always high.

were 8 to 12F above OAT. Average Type-K thermocouple clamp probes had accuracy of 1.1 +/- 0.6F on suction lines at 115F OAT. Some Type-K clamp probes had suction line accuracy of 6.8 +/- 1.0F when tested at 115F OAT. Type-K insulated bead probe accuracy was 10.7 +/- 3.3F, insulated cylindrical thermistor accuracy was 9.7 +/- 7.1F, and clamp thermistor accuracy was 5.4 +/- 2.1F. Differences in accuracy were attributable to variations in design (i.e., sensor, clamps, thermal contact, insulation, etc.) and manufacturing (quality of materials, fit, finish, operability. durability, etc). Tests indicated it can take 5 to 10 minutes or longer for sensors to reach steady-state and correctly measure air and refrigerant temperatures. Not allowing sensors to reach steady-state can cause inaccurate measurements. Field technicians are under pressure to complete work as quickly as possible.

- Refrigerant pressure sensors were tested by placing pressure manifolds with refrigerant hoses inside an oven at 130F to emulate typical field-service conditions and then testing the same pressure sensors at low, medium and high pressures for R-22 and R-410A refrigerants in a repeatable manner using a test bench. The average difference between laboratory and digital pressure measurement instruments was 0.57 +/- 0.24% based on measurements at ten different pressures with 15 instruments from 6 manufacturers. The average difference between laboratory and analog pressure measurement instruments was 1.76 +/- 0.57% based on measurements at ten different pressures with 7 instruments from 2 manufacturers.
- Refrigerant pressure measurement impacts were tested by attaching and detaching refrigerant hoses without EPA low-loss fittings to suction and liquid service pressure valves 10 times for 6 cycles or 60 total tests to evalaute loss of refrigerant and application efficiency impacts due to technicians hooking up refrigerant hoses to HVAC units over the effective useful life of the units. Based on 60 total tests, the average factory charge impact was 0.45 +/- 0.05% loss of refrigerant charge per test and the average application efficiency impact was 0.2% per test.
- Standard fan belts and notched v-belts were tested by varying the alignment and tension of the belts from manufacturer recommended alignment and tension. Tests were performed by varying fan belt tension and alignment using manual tools and laser-guided measurement tools. Belts were tested with proper tension and alignment, as well as loose and tight tension and misalignment of 0.25 and 0.375 inches. Out-of-box tests indicated belt tension was looser than manufacturer recommendations and belts were properly aligned. The EER*s improvement for the 7.5-ton non-TXV RTU3 from worn standard v-belt to new notched v-belt was 0.8%. Preliminary tests indicated belt tension and alignment did not have a significant impact on FDD, airflow, cooling capacity, or efficiency.
- Refrigerant charge recovery, evacuation, and factory recharge were tested in the field and in the laboratory using industry or manufacturer recommended practices. Field measurements of 35 units found an average difference between recovered and pre-existing refrigerant charge of 15.1 +/- 3.2% corresponding to an average EER*s impact of 8.9 +/- 3.9%.

3 LABORATORY TEST METHODS AND PLANS

The laboratory test methods followed the ANSI/ASHRAE Standard 37.[53] Intitial tests were performed to evaluate the "out-of-box" as-purchased performance with factory fan speed and refrigerant charge. After completing the initial tests, refrigerant was recovered into reclaim tanks, accurately weighed and each refrigerant circuit was evacuated below 500 microns of mercury (µHg) held at or below 1000 µHg for 30 minutes, before weighing in the factory refrigerant charge (ASHRAE 2010).[54] In order to perform the AHRI standard test procedure, a number of changes were made to each unit including installing larger diameter supply fan pulleys on some units to achieve AHRI airflow and ESP requirements, sealing the cabinet to reduce leakage, adding insulation to the cabinet base, or modifying refrigerant charge to achieve published ratings. The AHRI verification tests were performed per ANSI/AHRI Standard 210/240 or 340/360 at standard rating conditions to verify each unit was within 95% of the published AHRI ratings for performance ratios and cooling capacities.[55]

After intial tests were performed, each unit was subsequently tested at non-standard application conditions to emulate typical field conditions in the State of California.[56] Additional tests were performed on each unit with and without economizers installed and outdoor air damper positions varying from closed to fully open to evaluate proper ventilation to meet ASHRAE 62.1 and the impact of overventilation on application efficiency.[57] Economizer tests were also performed with the gap between the economizer perimeter frame and cabinet sealed with tape to evaluate unintended outdoor air leakage. Tests were performed on each unit to evaluate the application energy efficiency impacts of HVAC maintenance faults by varying refrigerant charge from 60 to 140% of factory charge, evaporator blockage from 5 to 50%, condenser blockage from 5 to 80%, airflow from 65 to 110%, economizere damper positon with unsealed and sealed perimeter, restrictions, non-condensables, and multiple faults. After each unit was tested at non-standard conditions with single or multiple faults, it was necessary to re-establish factory conditions and the baseline. Re-establishing the baseline after multiple refrigerant charge additions or removals or non-condensables involves recovery of refrigerant charge and evacuation to below 500 micron mercury (µHg) vacuum held at or below 1000 microns for 30 minutes and weighing in the factory charge (ASHRAE 2010).

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[53] ANSI/ASHRAE Standard 37-2009. Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment.

[54] ASHRAE. 2010. ASHRAE Handbook-Refrigeration. Page 8.2. Table 1. American Society of Heating Refrigeration and Air Conditioning Engineers, Inc. Carrier Corporation. 2010. Commercial Packaged Engineering Standard Work Procedure: System Evacuation and Dehydration. Carrier A United Technologies Company. JB 2007. Deep Vacuum: Its Principle and Application. JB Industries, Inc. www.jbind.com.

[55] ANSI/AHRI Standard 210/240 or 340/360 rated conditions at steady-state operation were performed at OAT of 95ºF [35C] drybulb and return air temperature of 80ºF [26.7C] dry bulb and 67ºF [19.4C] wet bulb.

[56] Non-standard application conditions at steady-state operation were performed at ambient OAT of 95F, 115F, 82F, and 55F drybulb and return air temperature of 75F drybulb and 62F wetbulb.

[57] ANSI/ASHRAE 2010. ANSI/ASHRAE 62.1-2010. Standard Ventilation for Acceptable Indoor Air Quality.

The laboratory-based test results are reported using the "application efficiency" defined as the "application rating" in ANSI/AHRI 210/240 and ANSI/AHRI 340/360.[58] The application energy efficiency ratio (EER*) is calculated as cooling capacity divided by total electric power. The application sensible energy efficiency ratio (EER*s) is calculated as the sensible cooling capacity divided by total electric power. The EER*s is reported to indicate how efficiently the unit operates based on sensible drybulb thermostat settings which control air conditioning operational time. The application field conditions include non-standard return/outdoor air temperatures and external static pressure conditions appropriate to California climate conditions and economizers installed with and without maintenance faults. Laboratory test results were also used to evaluate the accuracy of manuacturer and generic CEC refrigerant charge and ariflow FDD protocols under non-faulted and faulted test conditions they were not intended to diagnose. The laboratory tests results of FDD protocols are provided to understand the limitations of using refrigerant charge and airflow FDD protocols to perform comprehensive HVAC maintenance services. Laboratory tests were performed using the following test conditions.

- Outdoor temperatures for HVAC maintenance fault tests (DB/WB): 82/62, 95/75, 115/80,[59]
- Indoor temperatures for HVAC maintenance fault tests (DB/WB): 70/57, 75/62, 80/67,
- Economizer outdoor temperature tests (DB/WB): 70/60, 65/57, 60/54, 55/51,[60]
- Airflow (cfm/ton): 250, 300, 350, 400, 450 cfm/ton,[61] and
- External static pressure (IWC): 0.15 to 2.0.[62]

Initial test equipment set-up can take 24 to 48 hours and removal of equipment can take 12 to 24 hours. Some of the tests were driven by findings discovered during the course of testing. Thus, not all tests have been conducted across all tested units. The test equipment schematic for a single-compressor packaged unit is shown in **Figure 1**. Refrigerant-side pressure/temperature measurements are installed before the expansion device, evaporator outlet, compressor suction, compressor discharge and condenser outlet. Setup requires digitally-controlled precision

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[58] Application ratings are based on tests performed at application conditions. Standard ratings are based on tests performed at standard rating conditions including airflow and external static pressure at 95F OAT and 80F drybulb and 67F wetbulb return temperatures. ANSI/AHRI 340/360-2007 Standard for Performance Rating of Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment. ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment Standard 210/240.

[59] Outdoor wetbulb temperatures are defined in the tests to measure the impact of economizer outdoor air leakage on total cooling capacity.

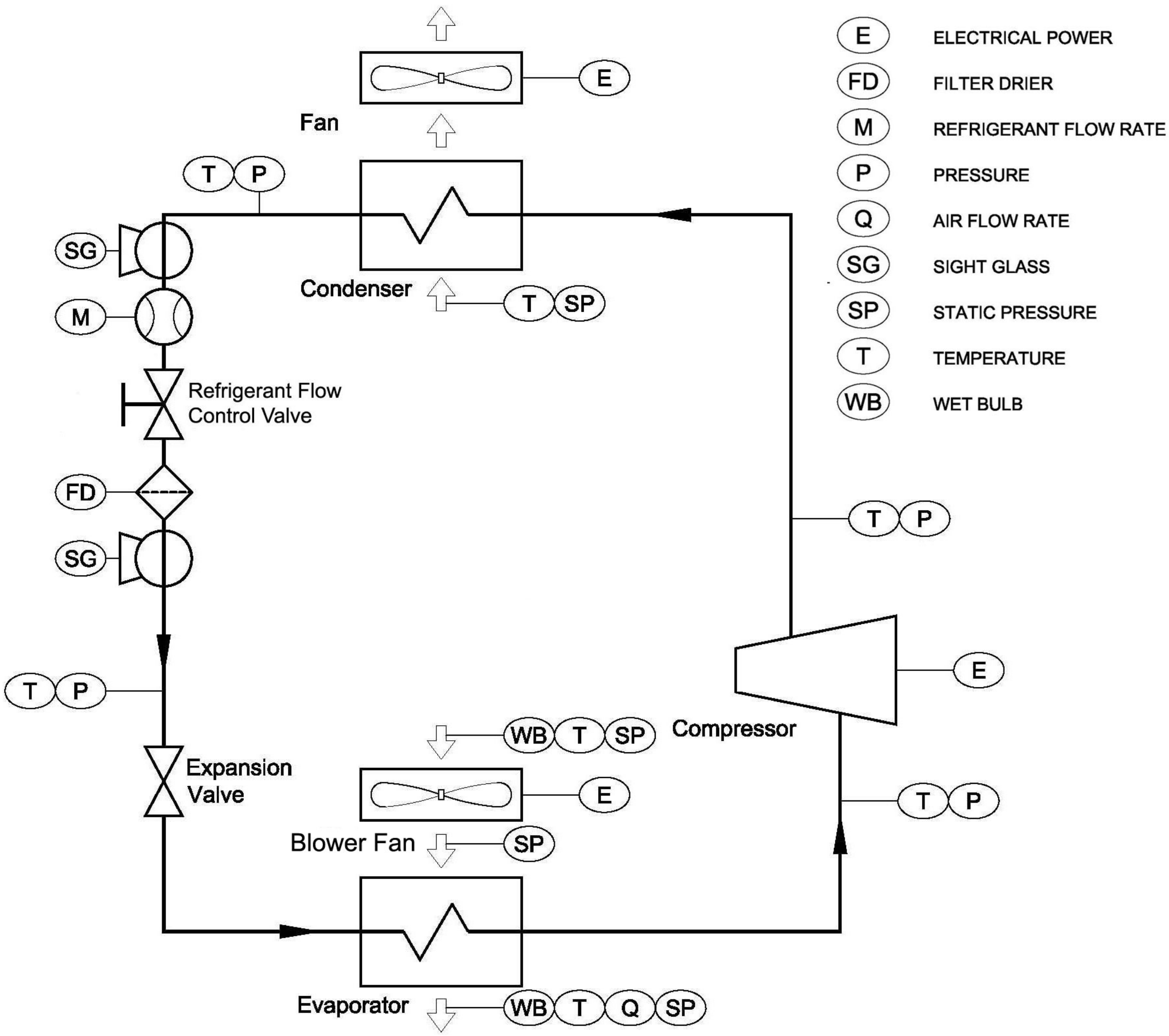
[60] Economizer outdoor temperature test conditions are selected to measure system EER*s and cooling capacity without compressor operation and with 1st-stage and 2nd-stage operation (for multi-compressor systems). The tests are performed to evaluate change-over settings and performance based on outdoor air provided by economizers.

[61] Airflow targets varied due to limitations of blower-drive system, motor, and external static pressure setup.

[62] External static pressure for each test varied depending on speed (rpm), airflow (cfm), and horsepower of the blower-drive system. Test conditions were based on field data available in the "Small HVAC Problems and Potential Savings Reports," October 2003, California Energy Commission 500-03-082-A-25. http://www.energy.ca.gov/2003publications/CEC-500-2003-082/CEC-500-2003-082-A-25.PDF.

louvered dampers installed on supply and return ducts to control inlet static pressure (ISP) and external static pressure (ESP) similar to in-situ conditions. Controlling inlet and total static pressure provided realistic test conditions to measure performance when varying airflow, fan speed and economizer outdoor-air damper positions from closed to fully open.

Figure 1: Test Equipment Schematic



3.1.1 Uncertainty of Laboratory Measurements

Figure 2 provides the uncertainty of laboratory test measurements calculated using the Engineering Equation Solver for the Intertek baseline test of the 3-ton non-TXV RTU4.[63] The average uncertainty for the laboratory tests of sensible capacity and application sensible efficiency (EER*s) were 0.6% and 0.8% respectively. Steady-state test data were collected every 4 seconds for 15 to 30 minutes per test.

Figure 2: Uncertainty of Laboratory Test Measurements

Solution

Uncertainty Results | Solution

Unit Settings: Eng F psia mass deg

Variable±Uncertainty	Partial derivative	% of uncertainty
EERs = 5.162±0.04083 [Btu-cfm-lb$_m$/lbm-ft^3-W]		
B1 = 61.99±0.04 [F]	∂EERs /∂B1 = 0.006254	0.00 %
B2 = 55.61±0.04 [F]	∂EERs /∂B2 = 0	0.00 %
CFM1 = 1090±5.45 [cfm]	∂EERs /∂CFM1 = 0.004735	39.93 %
P1 = 14.33±0.1433 [psia]	∂EERs /∂P1 = -0.008118	0.08 %
P2 = 14.34±0.1433 [psia]	∂EERs /∂P2 = 0	0.00 %
Power = 3549±17.75 [W]	∂EERs /∂Power = -0.001455	39.98 %
T1 = 75.03±0.04 [F]	∂EERs /∂T1 = 0.3186	9.74 %
T2 = 58.93±0.04 [F]	∂EERs /∂T2 = -0.3206	9.87 %
V1 = 14.0276±0.0070 [ft^3/lb$_m$]	∂EERs /∂V1 = -0.3680	0.40 %
NetAirSensible = 18320±112.3 [Btu-cfm-lb$_m$/lbm-ft^3]		
B1 = 61.99±0.04 [F]	∂NetAirSensible /∂B1 = 22.2	0.01 %
B2 = 55.61±0.04 [F]	∂NetAirSensible /∂B2 = 0	0.00 %
CFM1 = 1090±5.45 [cfm]	∂NetAirSensible /∂CFM1 = 16.8	66.53 %
P1 = 14.33±0.1433 [psia]	∂NetAirSensible /∂P1 = -28.81	0.14 %
P2 = 14.34±0.1433 [psia]	∂NetAirSensible /∂P2 = 0	0.00 %
Power = 3549±17.75 [W]	∂NetAirSensible /∂Power = 0	0.00 %
T1 = 75.03±0.04 [F]	∂NetAirSensible /∂T1 = 1131	16.23 %
T2 = 58.93±0.04 [F]	∂NetAirSensible /∂T2 = -1138	16.44 %
V1 = 14.0276±0.0070 [ft^3/lb$_m$]	∂NetAirSensible /∂V1 = -1306.0189	0.66 %

63 Klein, S.A. 2016. Engineering Equation Solver V10.,039, www.fchart.com.

Table 4 through **Table 8** provide the status of laboratory tests completed on the 7.5-ton non-TXV RTU3, 7.5-ton TXV RTU1 and RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4.

Table 4: Tests for Manufacturer #1 R-22 7.5-ton non-TXV, 2-Circuit (2 circuit) RTU3

Test	Type	Status	Section
Out-of-Box and Cycling Tests (1st-stage only and both compressors)	Vertical	Finished	4.1.1
Refrigerant Charge -20 to +60% (in 20% intervals) of factory charge (1st set of tests)	Vertical	Finished	4.1.8
Out-of-Box and Cycling Tests (1st-stage only and both compressors)	Vertical	Finished	4.1.1
Measurement Instruments (remainder were tested on horizontal setup)	Vertical	Finished	4.5
AHRI Verification	Horiz.	Finished	4.1.2
Manufacturer Refrigerant Charge Diagnostics	Horiz/Vert	Finished	4.1.3
Economizer Damper Leakage Tests at 55F (C, 1, 2, 3, O) Economizer #4	Horiz.	Finished	4.1.4
Economizer Damper Tests at 95F (C, 1, 2, 3, O) Economizer #4	Horiz.	Finished	4.1.5
Economizer 55 to 70F OAT, No-1-2-compressors Economizer #4	Horiz.	Finished	4.1.6
Airflow 100%, 83%, 68% of 400 scfm/ton, at 82, 95, 115F closed & 1-finger open dampers	Horiz.	Finished	4.1.7
Restrictions: install service valve upstream of filter drier	Horiz.	Finished	4.1.11
Non-Condensables (0.33% nitrogen per factory charge)	Horiz.	Finished	4.1.12
Economizer Outdoor Airflow Damper Leakage Tests with and without perimeter tape (C, 1, 2, 3, O) at 55F OAT and no compressors (control ISP & ESP)	Horiz.	Finished	4.1.4
Economizer Damper at 95F OAT with and w/o perimeter tape (C, 1, 2, 3, O) (dampers control ISP & ESP)	Horiz.	Finished	4.1.5
Refrigerant Charge -40 to +40% (in 10% intervals) of factory charge at 95F OAT and 250, 330 (83% airflow) (control ISP & ESP) (2nd set of tests)	Horiz.	Finished	4.1.8
Evaporator Coil Blockage (30 to 50%) (supply/return dampers control ISP & ESP) reduce evaporator airflow by 8 to 18%	Horiz.	Finished	4.1.9
Condenser Coil Blockage (5 to 80%) (supply/return dampers control ISP & ESP) increase discharge pressure by 2 to 40%	Horiz.	Finished	4.1.10
Multiple Fault Tests low airflow, 2 finger-open damper, untaped economizer perimeter/gaps, 50% blocked coils, -10% refrigerant charge (control ISP & ESP)	Horiz.	Finished	4.1.13
Measurement Instruments and Refrigerant Hose Attach/Detach Tests	Horiz.	Partial	4.5

Table 5: Tests for Manufacturer #2 R-22 7.5-ton TXV 2-Circuit (2 compressor) RTU1

Test	Type	Status	Section
Out-of-Box 3HP fan	Horiz.	Finished	4.2.1
AHRI Verification 2HP fan	Horiz.	Finished	4.2.2

Table 6: Tests for Manufacturer #2 R-22 7.5-ton TXV, 2-Circuit (2 compressor) RTU2

Test	Type	Status	Section
Out-of-Box and Cycling Tests (both compressors)	Horiz.	Finished	4.2.1
AHRI Verification	Horiz.	Finished	4.2.2
Manufacturer Refrigerant Charge Diagnostics	Horiz.	Finished	4.2.3
Economizer Outdoor Airflow Damper Leakage Tests at 55F OAT with no compressors with and w/o perimeter tape (C, 1, 2, 3, O) (supply/return dampers control ISP & ESP) Economizer #0, #1, #2	Horiz.	Finished	4.2.4
Economizer Damper at 95F OAT with and without perimeter tape (C, 1, 2, 3, O) (supply/return dampers control ISP & ESP) Economizer #1, #2	Horiz.	Finished	4.2.5
Airflow Standard Static 2-hp Fan Motor 108%, 100%, 87%, 75%, 63% of 400 cfm/ton (dampers control ISP & ESP)	Horiz.	Finished	4.2.6
Airflow High Static 3-hp Fan Motor 100%, 88%, 80%, 76%, 63% of 400 cfm/ton (dampers control ISP & ESP)	Horiz.	Finished	4.2.6
Refrigerant Charge -40 to +40% (+/-5% or 10% intervals) of factory charge at 82F, 95F and 115F OAT and 250, 300, 350, 400 cfm/ton (control ISP & ESP)	Horiz.	Partial	4.2.7
Condenser Coil Blockage (5, 10, 20, 30, 40, 50, 60, 70, 80%) (supply/return dampers control ISP & ESP) increase discharge pressure by 2 to 33%	Horiz.	Finished	4.2.8
Evaporator Coil Blockage (base, 30 to 80%) (supply, return dampers to control ISP/ESP) decrease evaporator airflow by 8 to 18%	Horiz.	Finished	4.2.9
Restrictions (supply/return dampers control ISP & ESP) Multiple Fault Tests	Horiz.	Finished	4.2.10
Non-Condensables 0.25 to 1% nitrogen per factory charge both circuits (supply/return dampers control ISP & ESP) Multiple Fault Tests	Horiz.	Finished	4.2.11
Multiple Fault Tests (supply/return dampers control ISP & ESP)	Horiz.	Finished	4.2.12
Measurement Instruments and Refrigerant Hose Attach/Detach Tests	Horiz.	Partial	4.5.1

Table 7: Tests for Manufacturer #1 R-22 3-ton non-TXV 1-Circuit (1 compressor) RTU5

Test	Type	Status	Section
Out-of-Box	Horiz.	Finished	0
AHRI Verification A, B, C and D	Horiz.	Finished	4.3.2
Manufacturer Refrigerant Charge Diagnostics	Horiz.	Finished	4.3.3
Economizer Outdoor Airflow Damper Leakage Tests with and without perimeter tape (C, 1, 2, 3, O) at 55F OAT and no compressors (supply/return dampers control ISP & ESP) Economizer #5	Horiz.	Finished	4.3.4
Economizer Damper at 95F with and w/o perimeter tape (C, 1, 2, 3, O) (dampers control ISP & ESP) Economizer #5	Horiz.	Finished	4.3.5
Refrigerant Charge -40 to +40% (+/-10% intervals) of factory charge at 82F, 95F and 115F OAT and 250 to 450 cfm/ton (supply/return dampers control ISP & ESP)	Horiz.	Partial	4.3.6
Evaporator Coil Blockage (30 to 80%) (supply/return dampers control ISP & ESP) reduce evaporator airflow by 1 to 13%	Horiz.	Finished	4.3.8
Condenser Coil Blockage (5 to 80%) (supply/return dampers control ISP & ESP) increase discharge pressure by 2 to 30%	Horiz.	Finished	4.3.7
Multiple Fault Tests (supply/return dampers control ISP & ESP)	Horiz.	Finished	4.3.9
Measurement Instruments and Refrigerant Hose Attach/Detach Tests	Horiz.	Partial	4.5

Table 8: Tests for Manufacturer #3 R-22 3-ton TXV 1- Circuit (1 compressor) RTU4

Test	Type	Status	Section
Out-of-Box	Horiz.	Finished	4.4.1
AHRI Verification A, B, C and D	Horiz.	Finished	4.4.2
Manufacturer Refrigerant Charge Diagnostics	Horiz.	Finished	4.4.3
Economizer Outdoor Airflow Damper Leakage Tests at 55F OAT and no compressors with and without perimeter tape (C, 1, 2, 3, O) economizer mfr #6 (dampers control ISP & ESP)	Horiz.	Finished	4.4.4
Economizer Damper at 95F OAT with and without perimeter tape (C, 1, 2, 3, O) (supply/return dampers control ISP & ESP) Economizer manufacturer #6	Horiz.	Finished	4.4.5
Refrigerant Charge -40 to +40% (+/-10% intervals) of factory charge at 82F, 95F and 115F OAT and 250 to 450 cfm/ton (supply/return dampers control ISP & ESP)	Horiz.	Partial	4.4.6
Measurement Instruments and Refrigerant Hose Attach/Detach Tests	Horiz.	Partial	4.5

4 LABORATORY TEST RESULTS

Laboratory tests were performed over a range of operating conditions, including standard rating conditions specified in AHRI 210/240 and 340/360 standards and non-standard application conditions to emulate typical field conditions in California based on WO32 field data observations. The main differences between AHRI standard rating conditions and non-standard field conditions are lower return air temperatures (75F drybulb versus 80F), higher static pressure causing increased indoor fan power, and higher outdoor air leakage due to economizer, relief damper and/or cabinet outdoor air leakage. Tests performed at non-standard conditions reduced efficiency by 40 to 50%. Typical external static pressure (ESP) associated with the air distribution is 0.5 inches water column (IWC) for 3-ton units and 1.2 IWC for 7.5-ton units and these ESP values are five times higher than the AHRI minimum external resistance of 0.10 for 3-ton systems and 0.25 for 7.5-ton units.[64] Higher external static pressure causes fan power to increase by 63 to 85% which reduces efficiency by 5 to 8%.

For most tests, the configuration of test units included either economizers or outside air dampers. As noted above, economizers are required by building codes for tested units with rated cooling capacities greater than 54,000 Btu per hour (Btuh).[65] Economizers are also found on smaller units not required by building codes. Economizers and outside air dampers are never included in AHRI tests. The main reason is AHRI tests provide “appliance” ratings for various cooling systems. Economizers and/or outside air dampers are considered system components rather than part of the basic cooling “appliance.” A second reason for not including these components in AHRI tests is the difficulties they impose on testing – as was experienced in this effort causing unbalanced airflows between indoor and outdoor chambers during tests which causes problems

[64] Field measurements found average ESP of 0.5 to 0.6 IWC for 3-ton units and 1.1 to 1.3 IWC for 7.5-ton units.

[65] The 2013 CEC Building Energy Efficiency Standards required economizers on units with cooling capacity greater than 54,000 Btuh (CEC. 2012. Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC−400−2012-004-CMF-REV2). The 2010 California Building Energy Efficiency Standards required economizers on units with cooling capacities greater than 75,000 Btuh (CEC. 2008. Building Energy Efficiency Standards for Residential and Nonresidential Buildings CEC-400-2008-001-CMF).

obtaining steady-state conditions. However, economizers and outdoor air dampers are considered necessary components for inclusion in tests of commercial systems. This effort examined the operation and system faults associated with units in under typical operating configurations and conditions. There were concerns that excluding these components would not provide sufficiently realistic test conditions. Test results demonstrated this was an accurate concern.

The typical design rating for a cooling system is the energy efficiency ratio, or EER. For systems with a rated cooling capacity exceeding 65,000 Btuh, this is the standard published performance rating. For smaller systems, the seasonal energy efficiency ratio, or SEER, is the standard performance rating. The rated EER for smaller units is also published based on AHRI tests, but is more commonly referred to as the EER_A rating (EER at the “A” rating point of 95F outdoor dry bulb temperature and 80F dry bulb and 67F wet-bulb cooling coil entering air conditions). The EER rating for larger units are also based on these test conditions.

Application ratings at non-standard conditions are not obtained under the ANSI/AHRI Standard 37 setup configuration or the ANSI/AHRI 210/240 or 340/360 standard rating conditions. Application ratings include non-standard return air conditions, external static pressures, economizers or outdoor air dampers installed, and typical faults such as low airflow, refrigerant charge or condenser and evaporator coil blockage. With non-standard setup configurations, higher static pressure and lower return air conditions, the unit operating efficiency is reported as an application rating either as an EER* or SEER* representing the non-standard setup configuration and test conditions. The efficiency value provided by the EER* value is based on the cooling capacity delivered by the distribution system (Btuh) divided by the total power usage (Watts). The application cooling capacity rating equals the cooling capacity based on the difference between return and supply air stream enthalpy entering and leaving the unit divided by the total power used by the unit. It treats all cooling of outside air regardless of the source as a system capacity loss. Test results are also presented in terms of the application sensible efficiency referred to as EER*s. The EER*s is reported to indicate how efficiently the unit operates based on sensible drybulb thermostat settings which control operational run time. Whenever possible, system tests attempt to differentiate between proper ventilation to meet ASHRAE 62.1 and overventilation on system efficiency. Since overventilation and unintended outdoor air leakage are regarded as system faults, the impact on system efficiency is quantified under the EER* or EER*s definition.

4.1 Test Results for 7.5-ton Non-TXV Packaged HVAC RTU3

One 7.5-ton multiple fixed-orifice non-TXV packaged HVAC model (RTU3) was tested in the laboratory per the ANSI/AHRI 340/360 test procedure. RTU3 uses R22 refrigerant and was shipped with a 1.5 horsepower (HP) blower motor, forward-curved centrifugal blower wheel with 1” wide blades and 15” diameter x 15” width. RTU3 has two compressors, each compressor has a separate refrigerant circuit, and each circuit is equipped with multiple fixed-orifice expansion valves on the header of each circuit at the evaporator inlet. The manufacturer factory charge is 7.6 lbs in circuit 1 and 8.1 lbs in circuit 2. The unit was shipped from the factory with motor sheave at 3 turns out from maximum fan speed setting, 7 inch diameter fan pulley, and fan speed of 969 revolutions per minute (rpm). The manufacturer installation, start-up, and service instructions indicate that the motor sheave is typically factory set at 5 turns to provide 840 RPM fan speed.[66] RTU3 was tested in vertical and horizontal configurations.

4.1.1 Out-of-Box Tests for 7.5-ton non-TXV RTU3

The 7.5-ton non-TXV RTU3 was tested in the "out-of-the-box" as-purchased condition in vertical and horizontal configurations at 95F and 82F OAT and 80F DB and 67F WB indoor temperatures. The out-of-box fan speed for RTU1 with the 3-hp fan motor was 1047 rpm with a 7-inch diameter pulley and motor sheave at 3 turns open. **Table 9** provides out-of-box tests for RTU3. The 2-compressor total EER* tests were 24 to 27% less than the rated EER (see 1-2A and 1-2B out of box baseline). The one-compressor tests were 45 to 50 less efficient than the rated EER (1-1A and 1-1B out of box baseline). The out-of-box fan power was two times greater than the AHRI verification test (see **Table 13**).[67] The AHRI tests were performed at 0.25 IWC ESP with a 10-inch diameter pulley operating at 831 rpm (6 turns) which reduces fan power by 50% producing less fan heat and more cooling capacity.

Table 9: Out-of-Box Tests for 7.5-ton non-TXV RTU3 with 7" Diameter Blower Pulley and No Economizer in Vertical Configuration

Test	C1/C2 Charge %	Comp #	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER	Total EER*	ΔEER*	Sensible EER*s
1-2A Out of Box Baseline	100/100	2	0.893	400	1,720	8,987	75,548	11.0	8.41	-24%	5.92
1-2B Out of Box Baseline	100/100	2	0.897	401	1,740	7,845	76,834	13.5	9.79	-27%	6.72
1-1A Out of Box Baseline	100/100	1	0.877	400	1,720	5,684	34,255	11.0	6.03	-45%	3.60
1-1B Out of Box Baseline	100/100	1	0.883	400	1,720	5,103	34,761	13.5	6.81	-50%	4.22

The manufacturer is not required to publish cycling test data per the ANSI/AHRI Standard 210/240 which only applies to air-conditioning equipment with rated cooling capacities less than 65,000 Btu/hour. Cycling tests were performed at the request of the DEER DMQC team to evaluate part-load analysis for building energy simulations. The tests were performed with the 7-inch pulley operating at 969 rpm producing 0.88 IWC ESP typical of field conditions. **Table 10** provides out-of-box EER and SEER cycling test data for the 7.5-ton non-TXV unit with the 1st-stage circuit 1 compressor operating. The average SEER* was 6.69 based on three SEER tests and average cyclic degradation coefficient (Cd) was 0.089.[68]

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66 Carrier 2005. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Gas Heating/Electric Cooling Units. Installation, Start-up, and Service Instructions. Form 48HJ-32SI. Fig. 57 – Cooling Charging Charts. http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48hj-32si.pdf.

67 Fan power is 1720 to 1740 W versus 850 W for AHRI verification test.

68 Cycling degradation coefficient (C_d) measures the efficiency loss due to cycling of units as determined in Appendices C and D of ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-

Table 10: Out-of-Box EER* and SEER* Cycling Tests for 7.5-ton non-TXV RTU3 with Circuit 1 Compressor and No Economizer in Vertical Configuration

Test	ESP IWC	Airflow scfm/ton	EER Rated	SEER Rated	EER* Measured	SEER* Measured	C_d
1-1A Out of Box Baseline	0.88	400	11.0		6.03		
1-1B Out of Box Baseline	0.89	400	13.5		6.81		
Run 1-1C&D Out of the Box Baseline Cycle #1						7.10	0.084
Run 1-1C&D Out of the Box Baseline Cycle #2						6.51	0.089
Run 1-1C&D Out of the Box Baseline Cycle #3						6.48	0.097
Run 1-1C&D Out of the Box Baseline Cycle #4						6.52	0.086
Average						**6.69**	**0.089**

Table 11 provides out-of-box EER* and SEER* cycling tests for the 7.5-ton non-TXV unit with both compressors operating. The average SEER* was 9.38 based three SEER tests and the cyclic degradation coefficient (C_d) was 0.084.

Table 11: Out-of-Box EER* and SEER* Cycling Tests for 7.5-ton non-TXV RTU3 with Both Compressors and No Economizer in Vertical Configuration

Test	ESP IWC	Airflow scfm/ton	EER Rated	SEER Rated	EER* Measured	SEER* Measured	C_d
1-2A Out of Box Baseline	0.89	400	11.0		8.41		
1-2B Out of Box Baseline	0.89	400	13.5		9.79		
Run 1-2 C&D Out of the Box Baseline Cycle #2						9.41	0.079
Run 1-2 C&D Out of the Box Baseline Cycle #3						9.38	0.083
Run 1-2 C&D Out of the Box Baseline Cycle #4						9.35	0.091
Average						**9.38**	**0.084**

The airflow was too high with the 7" diameter pulley at 0.25 IWC ESP so it was replaced with a 10" diameter pulley operating at 969 rpm with the motor sheave at 3 turns. **Table 12** provides the out-of-box AHRI tests for RTU3 in the vertical and horizontal test configurations at 95F OAT, and 80F DB and 67F WB indoor temperatures. The total EER* for both tests were 13 to 14% less than the rated EER and the cooling capacities were 7 to 11% less than the 90,000 Btu/hr rating at 95F OAT and 80F drybulb and 67F wet bulb temperatures.

Source Heat Pump Equipment Standard 210/240. Air-Conditioning Heating and Refrigeration Institute.

Table 12: Out-of-Box AHRI Tests for 7.5-ton non-TXV RTU3 with 10" Diameter Blower Pulley 969 rpm (3 turns) and No Economizer in Vertical and Horizontal Configuration

Test	C1/C2 Charge %	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER	Total EER*	ΔEER*	Sensible EER*s
Run 2-21A no optimized charge (vertical)	100/100	0.25	395	1100	8297	79,761	11.0	9.61	-13%	6.77
Run 3-21A AHRI Out of the Box (horizontal)	100/100	0.25	395	1190	8521	83,658	11.0	9.47	-14%	6.85

4.1.2 AHRI Verification Tests for 7.5-ton non-TXV RTU3

AHRI verification tests were performed with the cabinet panel joints sealed with tape to reduce outdoor air leakage and minor modifications to refrigerant charge and fan speed to achieve +/- 5% of the AHRI efficiency rating. The fan speed was reduced from 969 to 831 RPM by turning the motor sheave from 3 turns to 6 turns. Cooling capacity was increased by adding 6.4 ounces of refrigerant to each circuit to achieve 105% of the factory charge. With these modifications the tested efficiency was 10.47 EER and total cooling capacity was 86,269 Btu/hr as shown in **Table 13**. The EER was within 4.8% of the published rating and the cooling capacity was within 4.1% of the published 90,000 Btu/hr rating per ANSI/AHRI tolerances.[69]

Table 13: AHRI Verification Test for 7.5-ton non-TXV RTU3 with 10" Diameter Blower Pulley 831 rpm (6 turns) and No Economizer

Test	C1/C2 Charge %	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER	Tested EER	ΔEER	Tested Sensible EERs
3-21A AHRI Verification H (horizontal)	105/105	0.25	339	850	8239	86,269	11.0	10.47	4.8%	7.24

RTU3 has an Integrated Part Load Value (IPLV) AHRI rating of 11.6 and does not have an IEER rating. **Table 14** provides the measured IPLV which was 11.03 and within 4.1% of the published 11.6 IPLV rating. The measured IEER was 11.22. IEER ratings are not available from the manufacturer for this vintage R22 model.

[69] Per ANSI/AHRI STANDARD 340/360-2007, 6.3 Tolerances, "To comply with this standard, measured test results shall not be less than 95% of Published Ratings for capacities, EER values, and COP values and not less than 90% of Published Ratings for IEER or IPLV values."

Table 14: AHRI IEER and IPLV Verification Tests for 7.5-ton non-TXV RTU3 with Lowest Fan Speed and No Economizer

Test	Fan HP	Fan Turn	Fan RPM	Tested IEER	Test	Tested IPLV
IEER Calculation 7.5 ton	1.5	6	831	11.22	IPLV Calculation 7.5 ton	11.03

4.1.3 Manufacturer Refrigerant Charge Diagnostics for 7.5-ton non-TXV RTU3

The circuit-specific manufacturer refrigerant charge diagnostic protocols for the 7.5-ton non-TXV RTU3 are based on suction temperature (ST) as a function of outdoor drybulb (DB) temperature (i.e., condenser entering air) and suction pressure (SP).[70] The manufacturer refrigerant charge ST tolerances are +/-5F.[71] The manufacturer service instructions do not mention closing and sealing economizer dampers to reduce excess outdoor airflow prior to evaluating refrigerant charge diagnostics. The manufacturer does not provide superheat target values, airflow diagnostic protocols, or liquid pressure ports so subcooling cannot be evaluated. The CEC superheat (ΔSH) protocols are used to diagnose refrigerant charge and CEC temperature split (ΔTS) protocols are used to diagnose airflow and sensible cooling capacity faults based on test results for RTU3.[72] For the manufacturer and CEC RC protocols, red indicates both circuits fail, yellow indicates one circuit fails, and green indicates both circuits pass manufacturer protocols and are correctly diagnosed. Also shown are results for the CEC temperature split (ΔTS) protocol with a tolerance of +/-3F. For the CEC ΔTS protocol, red indicates ΔTS is greater than 3F (low airflow), yellow indicates ΔTS is less than -3F (low sensible capacity), and green indicates ΔTS is between -3F and 3F. For information about the CEC protocols see **Section 2.1.3**. Sensible cooling capacity is highlighted in green if it is at least 105% of the ACCA Manual N minus ventilation loads. Yellow highlighting indicates sensible cooling capacity is between 100 and 105% of ACCA Manual N and no less than 10% of non-faulted capacity. Red highlighting indicates sensible cooling capacity is less than 100% of ACCA Manual N or less than 10% of non-faulted capacity. The ACCA Manual N sensible cooling load design values minus ventilation loads are provided in **Table 2** and described in **Section 2.2.1**.

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[70] Carrier 2005. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Gas Heating/Electric Cooling Units. Installation, Start-up, and Service Instructions. Form 48HJ-32SI. Fig. 57 – Cooling Charging Charts. http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48hj-32si.pdf.

[71] Manufacturer refrigerant charge protocol for undercharge: ΔST > 5F. Manufacturer refrigerant charge protocol for overcharge: ΔST <-5F. Manufacturer protocol for correct charge: -5F ≤ ΔST ≤ 5F.

[72] California Energy Commission (CEC). 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. RA3.2 Procedures for Determining Refrigerant Charge for Split System Space Cooling Systems Without a Charge Indicator Display. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

Table 15 shows manufacturer and CEC refrigerant charge and airflow (RCA) diagnostic test results for RTU3. For the two 100% factory charge tests, the manufacturer ΔST and CEC ΔSH protocols diagnosed "false alarm" undercharge highlighted in red. For 105% factory charge test the manufacturer and CEC protocols misdiagnosed correct charge for C1 and undercharge for C2 (highlighted in yellow) even though EER and cooling capacity are within 5% of AHRI ratings (3-21A AHRI Verification H). The CEC ΔTS protocol correctly diagnosed proper airflow and sensible cooling capacity for all tests which are at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 15: Manufacturer and CEC RCA Diagnostics for 7.5-ton non-TXV RTU3 with No Economizer at 95F OAT

Test	C1/C2 Charge %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Fan Power W	Total Power W	Total Cooling Capacity Btuh	Total EER	Sensible Cooling Capacity Btuh	Sensible EERs
Run 2-21A no optimized charge (vertical)	100/100	29/41	18/24	-1.2	395	1100	8297	79,761	9.61	56,211	6.77
Run 3-21A AHRI Out of the Box (horizontal)	100/100	10/44	8/26	-0.7	395	1190	8521	83,658	9.47	58,347	7.03
3-21A AHRI Verification H (horizontal)	105/105	-1/41	-2/23	2.1	339	850	8239	86,269	10.47	59,613	7.24

Table 16 shows manufacturer and CEC RCA diagnostics for four tests performed on RTU3 with economizer installed, damper closed, and similar airflow and ESP at 95F OAT and 75F DB and 62F WB return air temperature. The measured outdoor airflow was 20.2% with dampers closed (see **Table 17**). Test 3-295 had 100% factory charge in both circuits. The other tests had unequal percentage charge amounts in each circuit. Test 7-3A was undercharged by about 20%, test 5-295-recalc was overcharged by about 20% and test 7-5A was overcharged by about 60%. With factory charge (3-295), the total efficiency was 44% less than the published 11 EER rating due to typical field conditions, static pressure and economizer installed with closed damper. Undercharging reduced EER*s by 36%. Overcharging only reduced EER*s by 0 to 2%. The manufacturer and CEC refrigerant charge protocols correctly diagnosed the 20% undercharge test highlighted in red. Both protocols correctly diagnosed circuit 1 at 100% factory charge highlighted in yellow. With 60% overcharge, both protocols misdiagnosed both circuits as correctly charged. At 100% and 120% of factory charge both protocols misdiagnosed circuit 2 as undercharged. The CEC temperature split protocol provided 100% accuracy by correctly diagnosing low capacity for the undercharge test and correct airflow and capacity for the other tests. For test 7-3A the ΔTS is highlighted in yellow indicating low capacity due to low refrigerant charge. The sensible capacity for test 7-3A was 29,304 Btuh or 40% less than the 45,024 Btuh ACCA Manual N sensible cooling load (red highlight indicates low capacity). All other sensible capacities highlighted in green are at least 105% of the ACCA manual N loads.

Table 16: Manufacturer and CEC RCA Diagnostics for 7.5-ton non-TXV RTU3 with Economizer #4 Closed Damper at 95F OAT

Test	C1/C2 Refrig Chg %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	ESP IWC	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
3-295 (horiz.)	100/100	0/51	-4/32	-3.0	335	1.20	8,779	54,370	6.2	47,546	5.4
7-3A (vertical)	73/88	57/62	41/56	-8.9	337	1.13	8,021	26,390	3.3	29,304	3.7
5-295-recalc (vertical)	113/128	3/33	-1/26	-2.5	337	1.14	8,910	53,313	6.0	48,393	5.4
7-5A (vertical)	153/168	-2/-5	-2/-3	-1.7	338	1.10	9,521	55,459	5.8	50,215	5.3

4.1.4 Economizer Outdoor Airflow Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate outdoor airflow, overventilation, and unintended outdoor air leakage on the 7.5-ton non-TXV RTU3 with economizer #4 installed.[73] These tests measured the outdoor air fraction (OAF) defined as the percentage of outdoor airflow divided by the total evaporator airflow.[74] Tests were performed on RTU3 with the evaporator fan blower motor on, compressors on, outdoor conditions of 95 and 115F, and indoor conditions of 75F DB and 62F WB. A second set of tests were performed with evaporator fan on and compressors off at 55F OAT with the gap between economizer #4 and the cabinet unsealed and sealed and with evaporator coil blockage from 5 to 50%. Accurate measurements of return temperature, outdoor temperature, mixed air temperature entering the evaporator or mixed air leaving the fan, air pressure (p), airflow (scfm), and fan power were used to calculate outdoor airflow as a fraction of the total airflow across the evaporator coil. The outdoor, return, and mixed air drybulb temperatures were measured using resistance temperature detector (RTD) sensors in the outdoor, return, and supply air samplers. The OAT entering the economizer was also measured using an array of 6 thermocouple sensors installed in the economizer inlet. The average return air drybulb temperature was also measured using an array of 6 thermocouple sensors installed in the return duct. The volumetric flow rate of air was measured using the Code Tester.[75] For tests with blower fan and compressors operating, the mixed air temperature entering the evaporator was

[73] Overventilation is caused by minimum damper positions set too far open compared to ASHRAE 62.1 minimum ventilation requirements. Unintended outdoor airflow is caused by unsealed gaps around the economizer perimeter. When the blower fan is operating, closed dampers can leak 12 to 24% and open dampers can leak 32 to 75% of total system airflow. When the blower fan is off, economizer dampers have a spring closure system that closes the dampers to reduce exfiltration and infiltration caused by wind and air buoyancy pressure referred to as the stack effect. Some economizer dampers are stuck open with a "Molex" plug, screw or mechanical failure. Economizer damper leakage increases exfiltration and increases outdoor air infiltration when the blower fan is off which increases cooling or heating loads and shortens off time. Damper leakage also reduces cooling and heating system capacity and efficiency which increases on time. Overventilation and unintended ventilation have an energy penalty for both cooling and heating (ASHRAE 2005. Fundamentals, Chapter 27).

[74] ASHRAE 62.1 defines OAF as the fraction of outdoor air intake flow in the system primary airflow.

[75] The “code tester” is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92). Standard Methods for Laboratory Airflow Measurement.

measured with an array of 16 (or 22) shielded-drybulb temperature sensors located on the air filter inlet adjacent to the evaporator.

Technicians attempting to set damper position with analog economizer controllers typically use their fingers where 1-finger is assumed to be open 10%, 2-fingers 20% and 3-fingers 30%. Using fingers to set damper positions causes variations in the opening depending on finger size and placement with respect to the damper and frame. For consistency between tests in the laboratory wooden dowels were used to set damper positions as shown in **Figure 3**. Finger diameters are as follows: 1-finger = 0.7 inch (1.8 cm), 2-fingers = 1.3 inches (3.3 cm), and 3-fingers = 2 inches (5.1 cm). These dimensions were used to establish voltages for each position using digital economizer controllers. Digital economizer controllers allow economizers to be opened based on voltage signals where 2 Volts is closed, 10 Volts is 100% open and each 0.8V increment above 2V opens the dampers by 10%. Digital economizer controllers have a user interface for technicians to set minimum positions more accurately than analog controllers.

Figure 3: Economizer #1 Damper Positions Using 1, 2, and 3-Finger Dowels

The outdoor air fraction (OAF) is calculated using **Equation 1** for tests where the average mixed-air temperature is measured directly with blower fan and compressor(s) operating.

Equation 1 $$OAF_m = \frac{T_r - T_m}{T_r - T_o}$$

Where,

OAF_m = outdoor air fraction based on measured mixed-air drybulb temperature entering through economizer as a fraction of total airflow (dimensionless or percentage)

T_r = temperature of return air (F)

T_m = temperature of mixed air entering evaporator (F)

T_o = temperature of outdoor air through economizer, relief dampers or cabinet (F)

In order to use **Equation 1** to calculate OAF with compressors operating, the drybulb temperature difference between outdoor air and return air must be at least 20F. The OAF will be more accurate if the outdoor and return temperature difference is greater than 20F.

Tests were also performed to measure outdoor air leakage with the evaporator fan blower motor on, compressors off, 55F DB and 51F WB outdoor temperature conditions, and 75F DB and 62F WB indoor return temperature conditions. This method is more accurate since it uses enthalpy to calculate the outdoor air fraction including outdoor air humidity and sensible and latent heat. To distinguish between compressible and incompressible flow in ideal gases, the Mach number (ratio of speed of flow to speed of sound) must be greater than 0.3 before significant compressibility occurs. The Mach number of the tested HVAC equipment is less than 0.03 (10 times lower). Therefore, the standard Bernoulli energy equation is used to calculate the mechanical heat loss of the fan as shown in **Equation 2**.

Equation 2 $$\dot{W}_{fan} + \dot{m}\frac{v_1^2}{2} + gz_1 + \dot{m}\frac{p_1}{\rho} = \dot{m}\frac{v_2^2}{2} + gz_2 + \dot{m}\frac{p_2}{\rho} + \dot{E}_{loss\ fan}$$

Where,

Subscript 1	refers to entering return conditions and subscript 2 refers to leaving supply conditions
$\dot{W}_{fan}$	= measured electric power used by fan (W)
$\dot{m}$	= mass flow of air (kg/s)
ρ	= density of air (kg/m^3)
V	= velocity of air (m/s)
g	= acceleration of gravity 9.81 (m/s^2)
z	= elevation above reference plane (m)
p	= pressure of air (1) entering or (2) leaving the fan (Pa)
$\dot{E}_{loss\ fan}$	= mechanical heat loss of fan excluding mechanical work causing air movement (W)

The air velocity and elevation above the reference plane are the same at the inlet and outlet positions. Therefore, the Bernoulli equation can be simplified to calculate the heat loss of the fan as shown in **Equation 3**.

Equation 3 $$\dot{E}_{loss\ fan} = \dot{W}_{fan} - \dot{m}\frac{(p_2 - p_1)}{\rho}$$

The fan power is measured and the fan mechanical heat loss is calculated for each test performed at 55F outdoor conditions with compressors off. Heat is added to the air as it passes across the fan motor which increases the mixed air temperature leaving the fan.

The air temperature increase due to fan heat is calculated per **Equation 4**.

Equation 4 $$\Delta T_{fan} = \frac{\dot{E}_{loss\ fan}}{c_p \rho \dot{V}}$$

Where,

ΔT_{fan} = temperature increase of air due to fan heat (°C)

c_p = specific heat of air at constant pressure (J/kg-C)

$\dot{V}$ = volumetric flow rate of air (m^3/s)

The temperature in degrees Fahrenheit from Celsius is calculated in **Equation 5**.

Equation 5 $$\Delta T_{fan} = T(C)\frac{9}{5} + 32 = T(F)$$

The fan adds heat but not moisture to the airstream. Therefore the average mixed air humidity ratio (leaving the fan) is equal to the supply air humidity ratio measured for each test using RTD sensors in the supply air sampler. Additionally, the temperature of the mixed air before the fan equals that measured for the supply air minus temperature increase due to fan heat as provided by **Equation 4**. The enthalpy of the mixed air is determined from the mixed air temperature and the supply air humidity ratio. The average return air temperature, mixed air temperature leaving fan, and OAT entering economizer, relief dampers, or cabinet are measured for each test. The measurements are used to calculate the outdoor air fraction (OAF) entering the economizer, relief damper, or cabinet using **Equation 6**.

Equation 6 $$OAF_e = \frac{h_r - h_m}{h_r - h_o}$$

Where,

OAF_e = outdoor air fraction of air (based on enthalpy) entering unit through economizer, relief damper, or cabinet as a fraction of total airflow (dimensionless or percentage)

h_r = enthalpy of return air from conditioned space (Btu/lbm or J/kg)

h_m = enthalpy of mixed air leaving fan based on supply air humidity ratio and temperature minus temperature increase due to fan, ΔTfan (Btu/lbm or J/kg)

h_o = enthalpy of outdoor air through economizer, dampers or cabinet (Btu/lbm or J/kg)

Enthalpy for each condition is calculated using **Equation 7** from ASHRAE 2009.[76]

Equation 7

$$h = 0.24\,t + \left[\frac{\dfrac{(1093 - 0.556\,w)\,0.62198\,p_{ws}}{p_a - p_{ws}} - 0.24\,(t - w)}{1093 + 0.444\,t - w} \right] \left[1061 + 0.444\,t\right]$$

Where,

h = specific enthalpy of moist air (Btu/lbm)

t = dry bulb temperature (F)

w = wet bulb temperature (F)

p_a = atmospheric pressure for tests (psia)

p_{ws} = saturation pressure for wet bulb temperature defined in **Equation 8** (psia)[77]

Equation 8 $p_{ws} = EXP(C1 + C2 + C3)$

Where,

$C1 = -10440.397(w + 459.67)^{-1} - 23.71601592 - 0.027022355(w + 459.67)$

$C2 = 0.00001289036(w + 459.67)^2 - 2.4780681 \times 10^{-9}(w + 459.67)^3$

$C3 = 6.5459673 \ln(w + 459.67)$

The outdoor air fraction calculations are checked using **Equation 9**.

Equation 9 $$OAF_t = \frac{\left(T_r - T_m + \Delta T_{fan}\right)}{\left(T_r - T_o\right)}$$

Where,

OAF_t = outdoor air fraction (based on drybulb temperature measurements) entering unit through economizer, relief damper, or cabinet as a fraction of total airflow (dimensionless)

T_r = temperature of return air (F)

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[76] ASHRAE 2009. ASHRAE Handbook-Fundamentals. American Society of Heating Refrigeration and Air Conditioning Engineers, Inc.

[77] Hyland, R.W. and A. Wexler. 1983b. Formulations for the Thermodynamic Properties of the Saturated Phases of Water from 173.15 K to 473.15 K. ASHRAE Transactions 89(2A):500-519.

T_m = temperature of mixed air leaving fan (F)

T_o = temperature of outdoor air entering economizer, relief dampers or cabinet (F)

Fan heat is included in **Equation 6** and **Equation 9** since the mixed air temperature leaving the fan is measured downstream. Fan heat is not included in **Equation 1** since the mixed air temperature entering the evaporator is measured upstream of the fan.

Table 17 provides the economizer #4 outdoor air fractions calculated using **Equation 1** at 95F and 115F OAT and using **Equation 6** at 55F OAT for the 7.5-ton non-TXV RTU3 with unsealed perimeter. Economizer #4 was obtained from a manufacturer who makes OEM economizers for RTU3. Technicians typically establish minimum outdoor damper opening using 1, 2, or 3 fingers. Tests were performed with the following finger diameters: 1-finger 0.74 inch (1.88 cm), 2-fingers 1.289 inches (3.27 cm), and 3-fingers 1.972 inches (5.01 cm).

Table 17: Economizer #4 Outdoor Air Fractions Calculated Using Eq. 1 at 95F and 115F OAT and Eq. 6 at 55F OAT for 7.5-ton non-TXV RTU3 with Unsealed Perimeter

Description	Test	Evap Airflow scfm/ ton	Eq. 1 Calc OAF_m at 95F %	Test	Airflow scfm/ ton	Eq. 1 Calc OAF_m at 115F %	Test	Airflow scfm/ ton	Eq. 6 Calc OAF_e at 55F %
No Economizer	Run-3-20N95	348	6	3-20N115	345	7.5			
Closed (2V)	3-295	335	16.7	3-2115	334	17.3	1-CEH (comp off)	328	20.2
1F Open (0.74")	3-2951	335	26	3-21151	334	24.1			
2F Open (1.289")	3-2952	336	32.2	3-21152	334	30.5			
3F Open (1.972")	3-2953	338	36.5	3-21153	335	37.1			
100% Open (10V)	3-295OD	329	66.9	3-2115OD	326	60.9	1-OEH (comp off)	331	61.1

Table 18 provides the second tests of economizer #4 outdoor air fractions calculated using **Equation 6** at 55F OAT for the 7.5-ton RTU3 with unsealed and sealed perimeter. The second tests include all damper positions at 55F while the first tests only included closed and 100% open tests at 55F. The 55F tests at closed and 100% open are 3% lower than the first tests. Sealing the gap between the economizer perimeter and the cabinet reduced the OAF by 1.6 to 3.3%. The OAF difference is referred to as ΔOAF % which equals unsealed minus sealed OAF %.

Table 18: Second Tests of Economizer #4 Outdoor Air Fractions Calculated Using Eq. 6 at 55F OAT for 7.5-ton non-TXV RTU3 with Unsealed and Sealed Perimeter

Description	Unsealed Test	Airflow scfm/ ton	Eq. 6 Calc OAF_e at 55F %	Sealed Economizer Perimeter Test	Airflow scfm/ ton	Eq. 6 Calc OAF_e at 55F %	ΔOAF %
No Economizer	C8-55-NE-CAB	316	2.3				
Closed Damper	C8-55-CE3	324	16.0	C8-55-TCE3	322	13.4	2.6
1F Open (5.1V)	C8-55-1FE3	328	24.4	C8-55-T1FE3	326	21.1	3.3
2F Open (6V)	C8-55-2FE3	331	26.9	C8-55-T2FE3	329	25.4	1.5
3F Open (6.9V)	C8-55-3FE3	336	32.9	C8-55-T3FE3	333	31.3	1.6
100% Open (10V)	C8-55-OE3	330	55.4	C8-55-TOE3	328	58.1	

Table 19 provides second tests of economizer #4 outdoor air fractions calculated using **Equation 6** at 55F OAT for the 7.5-ton non-TXV RTU3 with 5 to 50% evaporator coil blockage and unsealed perimeter. Evaporator coil blockage reduced evaporator airflow by 1 to 12% and increased OAF by 4 to 7%. Dirty air filters and evaporator coils reduce airflow and increase inlet static pressure. This causes OAF to increase and cooling and heating efficiency to decrease underscoring the importance of enhanced HVAC maintenance services to clean coils and install clean air filters every 3 months depending on ambient air quality conditions.

Table 19: Second Tests of Economizer #4 Outdoor Air Fractions Calculated Using Equation 6 at 55F OAT for 7.5-ton non-TXV RTU3 with 0 to 50% Evaporator Blockage

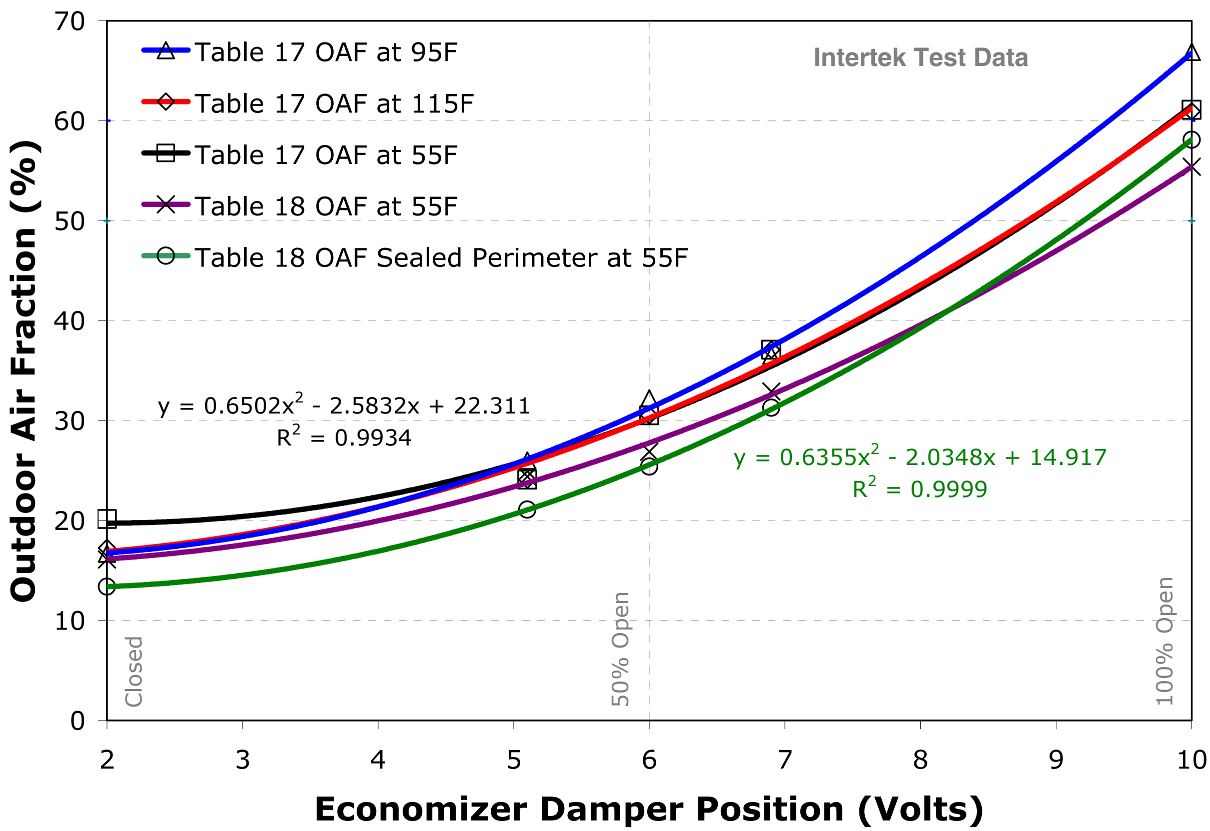
Description	Unsealed Test	Equation 6 Calc OAF_e at 55F %	OAF_e Impact %	Airflow scfm/ton	Airflow Impact %
Closed 0% Evaporator Blockage	C8-55-CE3	16.0	0%	324.1	0%
Closed 5% Evaporator Blockage	C8-EB5-55-CE-2	16.1	0%	320.9	-1%
Closed 10% Evaporator Blockage	C8-EB10-55-CE	16.6	4%	316.0	-2%
Closed 20% Evaporator Blockage	C8-EB20-55-CE	16.6	4%	313.8	-4%
Closed 35% Evaporator Blockage	C8-EB35-55-CE	16.7	4%	299.6	-8%
Closed 50% Evaporator Blockage	C8-EB50-55-CE	17.1	7%	290.6	-12%

Figure 4 shows the correlation between outdoor air fraction and damper position for RTU3 with 333 scfm/ton total airflow with economizer #4 perimeter unsealed at 55F, 95F, and 115F OAT and with the economizer perimeter sealed at 55F OAT.[78] The regression equations are based on **Table 17** using calculated unsealed perimeter OAF at 55F and **Table 18** using calculated sealed perimeter OAF at 55F. At the closed position, the average efficiency difference between unsealed and sealed economizer perimeter is 4.1 +/- 1%. Sealing the economizer perimeter under the hood reduces unintended outdoor airflow and improves cooling and heating efficiency. Fully

[78] Economizer damper position is proportional to electric potential in volts (V) from the controller to the actuator, and 2V is closed, 6V is 50% open, and 10V is 100% open.

open is 10 volts and fully closed is 2 volts. For a gear-driven damper this is roughly equivalent to percentage open. Establishing the most efficient minimum damper position is important for health, comfort, and energy efficiency. Outdoor airflow provided by each economizer will vary and manufacturers typically do not provide outdoor airflow as a function of damper position so technicians currently have no reliable method to establish optimal damper position.

Figure 4: Outdoor Air Fraction versus Economizer Damper Position for 7.5-ton non-TXV RTU3 with 333 scfm/ton Airflow Unsealed and Sealed Perimeter at 55, 95 and 115F OAT



4.1.5 Economizer 95F Efficiency Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the application efficiency impact of economizer outdoor air ventilation for the 7.5-ton non-TXV RTU3 with economizer #4 installed with outdoor air dampers closed, partially open, and 100% open and the economizer perimeter unsealed and sealed with tape. Tests were performed with factory charge and outdoor conditions of 95, 82, and 115F and indoor conditions of 75F DB and 62F WB. **Table 20** provides economizer #4 outdoor air ventilation impacts and FDD versus damper position with unsealed perimeter at 95F OAT. With no economizer installed the total EER* was 8.1 and the sensible EER*s was 6.4. With economizer #4 installed and dampers closed, the total EER* was 6.2 and sensible EER*s was 5.4. The outdoor airflow ventilation load can have a significant impact on cooling and heating efficiency especially when the minimum damper position is more open than necessary to meet the ASHRAE 62.1 minimum ventilation requirement. The reduction in efficiency with economizer #4 installed and closed dampers was 23.5% for total EER* and 15.4% for sensible EER*s. With closed dampers the economizer #4 efficiency was 44% less than the AHRI EER rating of 11.0 and 25% less than the sensible EER at the AHRI test conditions. Opening economizer dampers per the outdoor air leakage tests performed at 55F in the previous section significantly reduced efficiency. At 95F outdoor temperature with dampers 100% open, the total EER* was 0.8 and sensible EER*s was 0.8. These application efficiencies are 93 to 58% less than the AHRI rating and 90 to 53% less than the efficiency with no economizer installed. ASHRAE 62.1 typically requires 15% minimum outdoor air ventilation for most buildings.[79] This economizer would provide 16.7% OAF with closed dampers. If a technician set the minimum damper position at 2-fingers open, the economizer would provide 32.2% OAF or 93% more outdoor ventilation than with dampers closed. The overventilation at 2-fingers open would reduce EER*s to 4.7 EER*s and this is 13.2% less efficient than 5.4 EER*s with dampers closed. Providing adequate outdoor ventilation air is as important as providing comfortable indoor temperature control. The reduction in efficiency due to overventilation or outside air leakage beyond minimum requirements represents an important energy efficiency opportunity for space cooling and heating. The manufacturer ΔST and CEC ΔSH refrigerant charge protocols for one or both circuits diagnosed "false alarm" undercharge for all tests. The CEC ΔTS protocol diagnosed acceptable airflow and capacity for no economizer and closed dampers, and low capacity for dampers open. The no economizer and closed damper sensible capacities are greater than 105% of the 45,024 Btuh ACCA Manual N sensible cooling load and highlighted in green. All other tests are less than 100% of ACCA Manual N and highlighted in red.

Table 20: Economizer #4 Outdoor Air Ventilation Impacts and FDD versus Damper Position Unsealed Perimeter for 7.5-ton non-TXV RTU3 at 95F OAT

Description	Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	OAF %	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
No Economizer	Run-3-20N95	1/43	-2/29	-0.2	335	6.0	71,052	8.1	56,838	6.4
Closed Damper	3-295	0/51	-3/34	-3.0	335	16.7	54,370	6.2	47,546	5.4
1F Open (5.1V)	3-2951	-2/51	-5/31	-3.7	335	26.0	46,529	5.3	43,831	4.9
2F Open (6V)	3-2952	-2/51	-5/30	-4.5	336	32.2	40,832	4.6	40,832	4.7
3F Open (6.9V)	3-2953	7/52	5/29	-5.1	338	36.5	36,941	4.2	39,808	4.5
100% Open (10V)	3-295OD	13/53	12/25	-9.4	329	66.9	6,789	0.8	26,615	3.0

Table 21 provides economizer #4 outdoor air ventilation impacts and FDD versus damper position with unsealed perimeter at 82F OAT. With dampers closed the total EER* was 8.5 and

[79] ASHRAE 62 specifies outdoor airflow per person and per 1000 ft^2 of conditioned floor area and typical unit ft^2/ton and cfm/ton indicates average outdoor airflow of 15 +/- 10% for most building occupancies.

sensible EER*s was 6.7. With dampers 100% open, the total EER* was 5.6 or 34% less than the efficiency with dampers closed. Application sensible efficiency was 5.7 EER*s or 15% less than dampers closed. The manufacturer ΔST and CEC ΔSH protocols indicate both circuits have "false alarm" undercharge for all tests. With 82F outdoor air, the CEC ΔTS protocol passes for all tests except 100% open dampers and sensible capacity for this test is highlighted in red indicating low capacity. This finding indicates the difficulty of detecting excess outdoor airflow when the difference between outdoor and return air temperatures is less than 20F. At 82F, the 100% open damper test sensible cooling was 82% of the closed damper capacity and is highlighted in red. All other tests at 82F have sensible cooling capacities at least 105% of the 35,978 Btuh ACCA Manual N sensible cooling load and no less than 10% of non-faulted closed damper test highlighted in green.

Table 21: Economizer #4 Outdoor Air Ventilation Impacts and FDD versus Damper Position Unsealed Perimeter for 7.5-ton non-TXV RTU3 at 82F OAT

Description	Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	OAF %	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Closed Damper	3-282	14/55	8/29	-0.8	336	16.7	65,471	8.5	52,216	6.7
1F Open (5.1V)	3-2821	15/55	9/29	-1.4	338	26.0	62,293	8.0	50,907	6.6
2F Open (6V)	3-2822	18/58	11/29	-1.8	338	32.2	59,904	7.8	49,729	6.5
3F Open (6.9V)	3-2823	18/58	12/29	-2.3	340	36.5	57,398	7.4	48,533	6.3
100% Open (10V)	3-282OD	20/58	12/26	-3.8	332	66.9	43,868	5.6	42,929	5.7

Table 22 provides economizer #4 outdoor air ventilation impacts and FDD versus damper position with unsealed perimeter for RTU3 at 115F OAT. With dampers closed the total EER* was 4.0 and sensible EER*s was 3.9. With dampers 100% open, the total EER* was negative, and sensible EER*s was 0.9 or 77% less than the efficiency with dampers closed. If a technician set the minimum damper position at 2-fingers open, overventilation or unintended outdoor air leakage would reduce EER*s to 3.1 EER*s or 21% less efficient than with dampers closed. The manufacturer ΔST and CEC ΔSH protocols diagnosed one or both circuits have a "false alarm" undercharge for all tests. The CEC ΔTS protocol correctly diagnosed low cooling capacity for all tests and all sensible capacities are highlighted in red. The manufacturer ST protocol at 115F OAT and 100% open dampers (test 3-2115OD) diagnosed C1 was within tolerances due to excessive outdoor air evaporating enough refrigerant to increase the suction temperature to within the manufacturer protocol tolerance. These tests demonstrate the importance of reducing or eliminating excessive outdoor air by closing and sealing dampers when checking refrigerant charge diagnostics. Excessive outdoor air can have a significant impact on heating and cooling capacity and efficiency depending on how far the damper is stuck open. All tests at 115F have sensible cooling capacities much less than the 61,132 Btuh ACCA Manual N sensible cooling load at 115F highlighted in red.

Table 22: Economizer #4 Outdoor Air Ventilation Impacts and FDD versus Damper Position Unsealed Perimeter for 7.5-ton non-TXV RTU3 at 115F OAT

Description	Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	OAF %	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Closed Damper	3-2115	-3/39	0/35	-4.5	334	17.3	42,077	4.0	41,268	3.9
1F Open (5.1V)	3-21151	-8/40	-1/36	-6.3	334	24.1	31,660	3.0	31,660	3.4
2F Open (6V)	3-21152	-10/41	-3/35	-7.4	334	30.5	25,733	2.4	25,733	3.1
3F Open (6.9V)	3-21153	-13/43	-4/35	-8.9	335	37.1	16,752	1.6	16,752	2.7
100% Open (10V)	3-2115OD	2/45	11/25	-15.4	326	60.9	-16.350	-1.5	9,888	0.9

Figure 5 shows the decline in energy efficiency performance with increasing outdoor airflow as dampers are opened from closed to fully open for RTU3. At approximately 32% outdoor airflow (2-fingers) total EER* equals EER*s. As dampers open beyond 32%, the increased contribution of outdoor humidity causes negative latent cooling indicating more moisture is added from outdoor airflow than the cooling coil can remove.

Figure 5: Application Energy Efficiency versus Outdoor Airflow and Economizer #4 Damper Position for 7.5-ton non-TXV RTU3 at 95F OAT

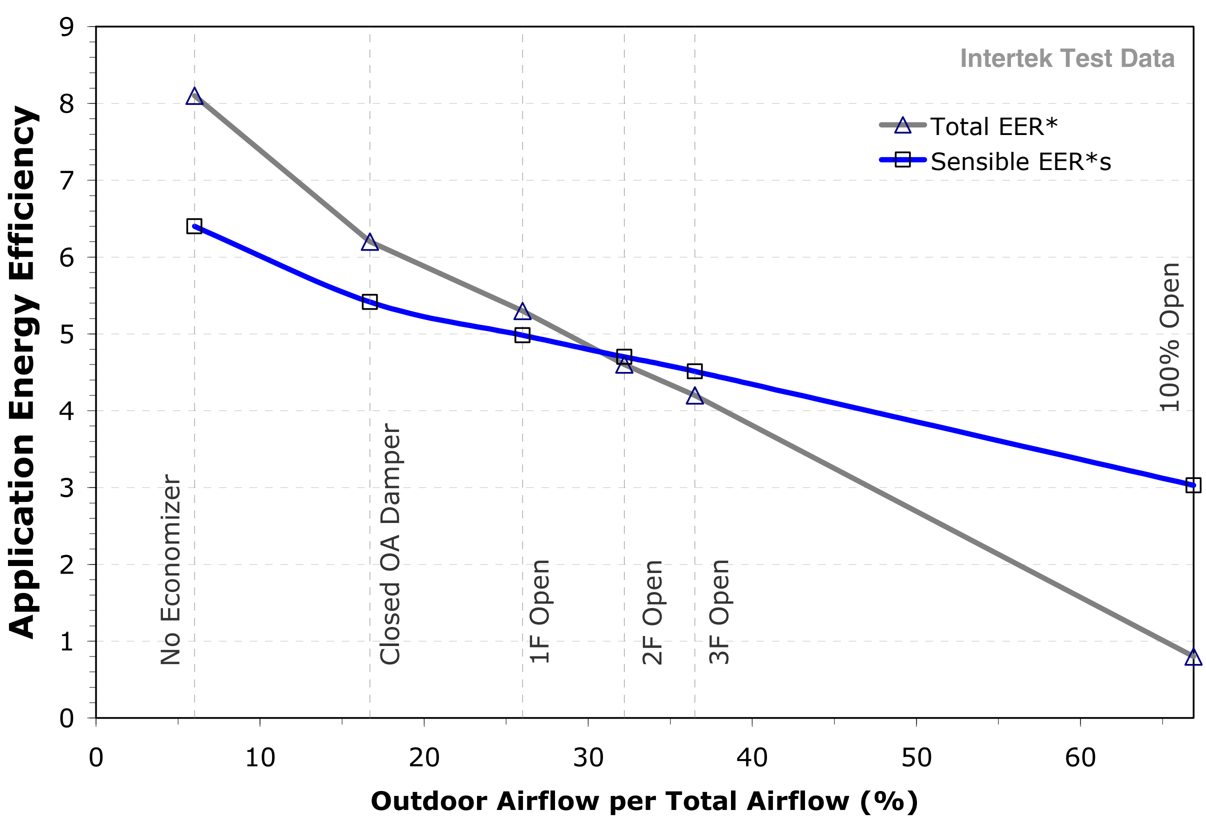


Table 23 provides second tests of economizer #4 outdoor air ventilation impacts versus damper position with unsealed perimeter for RTU3 at 95F OAT. With no economizer the total EER* was 7.6 and sensible EER*s was 6.0. With dampers closed total EER* was 5.7 and sensible EER*s was 5.1. With dampers 100% open, total EER* was 0.4 and sensible EER*s was 2.4 which was 53% less than EER*s with dampers closed. If a technician set the minimum damper position at 2-fingers open, overventilation or unintended outdoor air leakage would reduce EER*s to 4.4 EER*s. Setting dampers at 1-finger open would provide 26.4% OAF instead of 16% OAF with dampers closed and reduce EER*s by 14%. The manufacturer ΔST and CEC ΔSH protocols diagnosed both circuits with a "false alarm" undercharge for all tests. The CEC ΔTS protocol correctly diagnosed low cooling capacity for all tests except no economizer. Sensible capacities less than the 45,024 Btuh ACCA Manual N sensible cooling load at 95F are highlighted in red.

Table 23: Second Tests Economizer #4 Outdoor Air Ventilation Impacts versus Damper Position Unsealed Perimeter for 7.5-ton non-TXV RTU3 at 95F OAT

Description	Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	OAF %	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
No Economizer	C8-95-NE-CAB	49/36	36/27	-0.1	324	2.3	67,156	7.6	53,130	6.0
Closed Damper	C8-95-CE	52/39	35/29	-3.4	329	16.1	50,029	5.7	44,846	5.1
1F Open (5.1V)	C8-95-1FE	53/39	34/27	-4.6	330	24.4	42,258	4.8	41,093	4.7
2F Open (6V)	C8-95-2FE	53/39	34/27	-5.4	331	26.9	37,348	4.2	38,951	4.4
3F Open (6.9V)	C8-95-3FE	53/39	34/26	-6.6	334	32.9	30,388	3.4	35,887	4.1
100% Open (10V)	C8-95-OE	56/41	32/21	-11.4	322	55.4	3,639	0.4	21,029	2.4

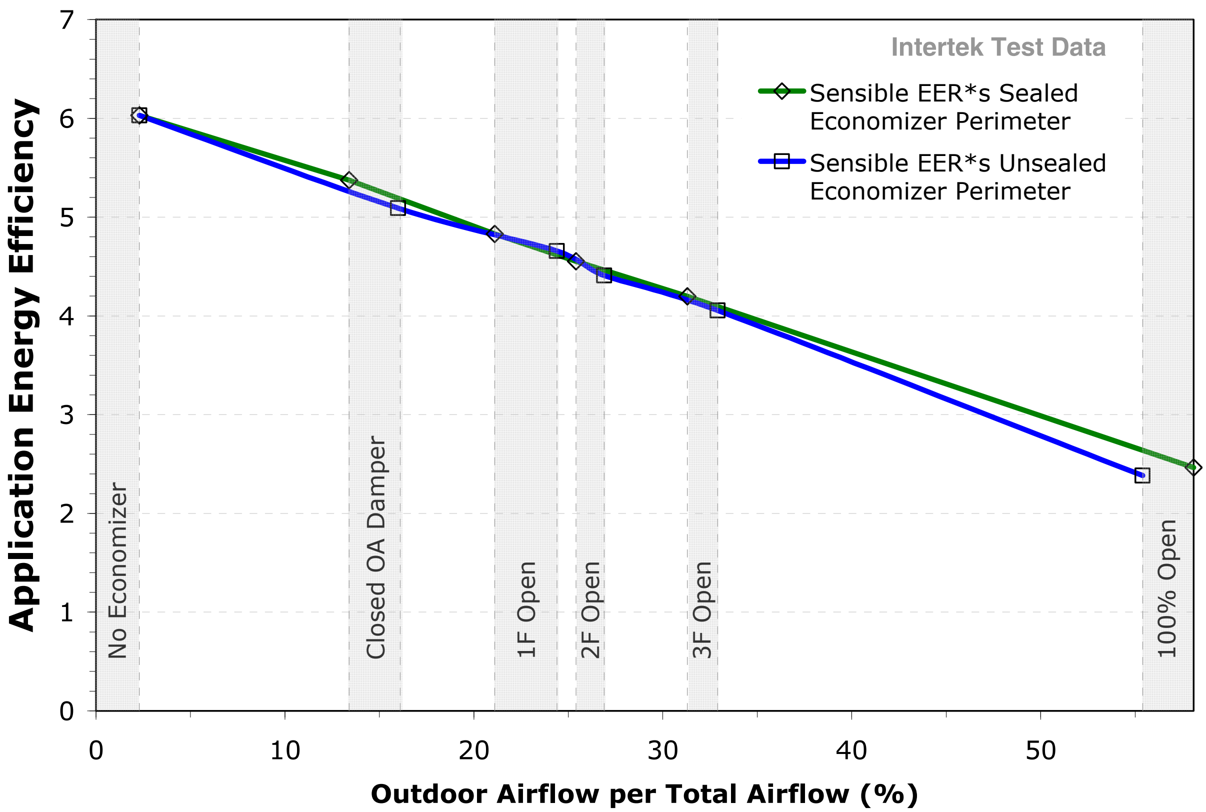
Table 24 provides second tests of economizer #4 outdoor air ventilation impacts versus damper position with sealed perimeter for RTU3 at 95F OAT. The closed damper sealed perimeter tests pass the CEC temperature split diagnostic and sensible cooling capacity. The manufacturer ΔST and CEC ΔSH protocols diagnosed both circuits with a "false alarm" undercharge for all tests.

Table 24: Second Tests Economizer #4 Outdoor Air Ventilation Impacts versus Damper Position Sealed Perimeter for 7.5-ton non-TXV RTU3 at 95F OAT

Description	Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	OAF %	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
No Economizer	C8-95-NE-CAB	49/36	36/27	-0.1	324	2.3	67,156	7.6	53,130	6.0
Closed Damper	C8-95-TCE	50/38	34/28	-2.3	327	13.4	55,289	6.3	47,425	5.4
1F Open (5.1V)	C8-95-T1FE	51/38	34/27	-4.0	330	21.1	45,712	5.2	42,721	4.8
2F Open (6V)	C8-95-T2FE	51/39	34/26	-4.9	331	25.4	40,646	4.6	40,300	4.6
3F Open (6.9V)	C8-95-T3FE	52/39	34/25	-6.1	333	31.3	33,862	3.8	37,188	4.2
100% Open (10V)	C8-95-TOE	55/40	31/21	-11.1	320	58.1	5,202	0.6	21,786	2.5

Figure 6 shows the impact on energy efficiency performance versus economizer #4 damper position with sealed and unsealed perimeter for RTU3. The efficiency decreased significantly as dampers were opened and 95F outdoor airflow increased. The sealed economizer perimeter sensible EER* values are 4 +/-0.6 % greater than unsealed tests at the same damper position. These test results demonstrate the impact of excessive outdoor air on efficiency and cooling capacity. The ASHRAE 62.1 outdoor ventilation rates for most building occupancies are 6 to 10% for offices, 22% for retail, 33% for auditoriums and schools, 40% for restaurants and health clubs, and 53% or more for cafeterias and sports arenas. Ventilation rates for unoccupied spaces can be minimized to save energy. While the decline in efficiency is a system load, unnecessary loads can be avoided if the optimal minimum damper position is established. Field observations found approximately 50% of units with economizers not working properly or dampers stuck 10 to 100% open with Molex plugs or other objects stuck between damper blades. **Figure 5** and **Figure 6** show realistic efficiency impacts caused by excessive ventilation loads (2 to 5 times greater than ASHRAE 62.1) on hot summer days when OAT is 95F or greater.

Figure 6: Application Sensible Efficiency versus Outdoor Airflow and Damper Position for 7.5-ton non-TXV RTU3 with Economizer #4 Sealed and Unsealed Perimeter at 95F OAT (Second Tests)

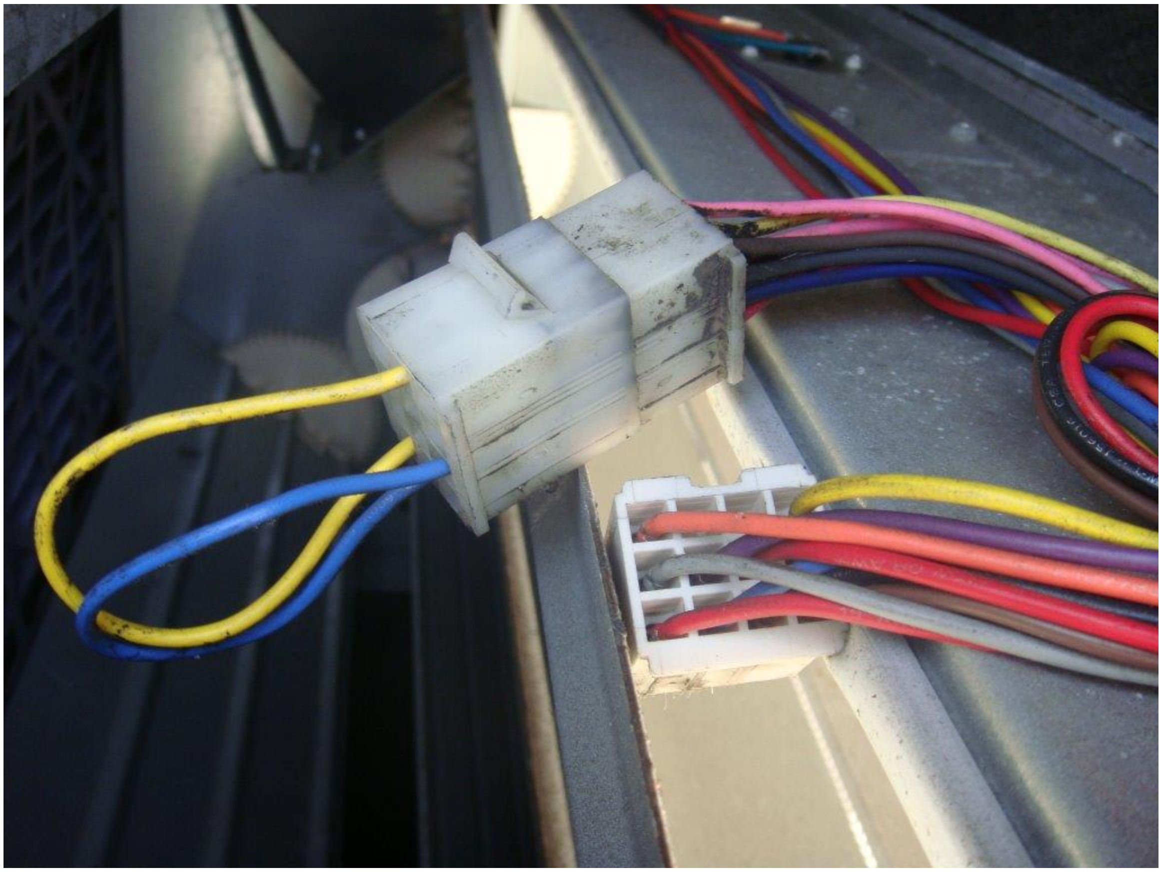


Overventilation and unintended outdoor airflow are common maintenance faults on all commercial buildings. Reducing overventilation can have a significant impact on thermal comfort, HVAC efficiency, and energy use. The manufacturer provides information for "troubleshooting" and diagnosing too much outdoor air.[80] The most common problems are inadequate cooling capacity (condenser operates continuously). These problems are caused by the following faults: 1) dirty air filter, 2) dirty/blocked condenser, 3) unit undersized for load (low cooling capacity or too much outdoor air), 4) thermostat set too low, 5) leaking compressor valves, or 6) non-condensables. For low heating capacity the faults are: 1) gas input to unit too low, 2) too much outdoor air, 3) restricted airflow, 4) blower speed too low, or 5) limit switch cycles main burners. Technicians can easily check and correct dirty air filters, dirty/blocked condenser, restricted airflow, and blower speed (fan belt tension/alignment, pulley and motor sheave). Technicians can also check gas input pressure and burner limit switch. Low cooling or heating capacity caused by too much outdoor air can be checked and corrected by adjusting the economizer minimum damper position. Field observations found approximately 50% of units with economizers not working properly or dampers stuck 10 to 100% open with Molex plugs or other objects stuck between damper blades (see **Figure 7**). These findings are consistent with previous studies of economizer performance indicating only about 25% working properly, with the remaining 75% providing poor performance due to maintenance deficiencies, improper control, or systemic problems.[81]

[80] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 34, Troubleshooting. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

[81] Energy Design Resources (EDR) 2011. Energy Design Resources Design Brief: Economizers. http://energydesignresources.com/media/2919091/edr_designbrief_economizers.pdf.

Figure 7: Economizer with Molex Plug Stuck Between Damper Blades



4.1.6 Economizer Efficiency Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the application efficiency (EER*) of RTU3 with factory charge and economizer #4 installed and outdoor air dampers 100% open to emulate economizer mode operation. **Table 25** provides laboratory test results of cooling capacity and application efficiency versus outdoor air conditions with economizer #4 dampers 100% open with economizer fan only and economizer plus 1st stage compressor (C1) cooling. Tests are also provided for closed dampers 1st stage cooling and closed dampers 2nd stage cooling to emulate unit with non-operational economizer and dampers closed. Indoor return air conditions are constant for all tests at 75F drybulb and 62F wetbulb. Outdoor conditions vary from 70F DB and 60F WB (70/60) to 65/57, 60/54 and 55/51. All sensible cooling capacities are greater than 105% of ACCA Manual N sensible cooling loads at these low outdoor temperature conditions highlighted in green. The economizer fan only tests at 70 and 65F OAT do not meet the ACCA Manual N sensible cooling loads at 65F and 70F outdoor temperatures highlighted in red.

Table 25: Functional and Non-Functional Economizer #4 at 70, 65, 60, and 55F OAT for 7.5-ton non-TXV RTU3 with Factory Charge at 333 scfm/ton

Description	Test	Outdoor DB/WB (F)	Total Cooling Capacity Btuh	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Economizer fan only	2-OEH	70/60	6,213	1,539	4.0	5,015	3.3
Economizer plus 1st Stage Compressor	2-1EH	70/60	47,449	4,586	10.3	35,264	7.7
Closed Dampers 1st Stage Compressor	2-1CH	70/60	41,183	4,596	9.0	29,896	6.5
Closed Dampers 2nd Stage Compressor	2-2CH Retest	70/60	77,433	6,989	11.1	57,495	8.2
Economizer fan only	3-OEH	65/57	18,620	1,550	12.0	12,989	8.4
Economizer plus 1st Stage Compressor	3-1EH	65/57	61,677	4,446	13.9	43,053	9.7
Closed Dampers 1st Stage Compressor	3-1CH Retest	65/57	45,236	4,454	10.2	32,385	7.3
Closed Dampers 2nd Stage Compressor	3-2CH Retest	65/57	80,323	6,651	12.1	59,145	8.9
Economizer fan only	4-OEH 2nd	60/54	31,657	1,585	20.0	20,697	13.1
Economizer plus 1st Stage Compressor	4-1EH	60/54	73,488	4,342	16.9	49,245	11.3
Closed Dampers 1st Stage Compressor	4-1CH Retest	60/54	47,494	4,325	11.0	33,940	7.8
Closed Dampers 2nd Stage Compressor	4-2CH Retest	60/54	79,328	6,341	12.5	57,990	9.1
Economizer fan only	1-OEH	55/51	43,825	1,583	27.7	28,942	18.3
Economizer plus 1st Stage Compressor	1-1EH	55/51	83,644	4,205	19.9	55,897	13.3
Closed Dampers 1st Stage Compressor	1-1CH Retest	55/51	49,182	4,199	11.7	34,837	8.3
Closed Dampers 2nd Stage Compressor	1-2CH Retest	55/51	80,023	6,052	13.2	58,384	9.6

Table 26 provides calculated sensible cooling and efficiency values for RTU3 with functional economizer dampers 100% open with fan-only and 1st-stage compressor (C1) cooling if needed to meet the load. Indoor return air conditions are constant at 75F drybulb and 62F wetbulb. Outdoor conditions vary from 70F DB and 60F WB (70/60) to 65/57, 60/54 and 55/51. For these example calculations, economizer fan only is assumed to operate for at least 2 minutes plus time required for space temperature to increase 2F above thermostat setpoint where C1 is energized.[82] Fan-only operation is sufficient to meet the cooling load at 60F and 55F OAT. At 70F OAT economizer fan-only operates 2 minutes and at 65F economizer fan-only operates 5 minutes before 1st-stage compressor (C1) is energized. Cooling loads are based on ACCA Manual N commercial load calculations assuming 3 W/ft^2 for lighting and equipment loads, 1.22 Btu/hr-ft^2 for occupant loads, 0.52 Btu/F-ft^2 for building envelope loads (vary based on indoor-outdoor temperature difference), and solar heat gains of 48 Btu/hr-ft^2 of window area.[83] Cooling loads

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[82] Manufacturer default locks out 2nd stage compressor with economizer operation. Default between Y1 and Y2 is 2 minutes plus time required for room temperature at thermostat to increase 2F above thermostat setpoint for Y1. Carrier 2001. Product Data 48HJD/HJE/HJF Single-Package Rooftop Units High-Efficiency Electric Cooling/Gas Heating. Form 48HJ-12PD. Page 59. http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48hj-12pd.pdf. Carrier 2006. Owner's Manual. 7-Day Programmable Digital Commercial Thermostat P/N 33CS450-01. http://dms.hvacpartners.com/docs/1005/public/0a/88-504.pdf.

[83] ACCA 2008. Manual N - Commercial Load Calculations for Small Commercial Buildings. Fifth Edition. ACCA http://www.acca.org/technical-manual/manual-n/

are used with laboratory test data of sensible cooling capacity and power to calculate time required to satisfy the thermostat for the functional and non-functional economizers. All delivered sensible cooling capacities satisfy the thermostat, are greater than the ACCA Manual N sensible cooling load highlighted in green.

Table 26: Functional Economizer #4 at 70, 65, 60, and 55F OAT for 7.5-ton non-TXV RTU3 with Factory Charge at 333 scfm/ton (Calculated EER*s and Sensible Capacity)

Description	Test	Outdoor DB/WB (F)	Fan-Only (min)	1st -Stage (min)	Total kWh	Sensible Cooling Btu	Sensible EER*s
Economizer plus 1-Stage Compressor	2-OEH/2-1EH	70/60	2.0	8.9	0.733	5,406	7.4
Economizer plus 1-Stage Compressor	3-OEH/3-1EH	65/57	5.0	3.9	0.419	3,888	9.3
Economizer fan only	4-OEH 2nd	60/54	9.9	0.0	0.262	3,427	13.1
Economizer fan only	1-OEH	55/51	3.2	0.0	0.085	1,554	18.3

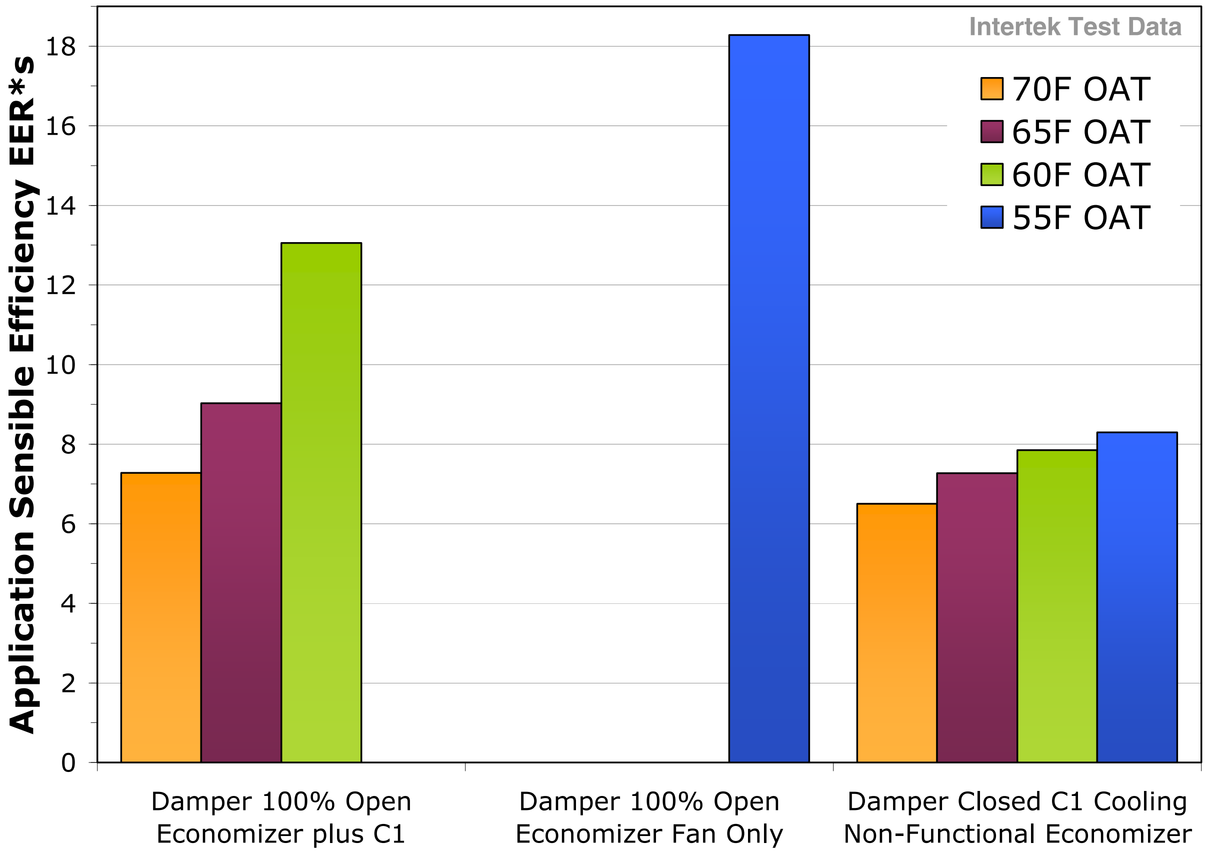
Table 27 provides calculated sensible cooling and efficiency values for the same unit and space conditions with non-functional economizer and dampers closed with 1 or 2 cycles of 1st-stage compressor operation (C1). The off cycle varies depending on how much time it takes for the space temperature to increase above the 75F setpoint based on internal loads. For comparison the total time of non-functional economizer operation is limited to not exceed the total time of functional economizer operation for each outdoor temperature. The total kWh usage for the non-functional economizer is 11 to 103% more than the functional economizer. All delivered sensible cooling capacities satisfy the thermostat, are greater than the ACCA Manual N sensible cooling load highlighted in green.

Table 27: Non-Functional Economizer #4 at 70, 65, 60, and 55F OAT for 7.5-ton non-TXV RTU3 with Factory Charge at 333 scfm/ton (Calculated EER*s and Sensible Capacity)

Description	Test	Outdoor DB/WB (F)	Cycle 1 1st-Stage (min)	Off Cycle (min)	Cycle 2 1st-Stage (min)	Total kWh	Sensible Cooling Btu	Sensible EER*s
Closed Damper 1st-Stage Compressor	2-1CH	70/60	10.6	0.3	0.0	0.811	5,276	6.5
Closed Damper 1st-Stage Compressor	3-1CH Retest	65/57	4.7	1.6	2.6	0.540	3,930	7.3
Closed Damper 1st-Stage Compressor	4-1CH Retest	60/54	3.2	3.6	3.2	0.460	3,607	7.8
Closed Damper 1st-Stage Compressor	1-1CH Retest	55/51	2.5	0.8	0.0	0.173	1,433	8.3

Figure 8 provides the application sensible efficiency (EER*s) with functional and non-functional economizer at 70, 65, 60, and 55F OAT. The functional economizer plus 1st stage (C1) compressor is approximately 13% more efficient than non-functional economizer with closed dampers plus 1st stage compressor at 70/60 OAT. At 65/57 the economizer plus 1st-stage compressor is 28% more efficient than the non-functional economizer. At 60/54 the economizer fan only meets the load and is 66% more efficient than the non-functional economizer. At 55/51 the economizer fan only is 120% more efficient than the non-functional economizer.

Figure 8: Application Sensible Efficiency (EER*s) with Functional and non-Functional Economizer at 70, 65, 60, and 55F OAT for RTU3 with Factory Charge at 333 scfm/ton



These tests demonstrate that if the economizer functions properly it can increase cooling capacity and save energy when outdoor temperatures are less than 70F. The test results also demonstrate how economizer savings are related to minimum damper position. As minimum damper position increases from closed to 100% open, economizer savings would decline to zero.

4.1.7 Airflow Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of airflow faults on the application efficiency (EER*) of RTU3 with 1.5-HP blower motor and economizer #4 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at 82, 95, and 115F outdoor conditions and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. Airflow fault tests at 95F outdoor temperature are provided in **Table 28** including diagnostics for manufacturer and CEC refrigerant charge protocols. The sensible efficiency (EER*s) was greatest at 400 scfm/ton. For C1, the manufacturer ΔST protocol diagnosed correct charge for all tests, but misdiagnosed undercharge for C2. For the CEC ΔSH protocol diagnosed correct charge for circuit 1 closed damper tests and the 333 scfm/ton test with 1-finger open. The CEC ΔTS diagnosed low capacity at 400 cfm/ton and 333 scfm/ton with dampers 1-finger open. The sensible capacity was 7 to 11% lower than comparable tests. At 95F OAT, the 400 scfm/ton closed and 1-finger open tests and 333 scfm/ton closed damper test sensible cooling capacities were at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green. Tests not meeting ACCA Manual N minimum sensible capacity are highlighted in yellow or red.

Table 28: Airflow Fault Tests for 7.5-ton non-TXV RTU3 Economizer #4 Dampers Closed and 1-Finger Open at 95F

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	SP C1/C2 psig	ESP IWC	Airflow scfm/ton	Fan Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
3-4295CF (closed)	0/55	-2/34	-3.5	74/64	0.95	400	1740	58,806	6.6	52,934	5.9
3-5295CF (closed)	1/54	-3/34	-2.1	73/64	1.26	333	1490	54,967	6.3	48,128	5.5
3-6295CF (closed)	4/55	-5/33	-0.8	70/63	1.53	267	1420	49,350	5.8	41,447	4.8
3-42951CF (1-Finger)	8/55	7/33	-4.3	76/64	0.91	400	1720	52,629	5.9	50,032	5.6
3-52951CF (1-Finger)	0/55	-3/32	-2.8	75/63	1.21	333	1520	50,993	5.8	45,941	5.3
3-62951CF (1-Finger)	1/55	-8/30	-2.4	73/63	1.47	267	1410	41,422	4.8	37,728	4.4

Airflow fault tests at 82F outdoor temperature are provided in **Table 29**. The sensible EER*s is greatest at 400 scfm/ton. With 17 to 33% low airflow, sensible cooling capacity was reduced by 7 to 17% and efficiency was reduced by 6 to 14%. The manufacturer ΔST protocol only diagnosed correct charge for C1 (test 3-6282CF) at 267 scfm/ton with damper closed. The CEC ΔSH protocol only diagnosed correct charge for C1 (test 3-62821CF) at 267 scfm/ton with damper 1-finger open. For all other tests the manufacturer ΔST and CEC ΔSH protocols diagnosed "false alarm" undercharge for both circuits except for test 3-6282CF where the CEC ΔSH protocol diagnosed "false alarm" overcharge for C1. The CEC ΔTS protocol passes all tests and does not detect 33% low airflow. All tests have sensible cooling capacities at least 105% of the 35,978 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 29: Airflow Fault Tests for 7.5-ton non-TXV RTU3 with Economizer #4 Dampers Closed and 1-Finger Open at 82F OAT

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	SP C1/C2 psig	ESP IWC	Airflow scfm/ton	Fan Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
3-4282CF (closed)	17/59	12/31	-2.2	70/56	0.96	400	1710	69,781	8.8	57,345	7.3
3-5282CF (closed)	16/59	8/31	-0.5	69/56	1.28	333	1540	67,187	8.7	53,180	6.9
3-6282CF (closed)	5/58	-7/29	1.8	66/56	1.55	267	1410	62,802	8.3	47,548	6.3
3-42821CF (1-Finger)	17/59	12/31	-2.7	71/56	0.92	400	1710	66,426	8.4	55,800	7.1
3-52821CF (1-Finger)	17/59	9/30	-0.9	69/56	1.23	333	1530	63,778	8.3	51,594	6.7
3-62821CF (1-Finger)	11/58	-1/28	1.2	67/56	1.51	267	1420	59,739	7.9	46,207	6.1

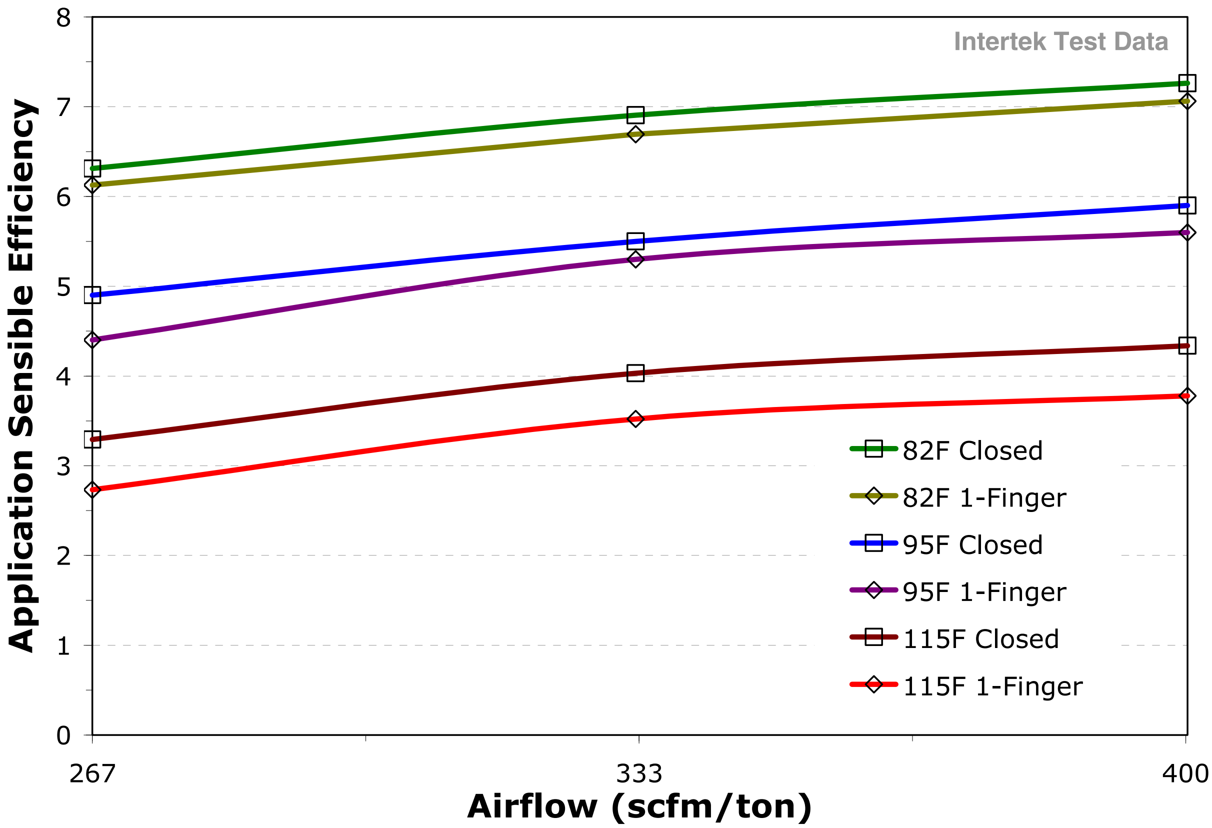
Airflow fault tests at 115F outdoor temperature are provided in **Table 30**. For all tests the sensible EER*s is greatest at 400 scfm/ton. Sensible cooling capacity and efficiency were reduced by 2 to 31% as airflow was reduced by 17 to 33%. For C1, the manufacturer ΔST protocol diagnosed correct charge for closed dampers and undercharge for 1-finger open. For C2, the manufacturer protocol misdiagnosed undercharge for all tests. The CEC ΔSH protocol diagnosed correct charge for C1 and misdiagnosed overcharge for C2. The CEC ΔTS protocol diagnosed low sensible cooling capacity for all tests caused by the 15 to 34% outdoor airflow and 115F OAT. All tests failed to meet the 61,132 Btuh ACCA Manual N sensible cooling load highlighted in red.

Table 30: Airflow Fault Tests for 7.5-ton non-TXV RTU3 with Economizer #4 Dampers Closed and 1-Finger Open at 115F OAT

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	SP C1/C2 psig	ESP IWC	Airflow scfm/ton	Fan Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
3-42115CF (closed)	-5/40	0/35	-5.2	80/75	0.92	400	1700	46,053	4.3	46,821	4.3
3-52115CF (closed)	-3/40	0/35	-4.0	78/74	1.23	333	1520	44,182	4.2	42,568	4.0
3-62115CF (closed)	0/42	0/37	-4.0	80/75	1.49	267	1390	35,040	3.4	34,079	3.3
3-421151CF (1-Finger)	-10/41	-1/35	-6.9	84/75	0.9	400	1700	34,502	3.2	40,970	3.8
3-521151CF (1-Finger)	-9/41	-1/34	-5.8	82/75	1.2	333	1500	33,589	3.2	37,417	3.5
3-621151CF (1-Finger)	-6/43	-3/35	-6.4	80/75	1.4	267	1410	23,197	2.2	28,478	2.7

Figure 9 shows EER*s increased by 15 to 38% with increasing airflow. The 7.5-ton non-TXV airflow tests need to be performed at different fan speeds using the code tester to match the fan ESP. The sensitivity of EER* to airflow as a percentage of the baseline 400 scfm/ton appears to be independent of temperature and outdoor airflow.

Figure 9: Application Sensible Efficiency Impacts of Airflow Faults for 7.5-ton non-TXV RTU3 with Factory Charge and Economizer #4 Dampers Closed and 1-Finger Open



The manufacturer provides information for "troubleshooting" and diagnosing insufficient evaporator airflow.[84] The most common symptom is suction pressure too low which can be caused by the following faults: 1) dirty air filter, 2) low refrigerant charge, 3) metering device or low-side restriction, 4) insufficient evaporator airflow, 5) temperature too low in conditioned space, 6) undersized unit (too much outdoor air), or 7) filter drier restriction. Technicians should first check and correct dirty air filter, thermostat set too low, and too much outdoor air. The remaining faults are undercharge, insufficient evaporator airflow, or restrictions. Refrigerant restrictions can be ruled out since the temperature drop across both filter driers are increased by 0.5 to 1.5F, evaporator saturation temperatures were above freezing at 36 to 42F, and suction

[84] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

pressures were within 10 to 13% of each other.[85] The C2 suction pressure (SP) was 5 to 21% less than C1 SP, and manufacturer and CEC protocols diagnosed C2 undercharged for all tests at 82, 95, and 115F OAT. Undercharge and insufficient airflow are difficult to diagnose from "false alarms" based on manufacturer and CEC protocols. The manufacturer ΔST and CEC ΔSH protocols diagnosed undercharge for 70 to 75% of tests, and CEC ΔTS protocols diagnosed low capacity for 44% of tests. For tests with economizer dampers 1-finger open, outdoor airflow was 26% and caused almost 40 to 50% of the reduction in cooling capacity. For these low-airflow-fault tests the cause of low evaporator airflow was high ESP, which wasn't in the manufacturer troubleshooting list. Field measurements of 15 commercial systems from 2 to 20 tons found average airflow of 337 +/- 29 cfm/ton and ESP of 1.2 +/- 0.3 IWC. Evaporator airflow at 350 cfm/ton with 1.2 IWC ESP is typical. There isn't any "rule of thumb" to correlate high ESP to low airflow for commercial systems. This is an example of the difficulty technicians have in properly diagnosing and correcting HVAC maintenance faults. If a technician was able to correctly diagnose insufficient airflow, the manufacturer's remedy is "increase air quantity or check filter and replace if necessary." Installing clean air filters and cleaning evaporator coils are important HVAC maintenance procedures, but solving a problem with high static pressure due to ductwork restrictions is generally beyond the scope of maintenance activities. Adding refrigerant charge to circuits that are at factory charge would produce little or no efficiency improvement.

4.1.8 Refrigerant Charge Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of refrigerant charge faults on the application efficiency (EER*) of RTU3 with economizer #4 installed and perimeter unsealed, dampers closed or 1-finger open, and airflow at 333 to 343 scfm/ton. Efficiency impacts are normalized per 100% factory charge. Two sets of refrigerant fault tests were performed. The first set was performed with unequal refrigerant charge percentages per circuit. The first faulted tests were performed in the vertical position and the non-faulted test (i.e., 100% factory charge) was performed in the horizontal position. The first set was performed with refrigerant charge varying from 80 to 160% of factory charge, outdoor temperatures of 95F, 82F, and 115F, return temperatures of 75F DB and 62F WB, and economizer dampers closed and 1-finger open. The second set of tests was performed with equal refrigerant charge percentages per circuit, and all tests were performed in the horizontal position. The second set was performed with refrigerant charge varying from 60 to 140% of factory charge, outdoor temperatures of 95F, return temperatures of 75F DB and 62F WB, and economizer dampers closed. For the first set of tests with dampers closed the unsealed outdoor air leakage was 17.3% and with dampers at 1-finger open the outdoor air leakage was 26% (see **Table 17**). For the second set of tests with dampers closed the unsealed outdoor air leakage was 16% (see **Table 18**). For the second tests, preliminary measurements were performed without code tester installed for each setup in order to match total static pressure with the code tester installed. Circuit-specific manufacturer refrigerant charge diagnostics are based on suction temperature (ST) as a function of outdoor

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[85] Tomczyk. J. 2002. Diagnosing A Restricted Liquid Line Can Be Tricky. AHRI News. http://www.achrnews.com/articles/90784-diagnosing-a-restricted-liquid-line-can-be-tricky

drybulb (DB) temperature (i.e., condenser entering air) and suction pressure (SP). The manufacturer ST tolerances and the CEC superheat (SH) tolerances are +/-5F. The CEC temperature split tolerances are +/-3F.

Table 31 provides refrigerant charge fault impacts and fault detection diagnostics (FDD) for the first set of tests at 95F OAT. **Table 32** provides results for the first set of tests at 115F OAT, and **Table 33** provides results for the first set of tests at 82F OAT. For the first set of tests, total and EER*s were maximized at 100% factory charge. For tests with closed dampers, undercharging by 20% reduced EER*s by 25 to 53%, and overcharging reduced EER*s by 1 to 16%. For tests with 1-finger open dampers, undercharging by 20% reduced EER*s by 24 to 66%, and overcharging reduced EER*s by 0 to 16% depending on OAT.

The manufacturer refrigerant charge protocols correctly diagnosed the 20% undercharge tests, but were only 38% accurate for these tests due to misdiagnosing overcharge or correct charge as undercharge. The CEC superheat protocols provided similar accuracy with 35% correct diagnoses for these tests. The CEC temperature split protocol provided 100% accuracy by correctly diagnosing low capacity for all undercharge tests and all tests at 115F OAT. It also correctly diagnosed proper airflow for all tests at 82F and 95F OAT. At 95F OAT, the 20% undercharge (closed or 1-finger open) and 20% overcharge (1-finger open) tests have sensible cooling capacities 10% less than non-faulted factory charge or less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red. All other tests have sensible capacities greater than ACCA Manual N highlighted in green or yellow. At 115F OAT, no tests meet the 61,132 Btuh ACCA Manual N sensible cooling load and all are highlighted in red. At 82F, the 20% undercharge (closed or 1-finger open) tests have sensible cooling capacities 10% less than non-faulted factory-charge capacity or less than the 35,978 Btuh ACCA Manual N sensible cooling load highlighted in red. All other tests at 82F have sensible capacities greater than the 35,978 Btuh ACCA Manual N sensible cooling load and highlighted in green or yellow.

At 115F OAT with 153% factory charge in C1 the discharge pressure (DP) is 350 psig. With 168% of factory charge in C2 the DP was approximately 400 psig. The manufacturer internal pressure relief valve is designed to open when the discharge pressure exceeds 375 to 450 psig.[86] The high pressure cut out is 428 psig and the reset is 320 psig.[87]

Figure 10 shows the application efficiency impacts versus refrigerant charge per factory charge with unequal percentage charge per circuit for the 7.5-ton non-TXV RTU3 with economizer #4 damper closed at 95F OAT. Total and EER*s are maximized at 100%+ factory charge. For the 20% undercharge, EER*s decreased by 33% and cooling capacity decreased by 52%. When overcharged by 20 to 60%, EER*s decreased by 2 to 8% and total power increased by 3 to 9%.

[86] Emerson Climate Technologies. 2011. Copeland Application Engineering Bulletin AE4-1374. ZR16 to ZR54K5E R-22 and R-407C 1.5 to 5 Ton Copeland Scroll® Compressors. January 2011.

[87] Carrier 2006. Product Data. WeatherMaster® 48HJ004-028 48HE003-006, Single-Package Rooftop Units, Gas Heating/Electric Cooling, 2 to 25 Nominal Tons. Page 17. http://dms.hvacpartners.com/docs/1009/Public/00/48H-1PD.pdf.

Figure 11 provides the application EER*s impact versus refrigerant charge per factory charge with unequal percentage charge per circuit for RTU3 with economizer #4 closed and 1-finger open at 82, 95, and 115F OAT. At 80% undercharge the EER*s was reduced by 24% (at 82F) to 66% (at 115F). At 120 to 160% overcharge the EER*s was reduced 1% (at 82F) to 17% (at 115F) depending on OAT. Higher outdoor air temperatures reduce efficiency and capacity due to hotter outdoor air entering the evaporator. Opening dampers from closed to 1-finger reduced efficiency by 3 to 36% depending on OAT.

Table 31: Refrigerant Charge Fault Impacts with Unequal Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed or 1-Finger Open at 95 OAT

Test	C1/C2 Refrig Charge %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Undercharge Closed										
7-3A (80%)	73/88	57/62	41/56	-8.9	337	8,021	26,390	3.3	29,304	3.7
Factory Charge Closed										
3-5295CF (100%)	100/100	1/54	-3/34	-2.1	333	8,727	54,967	6.3	48,128	5.5
Overcharge Closed										
5-295 recalc (120%)	112/128	3/33	-1/26	-2.1	337	8,910	53,313	6.0	48,393	5.4
7-4A (140%)	132/148	0/17	-2/15	-1.8	339	9,225	55,288	6.0	49,959	5.4
7-5A (160%)	153/168	-2/-5	-2/-3	-1.7	338	9,521	55,459	5.8	50,215	5.3
Undercharge 1-Finger										
8-3A (80%)	73/88	57/64	39/54	-10.0	338	8,021	18,744	2.3	26,050	3.2
Factory Charge 1-Finger										
3-52951CF (100%)	100/100	0/55	-3/32	-2.8	333	8,744	50,993	5.8	45,941	5.3
Overcharge 1-Finger										
5-2951 recalc (120%)	112/128	3/33	-4/25	-3.4	340	8,940	45,924	5.1	45,000	5.0
8-4A (140%)	132/148	-1/18	-5/16	-2.9	341	9,261	47,986	5.2	46,974	5.1
8-5A (160%)	153/168	-2/-8	-5/-6	-2.6	340	9,589	48,638	5.1	47,649	5.0

Table 32: Refrigerant Charge Fault Impacts with Unequal Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Closed or 1-Finger Open at 115 OAT

Test	C1/C2 Refrig Charge %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Undercharge Closed										
7-3F (80%)	73/88	62/70	45/60	-12.6	334	9,771	12,565	1.3	18,552	1.9
Factory Charge Closed										
3-52115CF (100%)	100/100	-3/40	0/35	-4.0	333	10,560	44,182	4.2	42,568	4.0
Overcharge Closed										
5-2115 recalc (120%)	112/128	0/19	3/26	-5.3	335	10,822	35,146	3.2	38,728	3.6
7-4F (140%)	132/148	0/-8	3/2	-5.0	337	11,182	35,781	3.2	40,016	3.6
7-5F (160%)	153/168	-2/-12	2/1	-5.5	338	11,466	33,695	2.9	39,061	3.4
Undercharge 1-Finger										
8-3F (80%)	73/88	62/74	44/62	-14.9	335	9,798	961	0.1	11,854	1.2
Factory Charge 1-Finger										
3-521151CF (100%)	100/100	-9/41	-1/34	-5.8	333	10,628	33,589	3.2	37,417	3.5
Overcharge 1-Finger										
5-21151 recalc (120%)	112/128	0/21	2/28	-6.9	336	10,857	26,716	2.5	34,326	3.2
8-4F (140%)	132/148	-1/-8	1/4	-6.7	339	11,279	24,841	2.2	35,506	3.1
8-5F (160%)	153/168	-2/-17	1/0	-7.1	340	11,661	23,825	2.0	34,342	2.9

Table 33: Refrigerant Charge Fault Impacts with Unequal Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Closed or 1-Finger Open at 82 OAT

Test	C1/C2 Refrig Charge %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Capacity Btuh	Sensible EER*s
Undercharge Closed										
7-3B (80%)	73/88	54/56	36/50	-6.3	339	7,093	41,291	5.8	36,976	5.2
Factory Charge Closed										
3-5282CF (100%)	100/100	16/59	8/31	-0.5	333	7,702	67,187	8.7	53,180	6.9
Overcharge Closed										
5-282 recalc (120%)	112/128	3/40	-5/23	-0.5	339	7,857	66,909	8.5	53,494	6.8
7-4B (140%)	132/148	-5/23	-8/14	0.6	340	8,186	71,505	8.7	57,321	7.0
7-5B (160%)	153/168	-8/-4	-8/-8	1.2	340	8,454	73,748	8.7	58,794	7.0
Undercharge 1-Finger										
8-3B (80%)	73/88	54/57	35/49	-6.7	340	7,103	38,916	5.5	36,109	5.1
Factory Charge 1-Finger										
3-52821CF (100%)	100/100	17/59	9/30	-0.9	333	7,708	63,778	8.3	51,594	6.7
Overcharge 1-Finger										
5-2821 recalc (120%)	112/128	4/40	-4/22	-0.8	341	7,870	64,314	8.2	52,649	6.7
8-4B (140%)	132/148	-5/23	-9/14	0.0	343	8,210	67,779	8.3	55,876	6.8
8-5B (160%)	153/168	-8/-5	-10/-9	0.8	341	8,499	71,602	8.4	57,969	6.8

Figure 10: Application Efficiency Impacts versus Refrigerant Charge per Factory Charge with Unequal Percentage Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 95F OAT

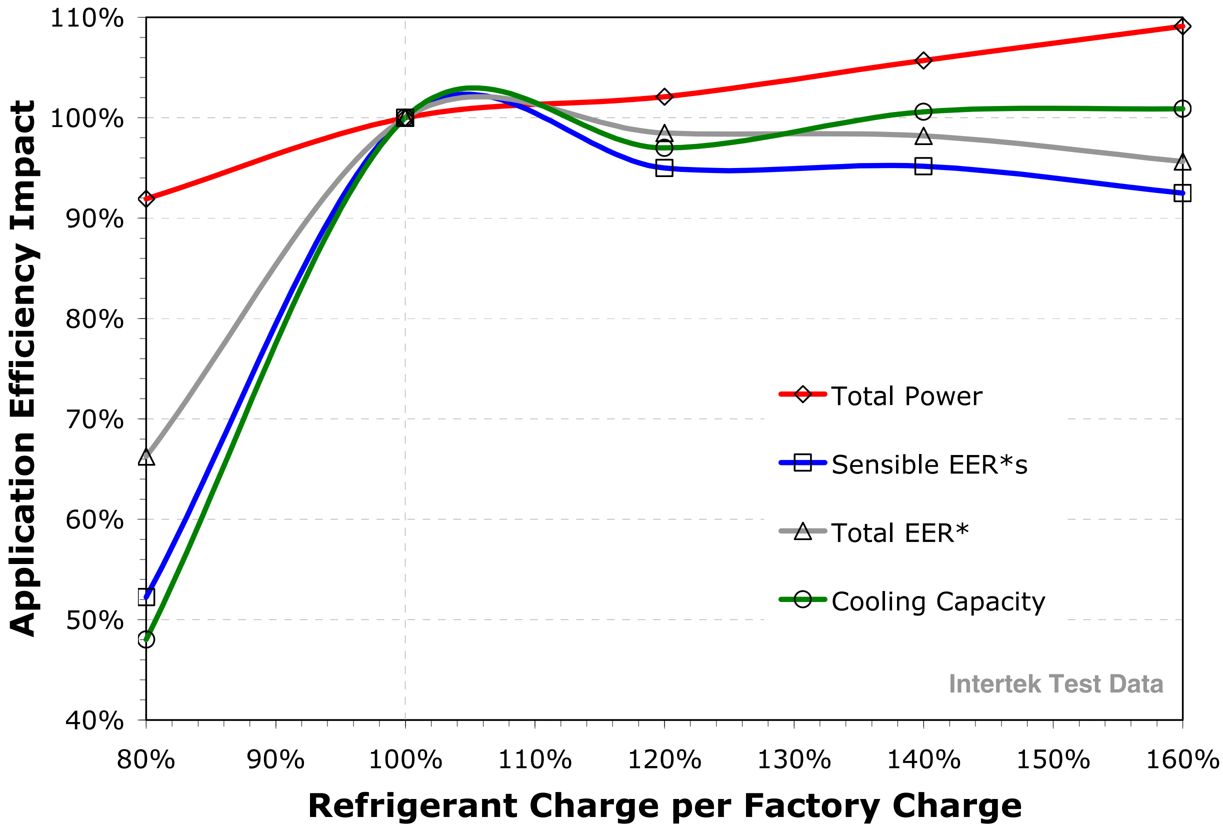
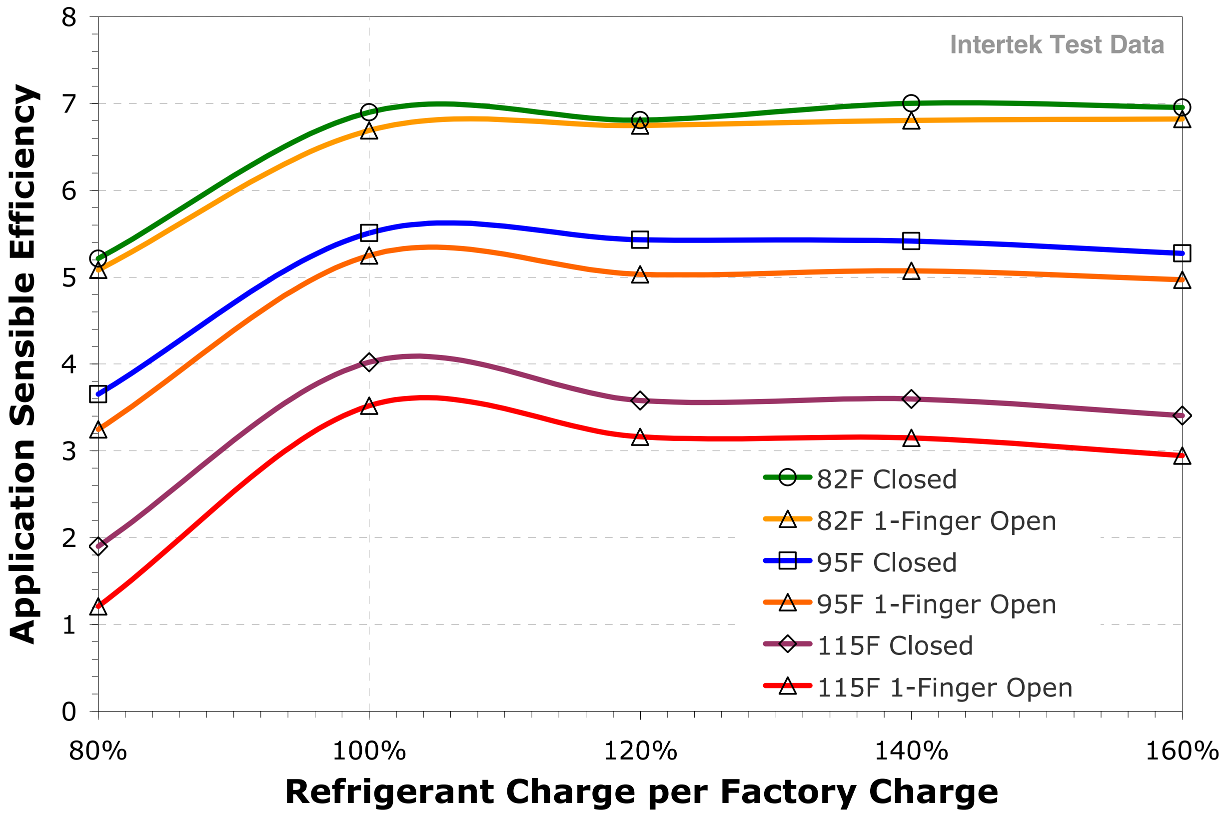


Figure 11: Application Sensible Efficiency Impacts versus Refrigerant Charge per Factory Charge with Unequal Percentage Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Closed and 1-Finger Open at 82, 95, and 115F OAT



One important purpose of laboratory testing is to develop accurate regression equations of sensible energy efficiency (EER^*_s) versus refrigerant charge per factory charge in order to evaluate energy efficiency improvements of refrigerant charge adjustment measures implemented in HVAC maintenance programs. The regression equations can be used to calculate EER*s impacts associated with refrigerant charge adjustments based on recovery and weigh-out of refrigerant charge and the reported charge adjustment per circuit. Regression lines cannot be accurately fitted through the zero intercept and sensible energy efficiency data shown in **Figure 10** and **Figure 11**. Therefore, second tests of refrigerant charge fault impacts were performed on RTU3 with equal percentage charge per circuit relative to factory charge at 95F OAT.

Table 34 provides the second tests of refrigerant charge fault impacts with equal percentage charge per circuit for the 7.5-ton non-TXV RTU3 at 95F OAT. Total EER* and EER*s were maximized from 100 to 130% of factory charge. Undercharging refrigerant by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% increased EER*s by 0 to 2%. The manufacturer protocols correctly diagnosed undercharge for both circuits and overcharge for circuit 2 at 130 to 140% factory charge. The CEC superheat protocols correctly diagnosed undercharge for both circuits. The CEC temperature split protocol provided 100% accuracy by correctly diagnosing low cooling capacity for all tests less than 110% of factory charge and correct airflow and capacity for 110 to 140% of factory charge. The undercharge and factory charge tests had sensible cooling capacities less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red. All other tests had sensible capacities greater than the ACCA Manual N sensible cooling load highlighted in yellow or green.

Table 34: Second Tests of Refrigerant Charge Fault Impacts with Equal Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 and Damper Closed at 95F

Test	C1/C2 Refrig Charge %	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-R60-95-CE	60%	84/62	78/51	-11.6	325	7,742	15,461	2.00	21,026	2.72
C8-R70-95-CE	70%	82/61	65/41	-8.6	326	7,988	27,633	3.46	29,440	3.69
C8-R80-95-CE	80%	82/55	56/35	-6.4	326	8,222	37,038	4.51	35,718	4.34
C8-R90-95-CE	90%	81/45	47/31	-4.8	327	8,451	44,381	5.25	40,386	4.78
C8-R95-95-CE	95%	76/40	44/29	-4.1	327	8,555	47,231	5.52	42,325	4.95
C8-R100-95-CE-1	100%	67/33	40/25	-3.4	330	8,652	50,340	5.82	44,673	5.16
C8-R105-95-CE	105%	64/30	39/24	-3.1	328	8,750	51,188	5.85	45,361	5.18
C8-R110-110C2-95-CE	110%	61/24	38/21	-2.7	329	8,855	52,542	5.93	46,352	5.23
C8-R120-95-CE	120%	54/11	35/11	-2.1	325	9,046	54,899	6.07	47,532	5.25
C8-R130-95-CE	130%	48/-5	32/-4	-1.7	326	9,268	56,291	6.07	48,789	5.26
C8-R140-95-CE	140%	40/-7	28/-5	-1.3	325	9,447	56,826	6.02	49,643	5.25

Figure 12 shows the second tests of application efficiency impacts versus refrigerant charge per factory charge with equal percentage charge per circuit for the 7.5-ton non-TXV RTU3 at 95F OAT. Total EER* and EER*s were maximized from 100 to 130% of factory charge. From 60 to 95% of factory charge the EER*s is reduced by 4 to 47%. From 105 to 140% of factory charge the EER*s is increased by 1 to 2%. Refrigerant overcharge from 5 to 40% doesn’t appear to have any negative impact on energy efficiency or cooling capacity. However overcharging can cause liquid refrigerant to enter the compressor at start-up which can cause compressor failure.

The regression equation curve-fit of sensible energy efficiency (EER*s) versus refrigerant charge per factory charge ratio for RTU3 at 95F OAT is shown in **Figure 12**. **Equation 10** and **Equation 11** can be used to calculate EER*s impacts associated with refrigerant charge adjustments based on recovery and weigh-out of refrigerant charge and the reported charge adjustment (assuming the reported charge adjustment is correct).

Equation 10 $$y = 5.218x^6 - 28.455x^5 + 63.204x^4 - 71.831x^3 + 41.208x^2 + 8.358x$$

Where,

y = EER*s impact at refrigerant charge per factory charge ratio (dimensionless)

x = refrigerant charge per factory charge ratio (dimensionless)

Equation 11 $y_{rc} = y_r - y_o$

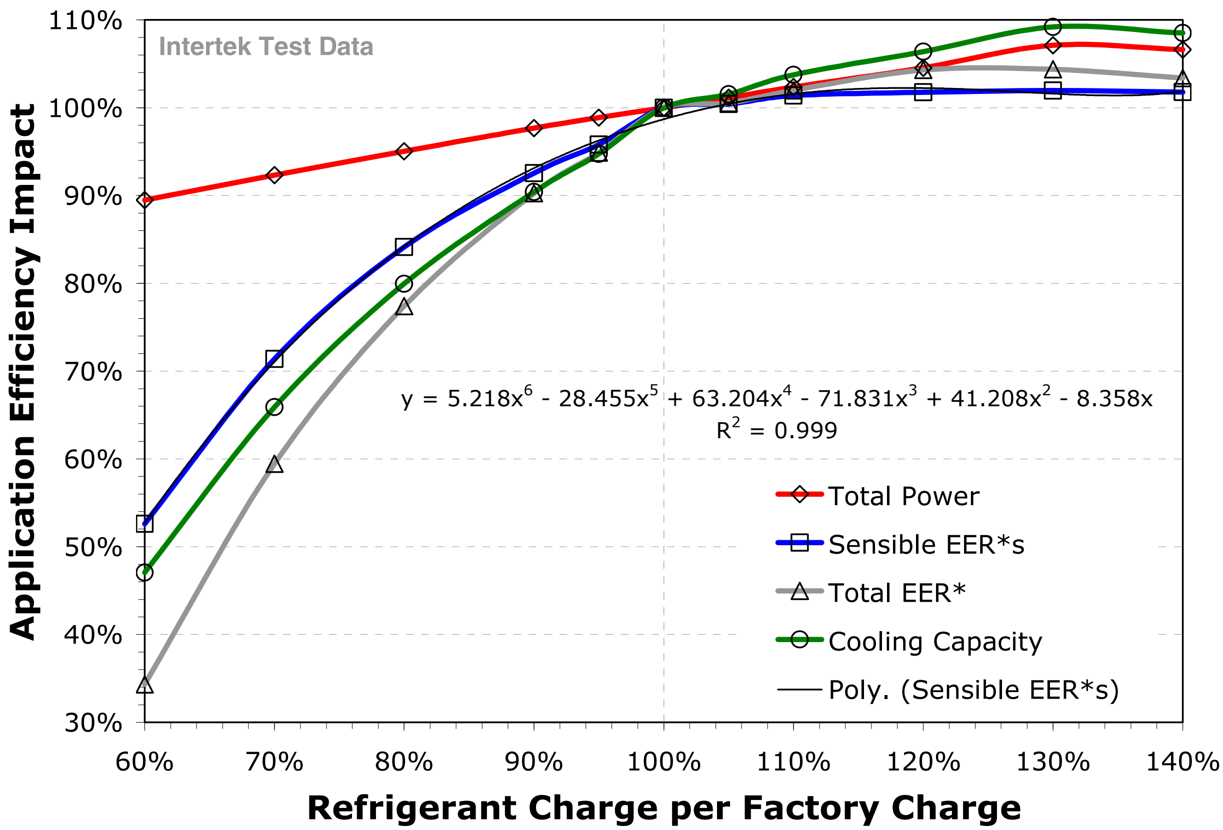
Where,

y_{rc} = EER*s impact of refrigerant charge measure (dimensionless)

y_r = EER*s impact at refrigerant charge per factory charge ratio (dimensionless)

y_o = EER*s impact at recovered refrigerant charge per factory charge (dimensionless)

Figure 12: Second Tests of Application Efficiency Impacts versus Refrigerant Charge per Factory Charge with Equal Charge per Circuit for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 95F OAT



Severe undercharge causes icing of the evaporator coil as evaporator saturation temperature decreases below freezing causing water that condenses on the coil to freeze into ice. Coil icing reduces airflow which decreases efficiency even more. Icing of the coil was avoided while performing undercharge tests by operating unit with fan only (no compressors) in between tests, checking evaporator coil to make sure no ice was present and condensate pan was dry. Overcharging can cause liquid refrigerant to flood the compressor during normal operation and start-up which dilutes oil causing inadequate bearing lubrication and premature failure.

The manufacturer provides information for "troubleshooting" and diagnosing refrigerant charge faults.[88] Procedures for troubleshooting and servicing air conditioning systems are also provided in technician training text books.[89] The most common problems are high or low discharge or suction pressure or continuous compressor operation. These problems are caused by a number of faults including: 1) dirty air filter, 2) blocked evaporator/condenser, 3) undersized unit (low cooling capacity or excessive outdoor air), 4) insufficient evaporator airflow, 5) refrigerant restriction, 6) non-condensables, 7) thermostat defective/set too low, 8) low line voltage (faulty contactor/transformer), 9) defective compressor/overload, or 10) refrigerant over/undercharge. Prior to adjusting refrigerant charge, technicians need to check and correct all other faults on the list. If none of the other faults are present and problem still exists, then refrigerant charge adjustments might be necessary.

4.1.9 Evaporator Blockage Fault Tests for 7.5-ton non-TXV RTU3

Evaporator blockage is primarily caused by dirty filters (DF) and secondarily caused by dirty coils (DC). Filters are designed to remove dirt from air to maintain indoor air quality. The Minimum Efficiency Reporting Value (MERV) is used to rate filters. Most commercial buildings have MERV 8 pleated filters to remove particles such as pollen, mold, and dust (3 to 10 microns).[90] ACCA 180 requires quarterly filter replacement to avoid excessive filter loading and maintain adequate airflow and energy efficiency. Dirty filters reduce airflow and sensible cooling capacity causing the air conditioner to operate longer to satisfy the drybulb thermostat. This causes the unit to operate longer or continuously. Low airflow causes reduced evaporator temperatures where the filter temperature can drop below the dew point causing moisture to collect on the coil and the filter. Moisture on the filter mixes with dirt further reducing airflow and increases condensation on the filter. Water on the coil and filter eventually freeze and the entire filter and coil can form a partial blockage of ice which reduces airflow, sensible capacity, and efficiency even more. **Figure 13** shows a filter and coil blocked with ice due to dirty filters. Filter blockage causes longer operational time and icing of the filter and coils which were not tested in the laboratory. Some evaporator blockage tests had to be paused between tests to operate the fan by itself to melt ice off the coil and filer. The ACCA 180 standard addresses this problem by requiring quarterly air filter replacement. The California HVAC maintenance program work papers did not fully understand the energy efficiency impacts of dirty filters. Consequently, no energy savings were claimed for quarterly filter replacement per ACCA 180 which is required in the programs. Field measurements of dirty filters cleaned 12.6 months previously found an average airflow reduction of 5.2% corresponding to 27% blockage and 4.4%

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[88] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

[89] Tomczyk, J. 1995. Troubleshooting and Servicing Modern Air Conditioning and Refrigeration Systems. ESCO Press. Mt. Prospect, Ill.: Educational Standards Corporation.

[90] ASHRAE Standard 52.5. 2006. Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size.

EER*s short-term impact. Dirty filters cause even larger EER*s impacts in the field due to longer operational times which were not measured in the laboratory.

Figure 13: Dirty Frozen Air Filter and Evaporator Coil Caused by Excessive Blockage

Laboratory tests were performed to evaluate the short-term impact of evaporator coil blockage faults on the application efficiency (EER*) of RTU3 with economizer #4 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. Evaporator coil blockage occurs over time as dirt and debris in the return air or outdoor air are deposited on the air filter and coil. The evaporator coil was blocked with plastic corrugated cardboard on the upstream side next to the air filter. The inlet area was blocked 5 to 50% to reduce evaporator airflow by 0.8 to 12.2%. Preliminary tests were performed without code tester installed for each coil blockage setup in order to match total static pressure with the code tester installed.

Evaporator coil blockage test results are provided in **Table 35** at 95F OAT. Diagnostic test results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols.

Table 35: Evaporator Coil Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 95F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Suction Press psig C1/C2	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-EB0-95-CE	0.0%	53/39	35/29	-3.2	328	64/71	8,802	5.69	44,855	5.10
C8-EB5-95-CE	-0.8%	53/38	34/27	-3.3	326	63/71	8,777	5.65	44,313	5.05
C8-EB10-95-CE	-2.2%	53/37	33/26	-3.3	321	63/71	8,741	5.64	43,792	5.01
C8-EB20-95-CE	-3.7%	53/37	33/25	-3.2	316	63/70	8,706	5.67	42,746	4.91
C8-EB35-95-CE	-7.7%	53/36	31/23	-2.8	303	62/69	8,626	5.80	41,922	4.86
C8-EB50-95-CE	-12.2%	53/34	30/19	-2.4	288	61/67	8,552	5.83	41,135	4.81

Figure 14 shows the application efficiency impacts versus evaporator airflow decrease due to blockage. Application sensible efficiency (EER*s) was maximized with no blockage. Evaporator coil blockage reduced EER*s by 1 to 4% and sensible capacity by 1 to 7%. **Equation 12** is the regression equation shown in **Figure 14**. **Equation 12** can be used to calculate the EER*s impact associated with blocked/clean airflow ratio due to evaporator coil blockage for the 7.5-ton non-TXV unit.

Equation 12 $y_e = -0.54 x_a + 1.0$

Where,

y_e = EER*s impact of evaporator coil blockage based on airflow ratio decrease (dimensionless)

$x_a = \frac{cfm_b}{cfm_c} - 1$ = airflow ratio decrease due to evaporator coil blockage (dimensionless)

cfm_b = evaporator airflow with blocked evaporator coil (cfm)

cfm_c = evaporator airflow with clean evaporator coil (cfm)

Table 36 and **Table 37** show evaporator coil blockage test results at 115F and 82F OAT. The manufacturer and CEC refrigerant charge protocols misdiagnosed undercharge for all tests. The CEC temperature split (ΔTS) protocol correctly diagnosed low capacity for all tests at 115F and all tests except 35 and 50% coil blockage at 95F. The CEC TS protocol misdetected low airflow for tests at 35 and 50% coil blockage. All tests at 95F and 115F had sensible cooling capacities less than the ACCA Manual N sensible cooling load of 45,024 and 61,132 Btuh highlighted in red. All tests at 82F OAT had sensible cooling capacities greater than 105% of the 35,978 Btuh ACCA Manual N sensible cooling load highlighted in green.

Figure 14: Application Efficiency Impacts versus Evaporator Airflow Decrease due to Coil Blockage for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 95F

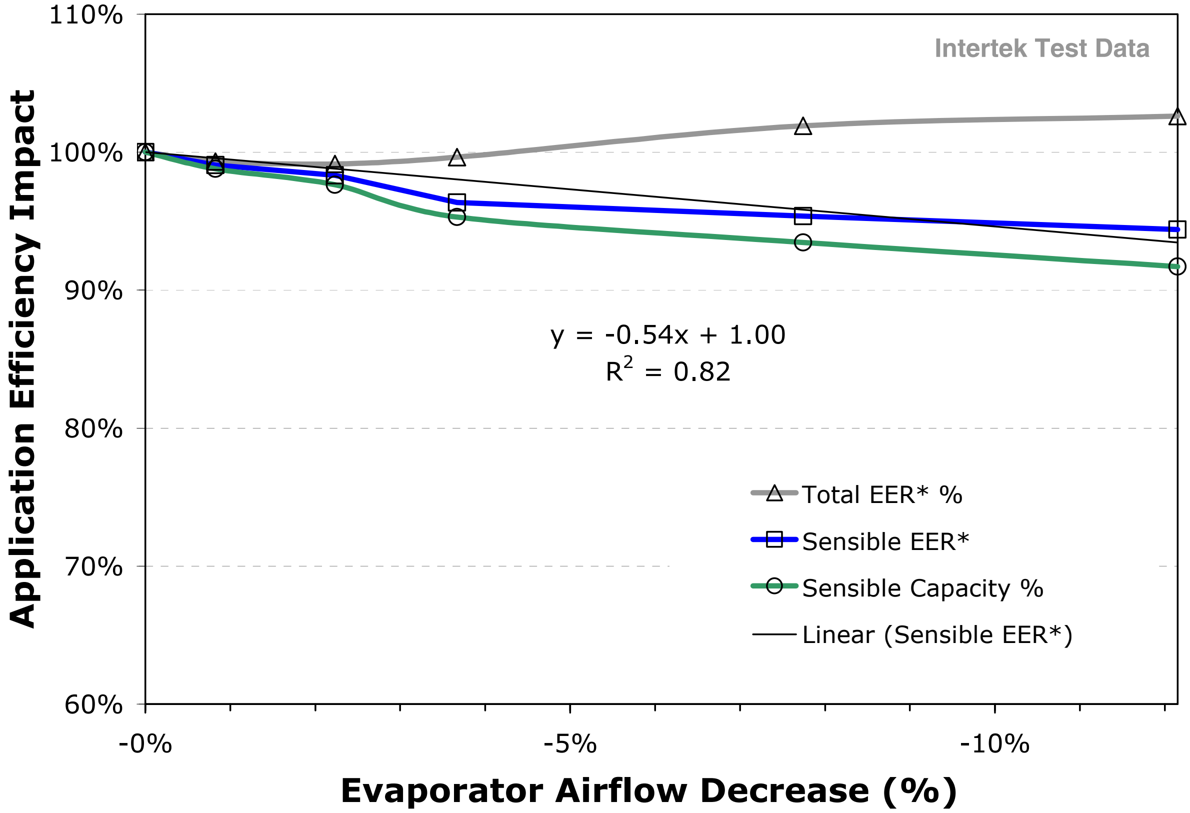


Table 36: Evaporator Coil Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 115F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Suction Press psig C1/C2	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-EB0-115-CE	0.0%	47/24	39/28	-4.6	327	70/82	10,508	3.91	40,779	3.88
C8-EB5-115-CE	-1.5%	47/23	38/26	-4.6	322	70/81	10,477	3.90	40,195	3.84
C8-EB10-115-CE	-2.5%	47/23	38/25	-4.6	318	70/81	10,452	3.91	39,826	3.81
C8-EB20-115-CE	-4.5%	46/20	37/22	-4.5	312	69/80	10,421	3.92	39,333	3.77
C8-EB35-115-CE	-8.7%	45/17	35/16	-4.3	298	68/78	10,355	3.91	38,039	3.67
C8-EB50-115-CE	-12.9%	44/7	33/4	-4.3	284	67/76	10,280	3.88	36,473	3.55

Table 37: Evaporator Coil Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 82F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Suction Press psig C1/C2	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-EB0-82-CE	0.0%	62/41	33/23	-1.9	332	56/64	7,622	7.99	49,317	6.47
C8-EB5-82-CE	-1.6%	61/40	32/22	-1.8	326	56/63	7,595	7.99	48,856	6.43
C8-EB10-82-CE	-2.9%	61/40	31/21	-1.8	322	56/63	7,570	7.95	48,148	6.36
C8-EB20-82-CE	-4.7%	61/39	30/19	-1.7	316	55/62	7,546	7.92	47,509	6.30
C8-EB35-82-CE	-8.5%	60/38	28/16	-1.4	304	55/61	7,486	7.95	46,580	6.22
C8-EB50-82-CE	-12.9%	60/37	27/15	-0.8	289	54/61	7,429	8.01	45,923	6.18

This manufacturer provides "troubleshooting" procedures to diagnose evaporator blockage faults.[91] The most common problem is low suction pressure caused by the following faults: 1) dirty air filter and evaporator coil, 2) low refrigerant charge, 3) metering device or low-side restriction, 4) insufficient evaporator airflow, 5) temperature too low in conditioned space, or 6) filter drier restriction. Technicians can easily check and correct dirty air filter and clean the evaporator coil. If these maintenance procedures eliminate low suction pressure faults, then there is no reason for additional FDD or correction. These tests indicate the importance of technicians following systematic procedures of checking and correcting obvious maintenance faults such as evaporator coil cleaning and installing clean air filters before performing FDD services.

Figure 15 shows an evaporator coil with dirt blocking about 50% of the coil. The air filter was also blocked with dirt. **Figure 16** shows an evaporator coil where one of the dirty air filters dropped into the economizer return damper blocking airflow and functionality of the damper.

[91] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

Figure 15: Evaporator Coil Blockage with Dirt Blocking 50% of the Coil

Figure 16: Evaporator Blockage with Dirty Filter Blocking Economizer Return Damper

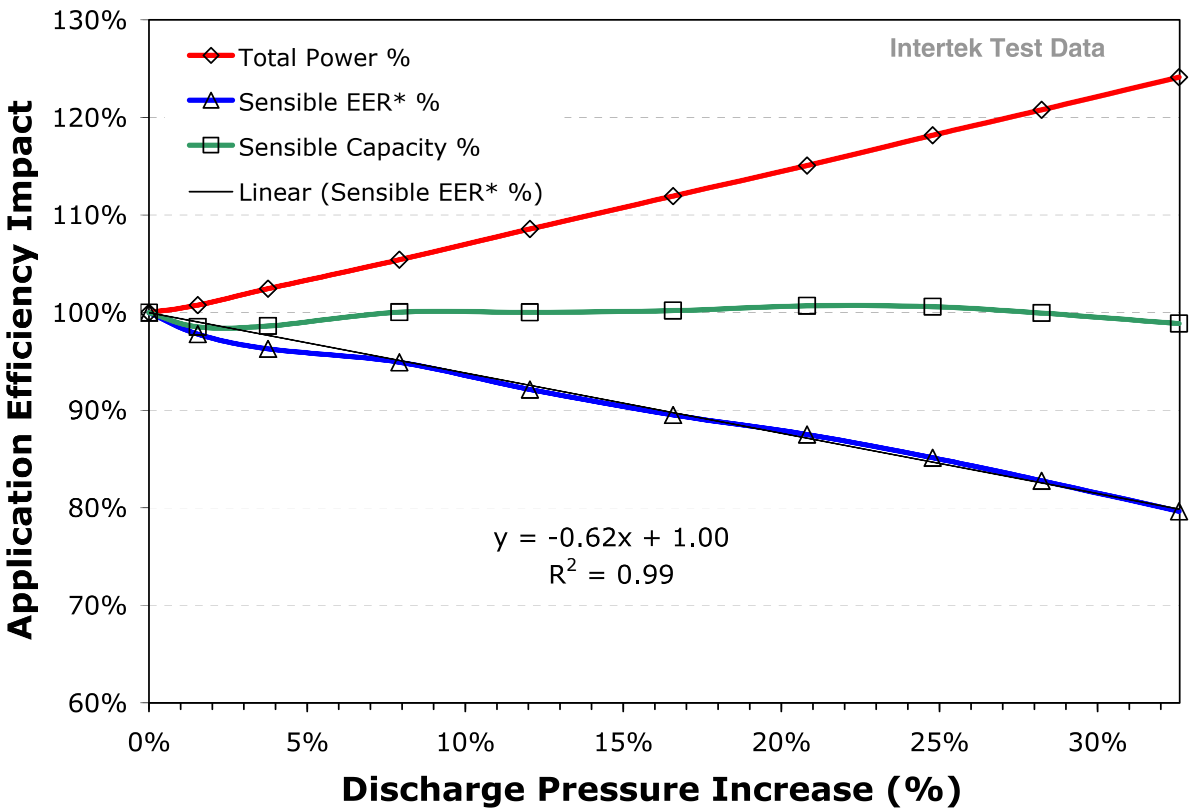
4.1.10 Condenser Blockage Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of condenser blockage faults on the application efficiency (EER*) of RTU3 with economizer #4 installed, dampers closed, economizer perimeter unsealed, and airflow of ~330 scfm/ton. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. The condenser coil was blocked on the outside of the coil with plastic corrugated cardboard used to ship condensers (to block but not damage fins). Coil blockage was increased by 5 to 80% to produce a 1 to 40% average discharge pressure increase across both refrigerant circuits. Condenser blockage test results are provided in **Table 38**. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 17** shows the application efficiency impacts versus discharge pressure increase due to condenser coil blockage at 95F OAT. Total EER* and EER*s are maximized with no blockage. Condenser coil blockage increased discharge pressure by 2 to 40%, increased compressor power by 2 to 26%, and reduced EER*s capacity by 1 to 20% compared to no blockage.

Table 38: Condenser Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed and 328 scfm/ton at 95F OAT

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-EB0-95-CE	100.0%	53/39	35/29	-3.2	328	258/268	8,802	5.69	44,855	5.10
C8-CB5-95-CE	101.5%	49/36	33/28	-3.5	329	261/273	8,868	5.31	44,197	4.98
C8-CB10-95-CE	103.8%	47/34	33/27	-3.5	331	267/279	9,017	5.25	44,241	4.91
C8-CB20-95-CE	107.9%	43/28	32/24	-3.3	330	277/291	9,278	5.17	44,871	4.84
C8-CB30-95-CE	112.0%	40/25	31/23	-3.3	329	287/302	9,556	4.97	44,861	4.69
C8-CB40-95-CE	116.6%	38/19	30/20	-3.2	328	296/317	9,853	4.79	44,942	4.56
C8-CB50-95-CE	120.8%	34/15	28/18	-3.0	327	308/328	10,127	4.66	45,167	4.46
C8-CB60-95-CE	124.8%	31/10	27/14	-3.0	326	317/339	10,401	4.48	45,125	4.34
C8-CB70-95-CE	128.2%	29/6	26/11	-3.0	324	326/348	10,631	4.32	44,832	4.22
C8-CB80-95-CE	132.6%	26/1	25/7	-3.1	323	336/361	10,926	4.10	44,343	4.06

Figure 17: Application Efficiency Impacts versus Discharge Pressure Increase due to Condenser Blockage for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed and 330 scfm/ton at 95F OAT



Equation 13 is the regression equation shown in **Figure 17**. **Equation 13** can be used to calculate the EER*s impact associated with blocked/clean discharge pressure ratio increase at constant OAT due to condenser coil blockage for the 7.5-ton non-TXV unit.

Equation 13 $$y_c = -0.62x_p + 1.0$$

Where,

y_c = EER*s impact of condenser coil blockage based on discharge pressure ratio increase (dimensionless)

$x_p = \frac{DP_b}{DP_c} - 1$ = discharge pressure (DP) ratio increase due to condenser coil blockage (dimensionless)

DP_b = blocked condenser coil discharge pressure DP (psig)

DP_c = clean condenser coil discharge pressure DP (psig)

Table 39 and **Table 40** provide condenser coil blockage test results at 115F and 82F OAT. The manufacturer refrigerant charge protocols misdiagnosed undercharge for all tests except circuit 2 for 80% coil blockage at 95F and 30% coil blockage at 115F OAT. The CEC refrigerant charge protocols misdiagnosed undercharge for all tests except circuit 2 for 40 and 50% coil blockage at 115F OAT. The CEC temperature split (ΔTS) protocol correctly diagnosed proper airflow and low capacity for all tests except 70% coil blockage at 95F OAT. All tests at 95F OAT except 60 and 70% coil blockage had sensible cooling capacities less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red or yellow. All tests at 115F OAT are less than the 61,132 Btuh ACCA Manual N sensible cooling load highlighted in red. All tests at 82F OAT had sensible cooling capacities greater than 105% of the 35,978 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 39: Condenser Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed and 326 scfm/ton at 115F OAT

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-CB0-115-CE	100.0%	53/18	41/24	-4.5	326	321/347	10,548	3.90	40,768	3.86
C8-CB5-115-CE	102.2%	45/18	39/24	-4.2	326	329/354	10,734	3.88	41,681	3.88
C8-CB10-115-CE	104.0%	43/15	38/22	-4.4	327	334/360	10,887	3.77	41,286	3.79
C8-CB20-115-CE	107.0%	40/8	36/16	-4.4	327	344/371	11,154	3.62	41,357	3.71
C8-CB30-115-CE	111.0%	38/3	36/12	-4.6	326	356/385	11,521	3.36	40,656	3.53
C8-CB40-115-CE	114.9%	37/-10	35/2	-4.9	327	364/402	11,881	3.14	40,036	3.37
C8-CB50-115-CE	117.6%	35/-12	34/1	-4.9	326	373/412	12,131	3.02	39,850	3.28

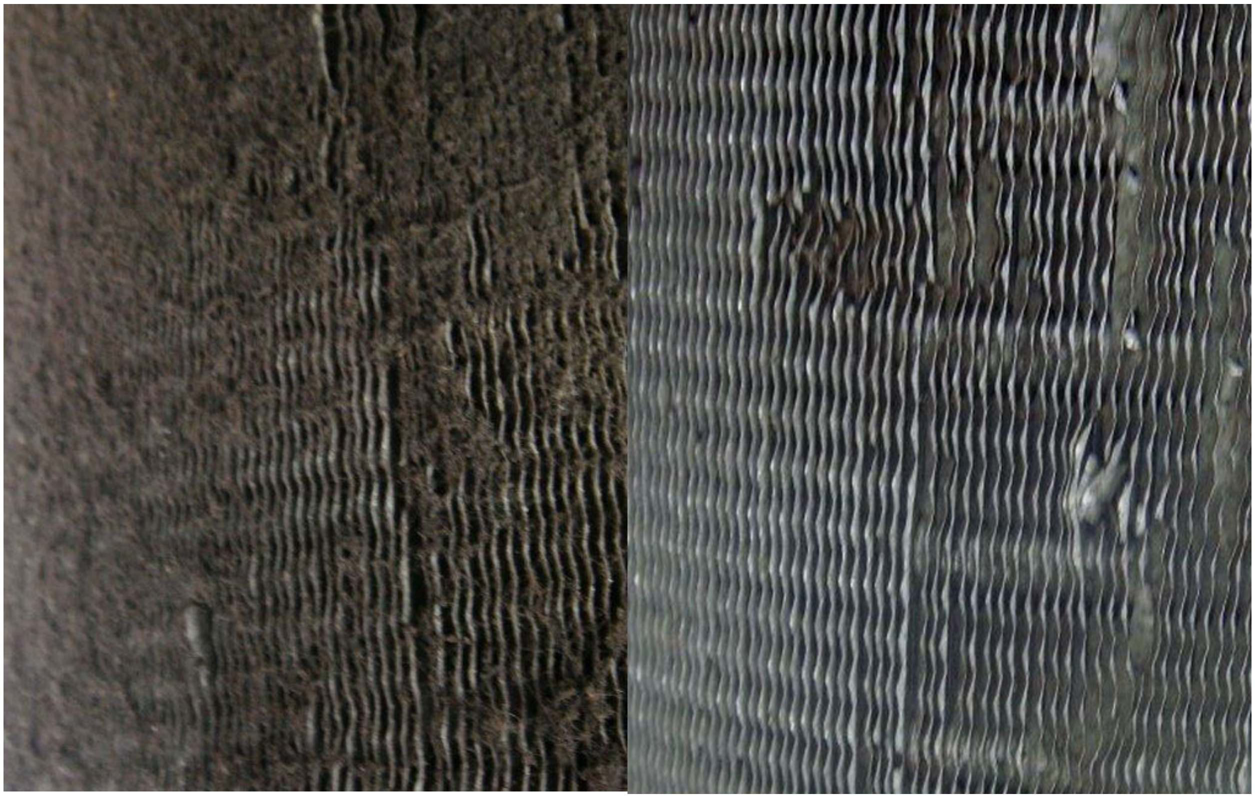
Table 40: Condenser Blockage Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed and 330 scfm/ton at 82F OAT

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*
C8-CB0-82-CE	100.0%	61/41	33/23	-1.9	330	207/221	7,623	7.89	48,952	6.42
C8-CB5-82-CE	102.1%	59/38	32/22	-1.6	333	211/226	7,734	7.91	50,084	6.48
C8-CB10-82-CE	104.0%	58/36	32/21	-1.6	330	214/231	7,815	7.84	49,754	6.37
C8-CB20-82-CE	107.6%	54/32	30/19	-1.4	331	221/239	7,994	7.75	50,438	6.31
C8-CB30-82-CE	112.3%	52/28	29/17	-1.4	331	230/250	8,240	7.52	50,559	6.14
C8-CB40-82-CE	116.6%	49/23	28/15	-1.2	329	238/260	8,447	7.36	50,841	6.02
C8-CB50-82-CE	120.9%	47/20	28/14	-1.1	329	246/271	8,663	7.17	50,929	5.88
C8-CB60-82-CE	124.8%	45/17	27/12	-1.0	331	253/280	8,872	7.05	51,531	5.81
C8-CB70-82-CE	131.0%	43/12	27/9	-1.1	329	264/296	9,196	6.72	51,213	5.57
C8-CB80-82-CE	140.1%	44/-10	27/-9	-1.3	327	270/329	9,659	6.22	50,290	5.21

This manufacturer provides “troubleshooting” procedures to diagnose condenser blockage from other faults.[92] The most common problem is excessive head pressure caused by the following faults: 1) dirty air filter, 2) dirty condenser coil, 3) refrigerant overcharge, 4) air in system (non-condensables), and 5) condenser air restricted or short-cycling. Technicians can easily check and correct dirty air filters and dirty (or blocked) condenser. If these corrections eliminate excessive head pressure, then there is no reason to adjust refrigerant charge or check for non-condensables.

[92] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 34, Troubleshooting. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

Figure 18: Field Observed Condenser Coil Blockage with Dirt Causing 28% Discharge Pressure Increase (left photo shows before and right photo shows after cleaning)



4.1.11 Restriction Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of refrigerant line restrictions on the application efficiency (EER*) of RTU3 with economizer #4 installed, dampers closed, economizer perimeter unsealed, and airflow of 267, 333, and 400 scfm/ton. The unsealed outdoor airflow rate for all tests was 16.7 to 22.5%. Tests were performed at 82F, 95F and 115F outdoor temperatures, and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In order to emulate liquid line restriction faults a "service" valve was installed upstream of the liquid line driers on each circuit. The service valve was partially closed to cause a suction pressure reduction and liquid line refrigerant temperature reduction to emulate a restriction at the liquid line drier or expansion device.[93] Refrigerant line restriction tests imposed a 34 to 44 psig suction pressure drop and 15 to 20F liquid temperature drop across the restriction. The restriction reduced suction pressure by 34 to 44% and evaporator saturation temperature was below freezing for all tests causing icing of the coil.

[93] Restriction "service" valve turns closed were recorded for each test.

Restriction test results at 95F OAT are provided in **Table 41**. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 19** shows the application efficiency impact versus OAT with the circuit 1 restriction normalized to 95F OAT and 333 scfm/ton airflow with economizer #4 and dampers closed. The restriction causes EER*s to decrease by 12 to 36% depending on OAT. **Figure 20** shows the application efficiency impacts versus evaporator airflow normalized to 400 scfm/ton for the C1 restriction at 95F OAT with economizer #4 and dampers closed. At 95F the restriction causes EER*s to decrease by 25 to 31% as a function of evaporator airflow. Restrictions cause short-cycling as indicated by the C1 restriction causing increased discharge temperatures of 298F which is within the compressor manufacturer discharge cut-out temperature limit of 295 +/- 7F.[94] The restriction caused a 346 psig discharge-to-suction pressure differential, and the manufacturer internal pressure relief valve is designed to open when the pressure differential exceeds 375 to 450 psig. The C1 suction pressure is 45 to 63% lower and sensible cooling capacity is 15 to 39% lower than the unrestricted tests.

Table 41: Restriction Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 95F OAT (Circuit 1 Restriction)

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Filter Drier ΔT C1/C2	SP C1/C2	EST C1/C2	Airflow scfm/ ton	Sensible Cooling Capacity Btuh	Total Power W	Total EER*	Sensible EER*s
No Restriction											
3-4295CF	0/55	-2/35	-3.5	-2/-1	74/64	44/36	400	52,934	8,923	6.6	5.9
3-5295CF	1/54	-3/34	-2.1	-2/-1	73/64	43/36	333	48,128	8,727	6.3	5.5
3-6295CF	4/55	-5/33	-0.8	-2/-1	70/63	41/36	267	41,447	8,550	5.8	4.8
C1 Restriction											
3-3000-95	64/55	58/35	-8.9	17/0	32/63	9/36	400	34,656	8,433	4.1	4.1
3-2500-95	69/55	54/34	-7.3	17/0	32/63	8/36	333	33,364	8,241	4.4	4.0
3-2000-95	11/54	0/31	-6.1	18/0	27/62	3/35	267	29,533	8,074	4.2	3.7

94 Emerson Climate Technologies. 2011. Copeland Application Engineering Bulletin AE4-1374. ZR16 to ZR54K5E R-22 and R-407C 1.5 to 5 Ton Copeland Scroll® Compressors. January 2011. Emerson Climate Technologies. 2001. Copeland Scroll Application Guidelines. Scroll Compressors for Air-Conditioning ZR 18 K4*... ZR 81 KC*. C060201/0702_1002/E. Emerson 2010. AE-1280 Application Guidelines for Copeland® Compliant Scroll Compressors (ZR*1 Models). Emerson Climate Technologies. 1675 West Campbell Road, Sidney, OH 45365.

Figure 19: Application Efficiency Impacts versus OAT with Circuit 1 Restriction Normalized to 95F OAT for 7.5-ton non-TXV RTU3 with Economizer #4 and Damper Closed and 333 scfm/ton

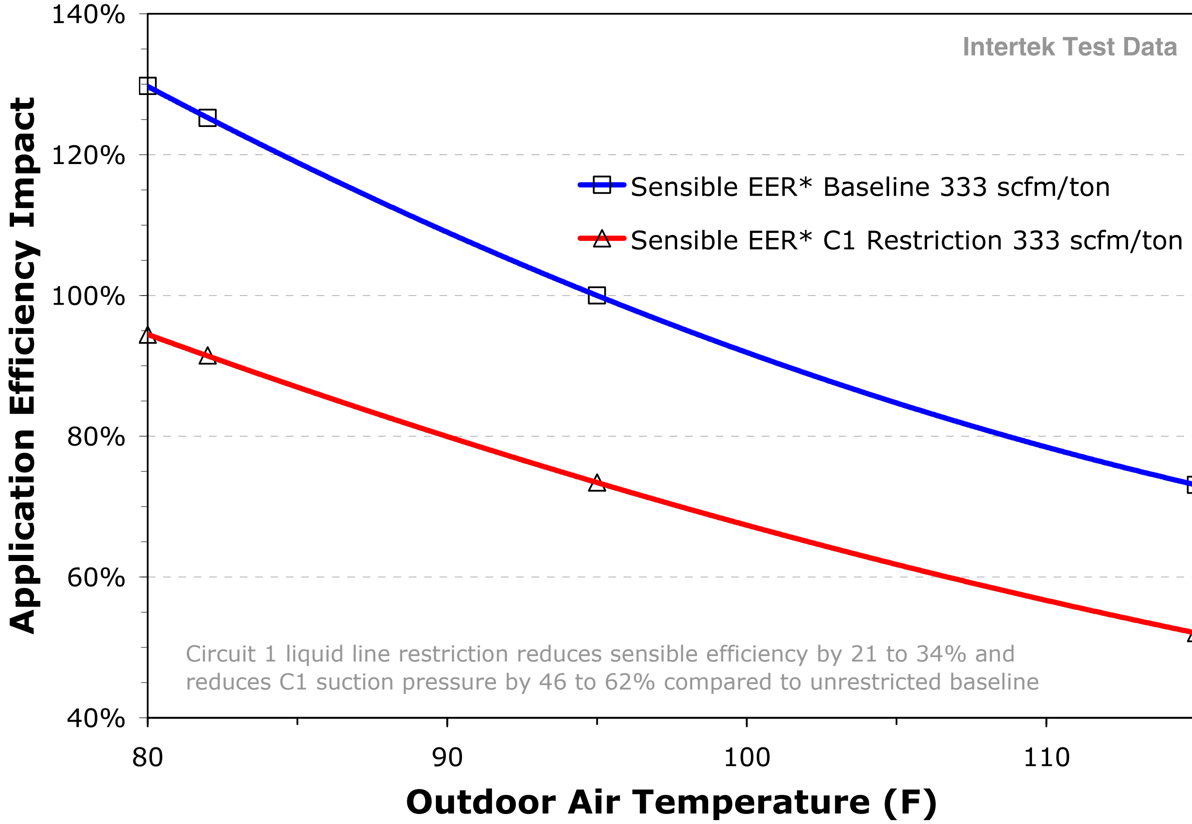


Table 42 and **Table 43** provide restriction test results at 115F and 82F OAT. With the circuit 1 restriction, EER*s decreased by 21 to 34%. For all tests the manufacturer and CEC refrigerant charge protocols misdiagnose C2 as undercharged. For all unrestricted tests the manufacturer and CEC protocols correctly diagnosed C1 as properly charged except for 267 scfm/ton at 82F OAT. For all restricted tests the manufacturer suction temperature protocols misdiagnosed C1 as undercharged. The CEC superheat protocols misdiagnosed C1 as undercharged for all tests except 267 scfm/ton at 95F OAT. Liquid line restrictions lower evaporator saturation temperature and increase superheat causing restricted circuits to be misdiagnosed as undercharged. The restricted circuit 1 evaporator saturation temperature (EST) is below freezing for all restricted tests. The CEC TS correctly diagnosed low capacity for all restriction tests. For restricted sensible capacities are less than the ACCA Manual N sensible cooling load and highlighted in red. The CEC ΔTS properly diagnoses 92% of tests with low sensible cooling capacity (i.e., ΔTS less than -3F) highlighted in yellow or red.

Figure 20: Application Efficiency Impacts versus Airflow with Circuit 1 Restriction Normalized to 400 scfm/ton for 7.5-ton non-TXV RTU3 with Economizer #4 and Damper Closed at 95F OAT

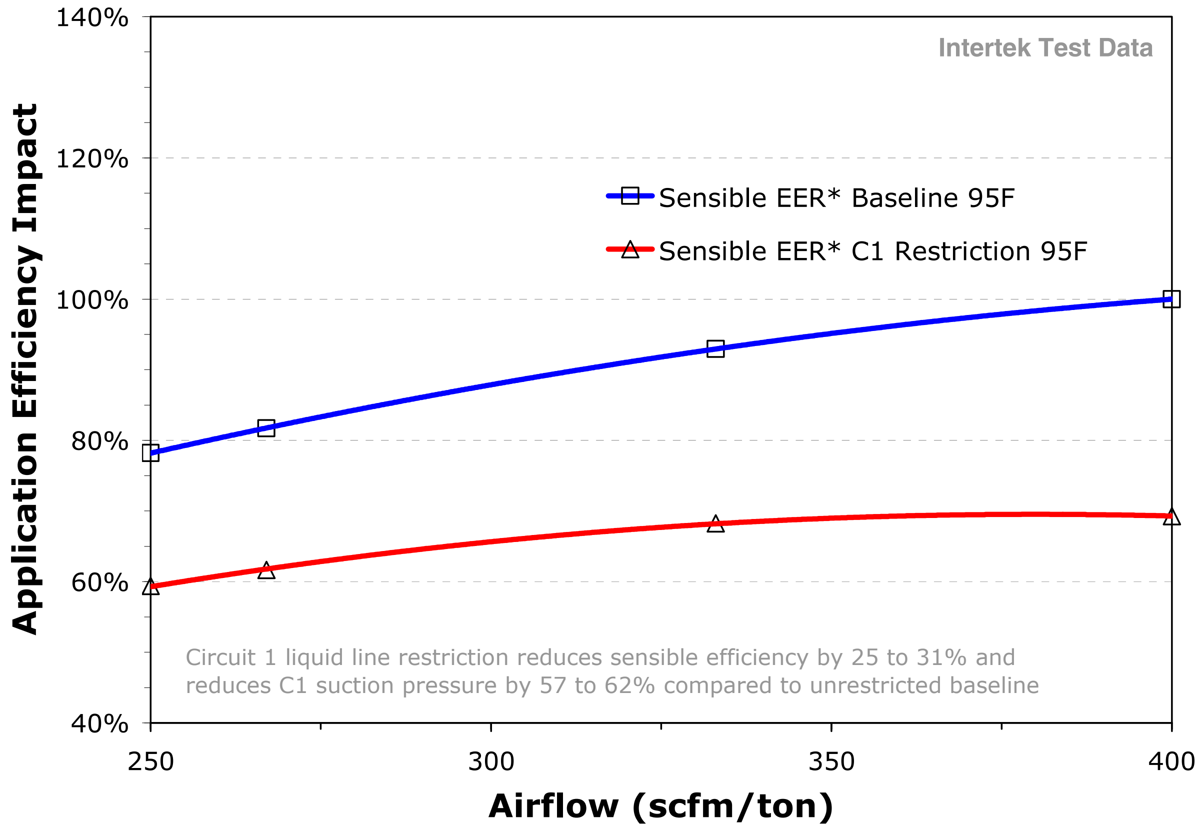


Table 42: Restriction Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 115F OAT (Circuit 1 Restriction)

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Filter Drier ΔT C1/C2	SP C1/C2	EST C1/C2	Airflow scfm/ ton	Sensible Cooling Capacity Btuh	Total Power W	Total EER*	Sensible EER*s
No Restriction											
3-42115CF	-5/40	0/35	-5.2	-2/-1	80/75	47/44	400	46,821	10,795	4.3	4.3
3-52115CF	-3/40	0/35	-4.0	-2/-1	78/74	46/44	333	42,568	10,560	4.2	4.0
3-62115CF	0/42	0/37	-4.0	-2/-1	76/75	45/44	267	34,079	10,351	3.4	3.3
C1 Restriction											
3-3000-115	63/41	61/36	-10.7	20/0	42/75	19/44	400	28,660	10,390	2.4	2.8
3-2500-115	63/41	61/35	-8.7	21/0	42/74	19/43	333	29,297	10,205	2.8	2.9
3-2000-115	62/40	60/32	-6.3	21/0	42/73	18/42	267	28,978	10,031	3.2	2.9

Table 43: Restriction Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed at 82F OAT (Circuit 1 Restriction)

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Filter Drier ΔT C1/C2	SP C1/C2	EST C1/C2	Airflow scfm/ ton	Sensible Cooling Capacity Btuh	Total Power W	Total EER*	Sensible EER*s
No Restriction											
3-4282CF	17/59	12/31	-2.2	-1/-1	70/56	41/31	400	57,345	7,897	8.8	7.3
3-5282CF	16/59	8/31	-0.5	-1/-1	69/56	40/31	333	53,180	7,702	8.7	6.9
3-6282CF	5/58	-7/29	1.8	-1/-1	66/56	38/31	267	47,548	7,534	8.3	6.3
C1 Restriction											
3-3000-82	52/60	57/31	-7.7	16/0	26/56	3/30	400	38,744	7,421	6.0	5.2
3-2500-82	46/60	52/31	-6.2	16/0	26/55	2/30	333	36,518	7,240	6.1	5.0
3-2000-82	25/59	14/29	-4.9	17/0	23/55	-2/30	267	32,237	7,084	5.7	4.6

Restrictions are generally caused by moisture, copper particles, flux/brazing residue, and particulates inside the system when installed, manufactured, or opened for repair. Oil in new refrigerant systems doesn't remain clean very long, especially in R410A systems with POE oils that have powerful solvent effects. Oil in the system quickly combines with moisture, acids, metal particles and other contaminants to produce sludge which plugs the filter drier or gets stuck on expansion devices. Contaminants causing restrictions can damage the compressor, clog metering devices, or make the metering device function improperly. Liquid line filter driers remove moisture, acid, and particulates (<10 microns) to prevent restrictions.

The manufacturer provides information for "troubleshooting" and diagnosing refrigerant restrictions from five other faults including refrigerant undercharge.[95] Refrigerant restrictions are identified as a possible cause of two problems: 1) head pressure too low, and 2) suction pressure too low. These problems are caused by the following faults: 1) dirty air filter, 2) insufficient evaporator airflow, 3) thermostat set to low, 4) compressor valves leaking, 5) restriction in liquid tube/filter drier/metering device, and 6) refrigerant undercharge. Technicians can easily check and correct dirty air filter, insufficient evaporator airflow, and thermostat set too low. If the system simultaneously has low discharge pressure and high suction pressure the cause is leaky compressor valves, worn compressor rings, or leaky oil separator.[96] These problems require compressor replacement. The remaining faults are refrigerant undercharge or restriction in liquid tube/ filter drier/metering device. Undercharge can be ruled out by the fact that C1 had suction pressure 2 times lower than C2 and C1 EST was below freezing and 4 times lower than C2 EST (EST ranged from -2 to 19F for C1 restriction). If the liquid line temperature 12 to 24 inches

[95] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

[96] Tomczyk. J, 2013. The Professor: Diagnosing Bad HVAC Compressor Valves When Low Head and High Suction Pressures Collide. AHRI News. http://www.achrnews.com/articles/124501-the-professor-diagnosing-bad-hvac-compressor-valves

upstream of the TXV entrance is 2 to 3F colder than ambient air, then there is a restriction upstream.[97] If the temperature drop across the filter drier is greater than 3F, then there is a filter drier restriction. Restriction tests found a 1 to 2F filter-drier inlet minus outlet temperature increase (ΔT is negative) while the C1 restriction caused a 16 to 21F temperature decrease. Clearly, C1 was restricted and not undercharged. The manufacturer's remedy is "recover refrigerant, remove restriction or replace filter drier, evacuate to 500 microns Hg hold for 20 minutes at or below 1000 microns Hg, and weigh in new refrigerant to factory charge."

4.1.12 Non-Condensable Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of non-condensables (NC) on the application efficiency (EER*) of RTU3 with economizer #4 installed, dampers closed, economizer perimeter unsealed, and airflow of 360 scfm/ton.[98] Tests were performed at outdoor conditions of 82F, 95F, and 115F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In order to emulate non-condensable air and water vapor faults, 0.4 ounces of nitrogen was added to circuit 1. The weight of nitrogen is normalized with respect to the factory charge (oz/oz) so 0.4 ounces of nitrogen represents 0.33% of the factory charge. Non-condensable test results are provided in **Table 44**. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 21** shows the NC fault energy efficiency impact versus OAT for the 7.5-ton non-TXV unit at 333 scfm/ton with economizer #4 and closed dampers. With 0.33% Nitrogen per factory charge by weight in circuit 1, the circuit 1 discharge pressure increased by 13 to 16%, total power increased by 5 to 6%, EER*s decreased by 14 to 18%, cooling capacity decreased by 10 to 14%, and total efficiency (EER*) decreased by 13 to 19%. For all tests the manufacturer and CEC protocols misdiagnosed circuit 2 as undercharged. For the 95 and 115F without NC, both protocols correctly diagnosed circuit 1 as properly charged and misdiagnosed C1 as undercharged at 82F OAT. For all NC tests the CEC ΔTS protocol diagnosed low capacity (highlighted in yellow) and the sensible cooling capacity was less than the ACCA Manual N sensible cooling load and highlighted in red. For the baseline tests at 82 and 95F, the CEC ΔTS protocol diagnosed proper airflow and the sensible cooling capacity was greater than 105% of ACCA Manual N sensible cooling load and highlighted in green.

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[97] Tomczyk. J. 2002. Diagnosing A Restricted Liquid Line Can Be Tricky. AHRI News. http://www.achrnews.com/articles/90784-diagnosing-a-restricted-liquid-line-can-be-tricky

[98] If proper vacuum is not achieved at installation or after being opened for repair, the refrigerant system will be contaminated with non-condensable air and water vapor which can mix with refrigerant oils causing sludge, which can lead to compressor failure. Non-condensables (NC) decrease condenser heat transfer and cooling capacity and increase condenser pressure and compressor power input.

Table 44: Non-Condensable Nitrogen Impacts versus OAT for 7.5-ton non-TXV RTU3 with Factory Charge (Circuit 1 0.33% Non-Condensable Nitrogen)

Test	Mfr Protocol C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Discharge Pressure C1/C2	Suction Pressure C1/C2	Sensible Cooling Capacity Btuh	Total Power W	Total EER*	Sensible EER*s
Baseline No NC									
3-5282CF	16/59	8/31	-0.5	225/209	69/56	53,180	7,702	8.7	6.9
3-5295CF	1/54	-3/34	-2.1	268/254	73/64	48,128	8,727	6.3	5.5
3-52115CF	-3/40	0/35	-4.0	341/331	78/74	42,568	10,560	4.2	4.0
C1 0.33% NC									
3-NC-2500-82	52/54	29/32	-3.1	260/205	56/55	45,784	8,092	7.1	5.7
3-NC-2500-95	51/58	31/36	-4.3	305/249	63/62	41,965	9,153	5.3	4.6
3-NC-2500-115	28/44	27/37	-5.5	386/327	75/73	38,521	11,153	3.6	3.5

Figure 21: Application Efficiency Impacts versus OAT for 7.5-ton non-TXV RTU3 with Factory Charge and Circuit 1 0.33% Non-Condensable Nitrogen

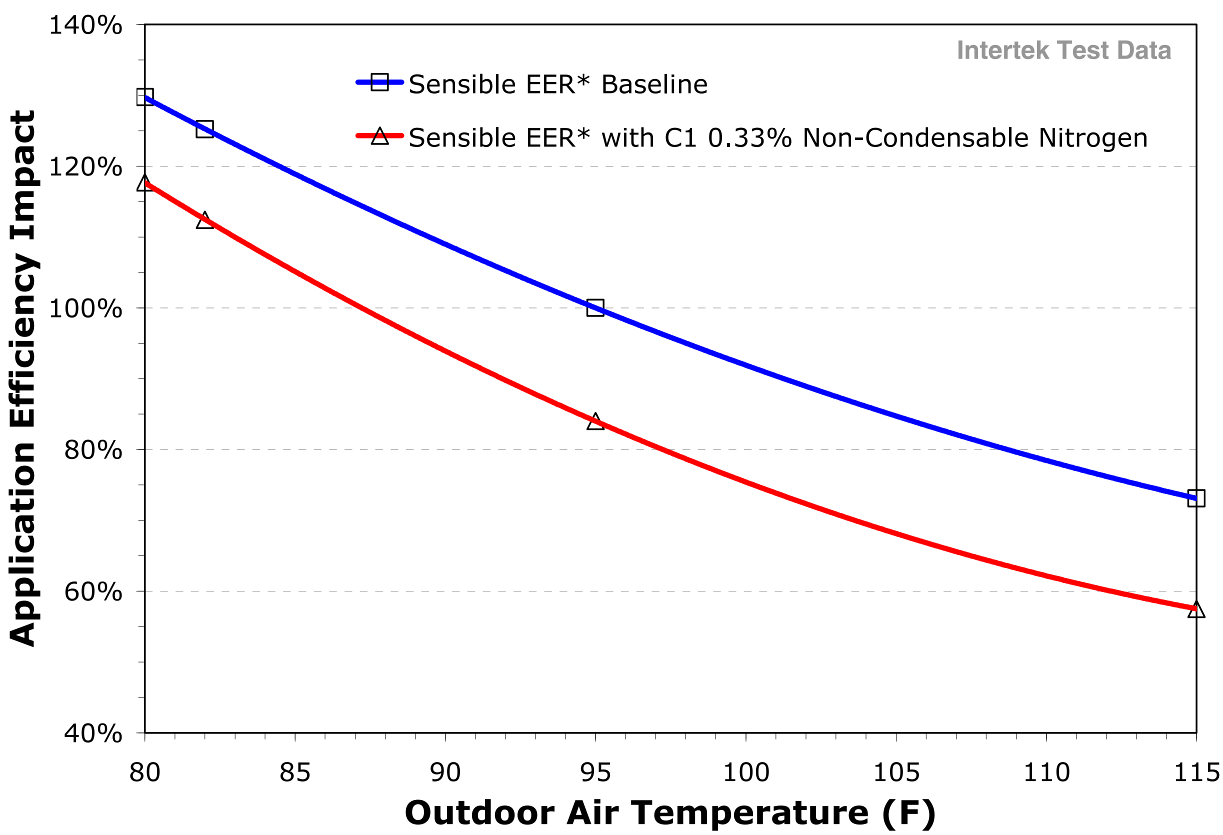
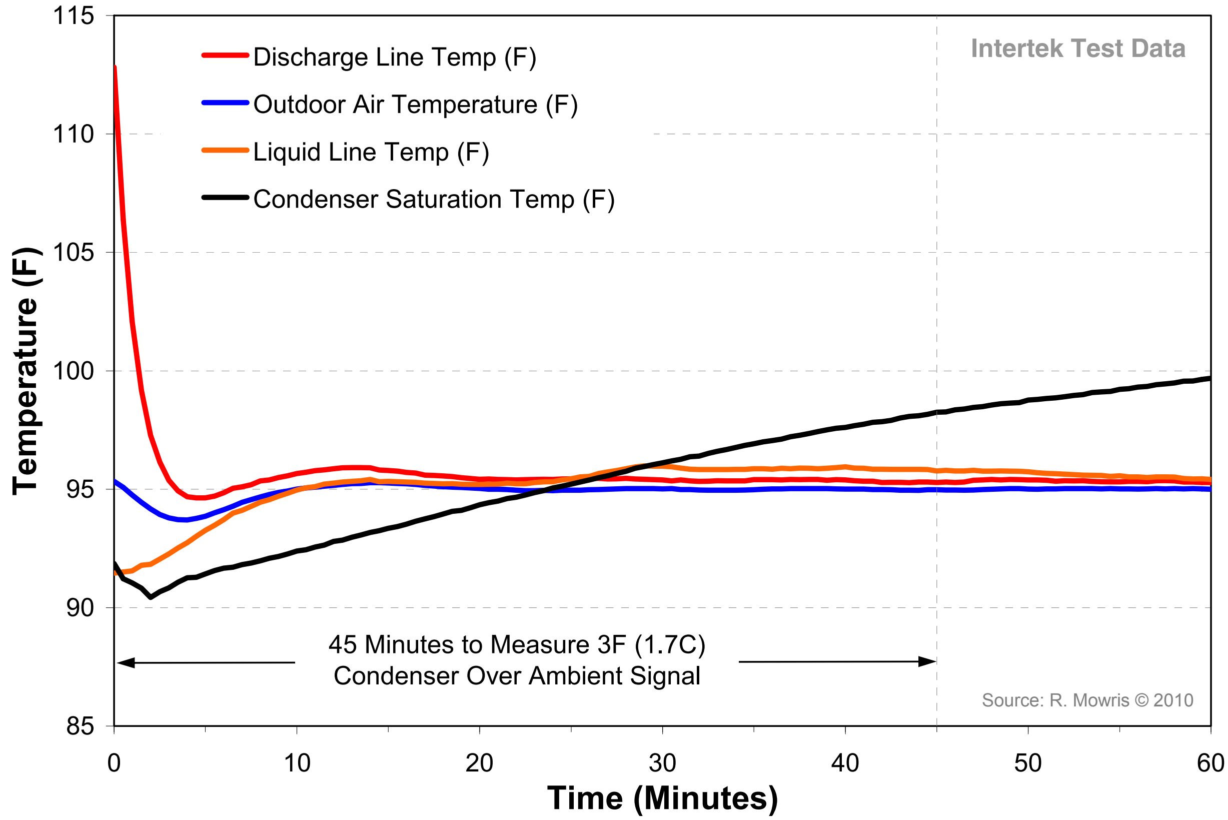


Figure 22 shows the laboratory test of the time required to check for non-condensable nitrogen in a 3-ton split-system air conditioner with 101 ounces of factory charge and 1 ounce of non-condensable nitrogen (1% by weight of factory charge).[99] Time to check for non-condensable Nitrogen was 45 minutes for the condenser pressure to reach 190.6 psig and saturation temperature to reach 98.2F or 3.2F above 95F condenser entering OAT. It took 10 to 25 minutes for discharge and liquid temperatures to reach equilibrium with the 95F condenser entering OAT plus 20 additional minutes for non-condensable nitrogen to coalesce in the condenser from being more dispersed throughout the system after the compressor and evaporator fan were turned off.

Figure 22: Laboratory Test of Time to Check for 1% Non-Condensable Nitrogen in a 3-ton non-TXV R22 Split-System with Condenser Fan Operating and Without Compressor or Evaporator Operating



Non-condensables are caused by not achieving a proper vacuum at installation causing the refrigerant system to be contaminated with non-condensable air and water vapor which combine

[99] Mowris, R. Jones, E. 2010. Mowris, R., Eshom, R., Jones, E. 2015. Laboratory Measurements and Diagnostics of Residential HVAC Installation and Maintenance Faults. 8th International Conference – EEDAL'15. http://iet.jrc.ec.europa.eu/energyefficiency/sites/energyefficiency/files/events/EEDAL15/21_Test-Methods/eedal15_submission_171.pdf

with refrigerant and oil to form acid and sludge leading to compressor failure. Non-condensables decrease condenser heat transfer and cooling capacity and increase condenser pressure and power input.

Refrigerant charge protocols misdiagnose non-condensables as undercharge because they do not include discharge pressure or troubleshooting faults identified by the manufacturer to evaluate condenser heat transfer faults (i.e., non-condensables or condenser blockage). The manufacturer provides information for "troubleshooting" and diagnosing non-condensables from nine other faults including refrigerant over and undercharge.[100] Non-condensable air in the system is identified as a possible cause for two problems: 1) compressor operates continuously, and 2) excessive head pressure. The manufacturer lists the following causes for these two problems: 1) dirty air filter, 2) blocked condenser, 3) low capacity or too much outdoor air, 4) thermostat set too low, 5) refrigerant undercharge, 6) leaking valves, 7) non-condensable air in system, 8) refrigerant overcharge, and 9) condenser air restricted or air short-cycling. Technicians can easily check and correct dirty air filter, blocked condenser, thermostat set too low, or condenser air restriction. They can also easily check and correct low cooling capacity or too much outdoor air by making sure the outdoor air damper is at the minimum position or closed and temporarily sealed for troubleshooting. If the system simultaneously has low discharge pressure and high suction pressure the cause is leaky compressor valves, worn compressor rings, or leaky oil separator.[101] These problems require compressor replacement. The remaining faults are undercharge, overcharge, or non-condensables. Overcharge can be ruled out by the fact that both circuits had similar suction pressure and C1 has 18 to 27% higher discharge pressure than C2. This would not be the case with an undercharge or overcharge since both discharge and suction pressures would both need to be high or low. For non-condensables the C1 discharge pressure is much higher than C2 and C1/C2 suction pressure are approximately the same. The following procedure to check for non-condensables is from AHRI News.[102] 1) Electrically disable the compressor to operate the condenser fan. 2) Attach a temperature probe to both the discharge and liquid line. 3) Place a third temperature probe to measure condenser entering air temperature. 4) Connect pressure gauge to discharge valve to measure refrigerant pressure in the condenser. 5) When all three probes are approximately the same temperature [and the fan has been operating for at least 45 minutes], record the pressure of the refrigerant in the condenser.[103] 6) Convert the

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[100] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

[101] Tomczyk. J. 2013. The Professor: Diagnosing Bad HVAC Compressor Valves When Low Head and High Suction Pressures Collide. AHRI News. http://www.achrnews.com/articles/124501-the-professor-diagnosing-bad-hvac-compressor-valves

[102] Marchese. J. 2007. Checking For Non-condensables. AHRI News. http://www.achrnews.com/articles/102428-checking-for-noncondensables

[103] Intertek laboratory test of total time required to check for non-condensable nitrogen was 45 minutes for the condenser pressure to reach 190.6 psig and saturation temperature to reach 98.2F or 3.2F above the 95F condenser entering OAT. It took 10 to 25 minutes for discharge and liquid temperatures to reach equilibrium with the 95F condenser entering OAT plus 20 additional minutes for non-condensable nitrogen to migrate back to the condenser after being dispersed within the R22 refrigerant after the compressor and evaporator fan were turned off.

measured condenser pressure to its saturation temperature (using refrigerant pressure-temperature chart). 7) If the condenser saturation temperature is greater than the discharge, liquid, and OAT entering condenser by more than 3F, then there are non-condensables in the system that need to be removed.[104] The manufacturer's remedy is "recover refrigerant, replace filter drier, evacuate to 500 microns Hg hold for 20 minutes at or below 1000 microns Hg, and weigh in new refrigerant to factory charge."

4.1.13 Multiple Fault Tests for 7.5-ton non-TXV RTU3

Laboratory tests were performed to evaluate the impact of multiple faults on the application efficiency (EER*) of RTU3 with economizer #4 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. Tests were performed with 80 to 120% refrigerant charge per factory charge, 0 to 20% blocked evaporator, and 0 to 30% blocked condenser. The predicted application efficiency ratios for multiple faults are calculated using **Equation 14**.

Equation 14 $$EER^*_{s_p} = EER^*_{s_b}\left[1 - \sum_{i=1}^{n} \varepsilon_i\right]$$

Where,

$EER^*_{s_p}$ = predicted application EER*s impact for multiple faults (Btuh/W)

$EER^*_{s_b}$ = measured baseline non-fault EER*s (Btuh/W)

n = number (n) of multiple faults

$\varepsilon_i = 1 - \left[\dfrac{EER^*_{s_i}}{EER^*_{s_o}}\right]$ = single-fault to non-fault EER*s impact ratio (dimensionless)

$EER^*_{s_i}$ = single-fault EER*s (Btuh/W)

$EER^*_{s_o}$ = single-fault baseline EER*s (Btuh/W)

The difference of measured minus predicted divided by measured EER*s is $\Delta\varepsilon$ or delta epsilon calculated using **Equation 15**.

[104] Laboratory test of 1% non-condensable indicated 45 minutes of time required for condenser saturation temperature to reach 98.2F or 3.2F above the 95F condenser entering OAT.

Equation 15 $$\Delta\varepsilon = \frac{EER^*_{s_m} - EER^*_{s_p}}{EER^*_{s_m}}$$

Where,

$\Delta\varepsilon$ = difference of measured minus predicted divided by measured EER*s (dimensionless)

$EER^*_{s_m}$ = measured EER*s (Btuh/W)

The predicted versus measured impacts for multiple faults are shown in **Table 45** for refrigerant charge (80 to 120%), evaporator coil blockage (0 to 20%), and condenser coil blockage (0 to 30%). Predicted impacts for multiple faults are calculated using **Equation 15** based on measured single-fault impacts. The average difference is 0.3% indicating predicted multiple fault impacts based on summing individual impacts are slightly greater than measured impacts. This is an important finding since HVAC maintenance involves multiple repairs and ex ante savings are typically summed for each repair.

Table 45: Measured versus Predicted Multiple Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed (16.1% OAF), 80-120% Factory Charge, 0-20% Blocked Evaporator, 0-30% Blocked Condenser

Test	Charge Impact ε_i	Evap Block Impact ε_i	Cond Block Impact ε_i	Predicted Sensible EER*s	Measured Sensible EER*s	Difference Measured vs Predicted $\Delta\varepsilon$ %
C8-R100-95-CE-1	0.000	0.000	0.000	5.16	5.16	0.0%
C8-CB30-95-CE	0.000	0.000	0.079	4.69	4.69	0.0%
C8-EB20-95-CE	0.000	0.022	0.000	4.98	4.98	0.0%
C8-R80-E20C0-95CE	0.148	0.022	0.000	4.29	4.20	-2.1%
C8-R80-E20C30-95CE	0.148	0.022	0.079	3.88	3.98	2.5%
C8-R80-E0C30-95CE	0.148	0.000	0.079	3.99	4.07	1.8%
C8-R100-E20C30-95CE	0.000	0.022	0.079	4.64	4.65	0.2%
C8-R120-E20C0-95CE	-0.018	0.022	0.000	5.14	5.12	-0.4%
C8-R120-E20C30-95CE	-0.018	0.022	0.079	4.73	4.74	0.1%
C8-R120-E0C30-95CE	-0.018	0.000	0.079	4.85	4.90	1.1%
Average						**0.3%**

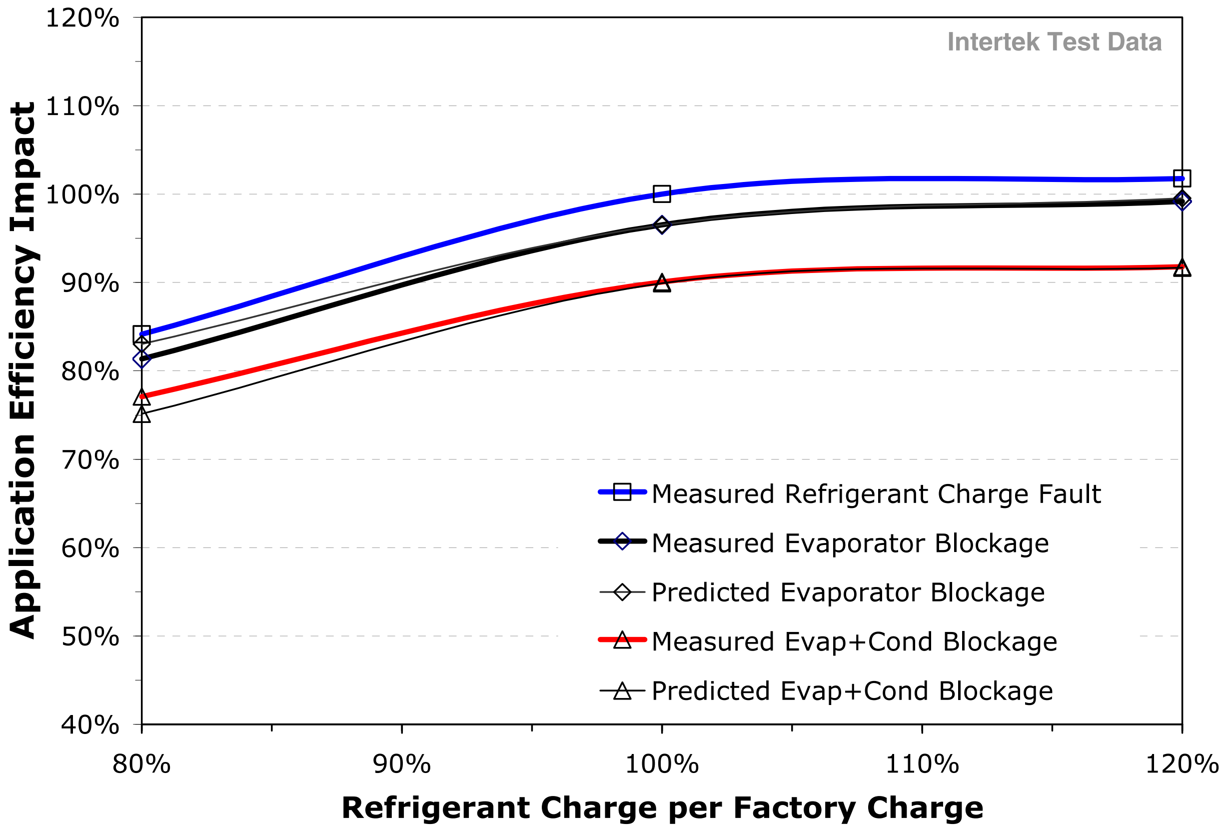
Multiple fault test results are provided in **Table 46**. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. For the three tests with 80% factory charge, the manufacturer and CEC refrigerant charge protocols correctly diagnosed undercharge. For four tests with 100% factory charge both protocols misdiagnosed undercharge. For three tests with 120% factory charge the manufacturer and CEC protocols misdiagnosed C1 as undercharged and the CEC protocol misdiagnosed C2 as properly charged. For 120% charge, the manufacturer protocol misdiagnosed C2 as properly charged except for 120% charge plus 30% condenser blockage which was correctly diagnosed as overcharged. The CEC ΔTS protocol was less than -3F for seven tests with 80 to 100% factory charge indicating low cooling capacity and highlighted in yellow. For three tests with 120% factory charge the CEC TS protocol diagnosed proper airflow and cooling capacity. The sensible cooling capacities for the seven tests with 80 to 100% factory charge are less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red. All other multiple-fault tests have sensible cooling capacities greater than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in yellow or green.

Table 46: Multiple Fault Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed, 80-120% Factory Charge, 0-20% Blocked Evaporator, 0-30% Blocked Condenser

Test	C1/C2 Charge %	Blocked Evap Coil %	Blocked Cond Coil %	Mfr C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C8-R80-E20C0-95CE	80/80	20		79/52	53/31	-6.5	4.47	34,196	4.20
C8-R80-E20C30-95CE	80/80	20	30	79/42	52/27	-6.3	4.22	34,637	3.98
C8-R80-E0C30-95CE	80/80		30	82/44	54/31	-6.5	4.19	35,707	4.07
C8-R100-95-CE-1	100/100			67/33	40/25	-3.4	5.82	44,673	5.16
C8-CB30-95-CE	100/100		30	40/25	31/23	-3.3	4.97	44,861	4.69
C8-EB20-95-CE	100/100	20		53/37	33/25	-3.2	5.67	43,381	4.98
C8-R100-E20C30-95CE	100/100	20	30	60/20	36/16	-3.1	5.34	43,218	4.65
C8-R120-E20C0-95CE	120/120	20		53/2	32/-3	-2.0	6.05	45,779	5.12
C8-R120-E20C30-95CE	120/120	20	30	39/-3	26/-4	-1.8	5.47	46,024	4.74
C8-R120-E0C30-95CE	120/120		30	40/-7	29/-4	-1.8	5.52	48,264	4.90

Figure 23 shows the measured and predicted application efficiency impacts versus refrigerant charge per factory charge with evaporator coil blockage and evaporator plus condenser coil blockage. Application sensible efficiency is maximized with 120% of factory charge. Efficiency decreased by 2 to 8% with 20% blocked evaporator or 20% blocked evaporator plus 30% blocked condenser coil. The predicted impacts (gray lines) for evaporator coil blockage and evaporator plus condenser coil blockage are 0.3% greater than measured impacts.

Figure 23: Measured and Predicted Multiple Fault Application Efficiency Impacts for 7.5-ton non-TXV RTU3 with Economizer #4 Damper Closed, 80-120% Factory Charge, 0-20% Blocked Evaporator, and 0-30% Blocked Condenser



Most HVAC systems have multiple faults. Troubleshooting multiple faults using manufacturer procedures will reduce or eliminate "false alarms," misdetection, and misdiagnosis. These examples indicate multiple faults such as undercharge or overcharge plus evaporator and condenser blockage cause FDD protocols to misdiagnose undercharge. Technicians need to visually diagnose and correct evaporator and condenser coil blockage and check overventilation and other more complicated faults before performing refrigerant charge FDD.

4.2 Test Results for 7.5-ton TXV Packaged HVAC RTU1 and RTU2

Two identical 7.5-ton TXV packaged HVAC models (RTU1 and RTU2) made by the same manufacturer were tested in the laboratory per the ANSI/AHRI 340/360 test procedure. Each model uses R22 refrigerant and they are identical except for evaporator blower motors. RTU1 was delivered with a 3 horsepower (hp) fan motor with 1047 rpm fan speed, 5.75-inch diameter pulley, and motor sheave at 3 turns open. RTU2 was delivered with a 2-hp fan motor with 924 rpm fan speed, 5.75-inch diameter pulley, and motor sheave at 3 turns open.[105] Each unit was shipped with the same forward-curved centrifugal blower wheel with 1” wide blades and 15” diameter x 15” widths. Each 7.5-ton TXV unit has two compressors, each compressor has a separate refrigerant circuit, and each circuit is equipped with one thermostatic expansion valve (TXV) on the liquid line at the evaporator inlet. Each unit was tested in the horizontal configuration.

4.2.1 Out-of-Box Tests for 7.5-ton TXV Units

The 7.5-ton TXV RTU1 and RTU2 were tested in the “out-of-the-box” as-purchased condition. **Table 47** provides out-of-box tests for RTU1 and RTU2. The EER* tests were 6 to 24% less than the rated 11 EER. With the factory fan speed and 0.25 IWC ESP, the measured airflow was between 544 and 678 scfm/ton. This was 21 to 51% greater than the 450 scfm/ton maximum airflow specified in ANSI/AHRI 340/360 and 55 to 94% greater than the AHRI-rated 350 scfm/ton airflow.[106] The out-of-box fan power was 2 to 3 times greater than the AHRI verification test (see **Table 49**).[107] A 2-hp fan motor was installed in RTU1 to reduce fan power and airflow and increase EER* from 8.39 to 9.53.

Table 47: Out-of-Box Tests for 7.5-ton TXV RTU1 and RTU2 at Factory Fan Speed without Economizer

Test	C1/C2 Charge %	Fan RPM	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER	Tested EER*	ΔEER
T2-ONE-3HP (RTU1)	100/100	1047	0.25	678	3070	10573	88,716	11.0	8.39	-24%
T2-ONE (RTU1 2-hp fan)	100/100	915	0.25	570	1970	9400	89,614	11.0	9.53	-13%
3T-ONE (RTU2 2-hp fan)	100/75	924	0.25	544	1916	9093	93,699	11.0	10.3	-6%

Table 48 provides EER and SEER cycling test data for the two-compressor 7.5-ton TXV RTU2. The RTU2 fan speed was reduced by turning the motor sheave to 6 turns to achieve 399 scfm/ton and reduce fan power by 47% which produced less fan heat and more cooling capacity. The manufacturer is not required to publish cycling test data per ANSI/AHRI Standard 210/240. The ANSI/AHRI 210/240 standard only applies to air-conditioning equipment with cooling capacities rated below 65,000 Btu/hour at ARI Standard Rating Conditions. Cycling tests were performed

105 RTU2 circuit 2 refrigerant charge was subsequently determined to be 25% undercharged.

106 Per ANSI/AHRI STANDARD 340/360-2007, 6.1.3.2 Indoor Coil Airflow Rate, “Equipment with indoor fans intended for use with field installed duct systems shall be rated at the indoor-coil airflow rate (not to exceed 37.5 scfm per 1000 Btu/h [0.06 m3/s per 1000 W] of rated capacity) delivered when operating against the minimum external resistance specified in Table 5 or at a lower indoor-coil airflow rate if so specified by the manufacturer.”

107 Fan power is 1720 to 1740 W versus 850 W for AHRI verification test.

at the request of the DEER DMQC team to evaluate part-load analysis for building energy simulations. For RTU2, the average SEER was 11.77 based on three SEER tests and the average cyclic degradation coefficient (C_d) was 0.215 based on tests 3T-22C Cyclic #2, #3, and #4.[108] The C_d test results are 2 to 3 times greater than previously assumed in DEER models.

Table 48: AHRI EER and SEER Cycling Tests for 7.5-ton TXV RTU2 with Both Compressors and without Economizer

Test	Fan HP	Fan Turn	Fan RPM	ESP IWC	Airflow scfm/ton	Rated EER	Rated SEER	Tested EER	Tested SEER	C_d
3T-22AA-0	2	6	776	0.28	399	11.0		10.91		
3T-22B	2	6	762	0.26	399	13.5		13.19		
3T-22C Cyclic #2	2	6							11.9	0.195
3T-22C Cyclic #3	2	6							11.75	0.219
3T-22C Cyclic #4	2	6							11.67	0.230
Average									**11.77**	**0.215**

4.2.2 AHRI Verification Tests for 7.5-ton TXV Units

The airflow for each unit was reduced to meet the nominal AHRI rating by turning the motor sheave out to 6 turns to reduce fan speed to approximately 750 RPM with the 2-hp blower motor. The cabinet panel joints on each RTU were sealed with tape to reduce outdoor air leakage for AHRI tests. **Table 49** provides the AHRI verification tests for RTU1 and RTU2 with 2-hp blower motors and lowest fan speed available with motor sheave at 6 turns.[109] At the lowest fan speed, the EER of each unit is within 95% of the AHRI rating. The EER for RTU1 was 10.56 and within -4% of the 11.0 EER rating in the horizontal configuration. Two EER measurements were performed for RTU2. The first measurement for RTU2 was 11.01 EER which is within 0.1% of the AHRI rating. After the first RTU2 test, Intertek technicians accurately weighed the refrigerant contained in each circuit by recovering the refrigerant into reclaim tanks. The out-of-box unit had 6.44 lbs in C1 and 4.68 lbs in C2. The required factory charge is 6.4 lbs in C1 and 6.2 lbs in C2. Therefore, C2 was undercharged by 25%. Intertek evacuated each circuit to industry-standard 500 microns vacuum and accurately weighed factory charge into each circuit. The second measurement of EER was 10.92 for RTU2 with factory charge which is within -0.7% of the AHRI rating. These tests indicate that 25% low refrigerant charge in C2 had less than 1% impact on the steady-state efficiency of RTU2 under AHRI test conditions and without an economizer installed. **Table 49** shows that the AHRI-ratings for these units are only achievable at the lowest fan speed and 0.25 IWC ESP which is not typical of field conditions. Very few

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108 Cycling degradation (CD) coefficient measures the efficiency loss due to cycling of units as determined in Appendices C and D of ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment Standard 210/240. Air-Conditioning Heating and Refrigeration Institute.

109 The motor sheave can be set from 0 turns (highest fan speed) to 6 turns (lowest fan speed). The factory fan speed is 3 turns.

applications would have such low total static pressure and few packaged rooftop units will be setup with the lowest possible fan speed. In addition, all units installed in California with cooling capacities greater than 75,000 Btuh are required by law to have an economizer installed.[110] Subsequent tests with typical external standard pressure (0.92 IWC) and no economizer found application efficiency was 20% less than the rated EER (i.e., 8.8 EER* versus 11 EER). With an economizer installed and dampers closed the application efficiency was 31% less than the rated EER (7.6 EER* for test T2-TRN-95-CE-DM versus 11 EER rating).

Table 49: AHRI Verification Tests for 7.5-ton TXV Units with Lowest Fan Speed without Economizer

Test (Unit)	C1/C2 Charge %	ESP IWC	Airflow scfm/ton	Fan Power W	Total Power W	Total Cooling Capacity Btuh	Rated EER	Tested EER	ΔEER	Tested Sensible EER
3T-22A-Retest-OB (RTU2)	100/75	0.25	378	1,020	8,102	89,187	11.0	11.01	0.1%	8.14
3T-22A-Retest (RTU2)	100/100	0.25	381	1,010	8,211	89,655	11.0	10.92	-0.7%	8.10
T2-22A-ONE-6T-US (RTU1)	100/100	0.25	416	745	1,100	89,961	11.0	10.56	-4%	8.04

The 7.5-ton TXV RTUs have IPLV AHRI ratings of 11.5. **Table 50** provides the measured IPLV which was 11.92 for RTU2 and 11.56 for RTU1. Measured IPLV values were 0.5 to 4% greater than rated values. The IEER for RTU2 was 11.91 and the IEER for RTU1 was 11.25. IEER ratings are not available from the manufacturer for these vintage R22 models.

Table 50: AHRI IEER and IPLV Verification Tests for 7.5-ton TXV Units without Economizer

Test	Fan HP	Fan Turn	Fan RPM	Tested IEER	Test	Tested IPLV
IEER Calculation 7.5 ton RTU2 6 turns.xls	2	6	746	11.91	Retest IPLV Calculation 7.5 ton RTU2 6 turns.xls	11.92
IEER Calculation 7.5 ton RTU1 6 turns.xls	2	6	744	11.25	IPLV Calculation 7.5 ton RTU1 6 turns.xls	11.56

[110] CEC 2004. 2005 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. Effective Date October 1, 2005. P400-03-001F-M. Economizers were required for each individual cooling fan system that has a design supply capacity over 2,500 cfm and a total mechanical cooling capacity over 75,000 Btu/hr shall include either an air economizer capable of modulating outside-air and return-air dampers to supply 100% of the design supply air quantity as outside-air; or a water economizer capable of providing 100% of the expected system cooling load as calculated in accordance with a method approved by the commission, at outside air temperatures of 50F dry-bulb/45F.

4.2.3 Manufacturer Refrigerant Charge Diagnostics for 7.5-ton TXV Units

The circuit-specific manufacturer refrigerant charge diagnostic protocols for the 7.5-ton TXV units are based on discharge pressure (DP), suction pressure (SP), and superheat (SH) as a function of outdoor and return (i.e., evaporator coil entering) drybulb/wetbulb (DB/WB) temperatures.[111] The manufacturer refrigerant charge tolerances are +/-10 psig for DP, +/-5 psig for SP and +/-5F for SH.[112] The manufacturer refrigerant charge tolerances are +/-10 psig for DP, +/-5 psig for SP and +/-5F for SH.[113] The manufacturer does not provide superheat target values, airflow diagnostic protocols, or liquid pressure ports so subcooling cannot be evaluated. The manufacturer service instructions do not mention closing and sealing economizer dampers to reduce excess outdoor airflow prior to evaluating refrigerant charge diagnostics. The CEC ΔTS protocol is used to evaluate airflow and sensible cooling capacity based on test results for the 7.5-ton TXV units.[114] For information about the CEC protocols see **Section 2.1.3**. The laboratory tests provide sensible cooling minus ventilation loads which are included in the measurements. Sensible cooling capacity test results are diagnosed using ACCA Manual N sensible cooling load design values minus ventilation loads provided in **Table 2** and described in **Section 2.2.1**.

Table 51 shows manufacturer refrigerant charge and CEC temperature split diagnostics for RTU2 and RTU1. The manufacturer ΔDP, ΔSP, and ΔSH protocols correctly diagnosed proper charge all tests except C2 with 75% factory charge (3T-22A-retest-OB) highlighted in yellow, but EER and sensible cooling capacity are within 95% of the published AHRI rating. The CEC ΔTS protocol correctly diagnosed proper airflow and sensible capacity for all tests highlighted in green. The sensible cooling capacities for all tests are greater than 105% of 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green.

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[111] Trane Service Facts THC092-SF-1, Packaged Electric/Electric 7 1/2 Ton Dual Compressor Rooftop Units, THC092-SF-1.pdf. www.comfortsite.com, 2701 Wilma Rudolph Blvd., Clarksville, TN 37040. Manufacturer protocols are “based on indoor airflow of 400 cfm/ton.”

[112] Manufacturer refrigerant charge protocol for undercharge: ΔDP < -10 psig, ΔSP <-5 psig, ΔSH >+5F. Manufacturer refrigerant charge protocol for overcharge: ΔDP > 10 psig, ΔSP > 5 psig, ΔSH <-5F. Manufacturer protocol for correct charge: -10 psig ≤ ΔDP ≤ 10 psig, -5 psig ≤ ΔSP ≤ 5 psig, -5F ≤ ΔSH ≤ 5F.

[113] Manufacturer refrigerant charge protocol for undercharge: ΔDP < -10 psig, ΔSP <-5 psig, ΔSH >+5F. Manufacturer refrigerant charge protocol for overcharge: ΔDP > 10 psig, ΔSP > 5 psig, ΔSH <-5F. Manufacturer protocol for correct charge: -10 psig ≤ ΔDP ≤ 10 psig, -5 psig ≤ ΔSP ≤ 5 psig, -5F ≤ ΔSH ≤ 5F.

[114] California Energy Commission (CEC). 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. RA3.2 Procedures for Determining Refrigerant Charge for Split System Space Cooling Systems Without a Charge Indicator Display. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

Table 51: Manufacturer Refrigerant Charge Diagnostics for 7.5-ton TXV RTUs without Economizer at 95F OAT

Test (Unit)	C1/C2 Charge %	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Fan Power W	Total Power W	Total EER*	Sensible Cooling Capacity kBtuh	Sensible EER*s
3T-22A-Retest (RTU2)	100/100	-1/3	0/-2	-1/-4	1.3	381	1,010	8,211	10.92	66,480	8.1
3T-22A-Retest-OB (RTU2)	100/75	-4/-3	0/-2	1/-3	1.3	378	1,020	8,102	11.01	65,989	8.1
T2-22A-ONE-6T-US (RTU1)	100/100	1/5	2/-3	4/1	0.2	416	1,100	8,516	10.56	68,462	8.0

Table 52 shows diagnostic test results for three tests with economizer installed and damper closed: 1) test 3T-75629575-E6 and 330 scfm/ton (756 rpm) with 75% charge in C2, 2) test 3T-75629575-E6-Retest and 301 (474 rpm) scfm/ton with 100% charge in C1/C2, and 3) test 3T-75629575-E3-Retest and 378 scfm/ton (928 rpm) with 100% charge in C1/C2. For the test with 75% charge in C2, the total EER* was 7.7 or 8.3% less than 8.4 EER* with correct charge, but sensible EER was the same. The application efficiency was 23 to 30% less than the AHRI rating due to the economizer and typical static pressure. The manufacturer protocols misdiagnosed C2 as correctly charged when it was 25% undercharged and 8.3% less efficient than 100% charge. For test 3T-75629575-E6-Retest, the manufacturer misdiagnosed C1 as undercharged and for 3T-75629575-E3-Retest, C1 was misdiagnosed as undercharged. Test 3T-75629575-E6-Retest was performed with motor sheave at 6 turns (474 rpm) providing 301 scfm/ton or 15% less than rated airflow. Test 3T-75629575-E3-Retest was performed with motor sheave at 3 turns (928 rpm) providing 8% more airflow. The CEC protocol ΔTS protocol misdiagnosed proper airflow for test 3T-75629575-E6-Retest when the airflow was 15% less than the rated 350 scfm/ton airflow. The CEC TS protocol correctly diagnosed the other tests with proper airflow and cooling capacity. The sensible cooling capacities for all tests are greater than 105% of 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 52: Manufacturer Refrigerant Charge Diagnostics for 7.5-ton TXV RTU2 with Economizer #1 and Closed Damper at 95F Outdoor Temperature

Test	C1/C2 Refrig Chg %	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	ISP IWC	ESP IWC	Airflow scfm/ ton	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
3T-75629575-E6	100/75	-5/-3	-3/-3	5/3	0.2	-0.27	0.37	330	8,008	7.7	54,180	6.8
3T-75629575-E6-Retest	100/100	-7/0	-6/-5	6/3	2.0	-0.29	0.56	301	7,953	8.4	54,188	6.8
3T-75629575-E3-Retest	100/100	-4/3	-4/-1	7/4	-0.7	-0.47	0.92	378	8,610	7.7	59,126	6.9

4.2.4 Economizer Outdoor Airflow Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate economizer outdoor airflow, overventilation, and untended outdoor air leakage on the 7.5-ton TXV RTU2 with three economizers installed. Three economizers were tested on RTU2 to measure OAF. Economizer #0 was purchased with an original equipment manufacturer (OEM) analog controller/actuator and was only able to open dampers slightly beyond 2-fingers. Economizer #0 was retrofitted with a digital controller/actuator (referred to as economizer #1) to test from closed to fully open positions. Economizer #2 was purchased from another manufacturer to test another unit with digital controller/actuator. Tests were performed with the fan motor on, both compressors off, 55F OAT and indoor conditions of 75F DB and 62F WB. The outdoor, return, and mixed-air drybulb temperatures were measured using resistance temperature detector (RTD) sensors in the outdoor, return, and supply-air samplers. The OAT entering the economizer was also measured using an array of 6 thermocouple sensors installed in the economizer inlet. The volumetric flow rate of air was measured using the Code Tester.[115] For tests with blower fan and compressors operating, the mixed-air temperature entering the evaporator was measured with an array of 22 shielded-drybulb temperature sensors located on the air filter inlet adjacent to the evaporator.

Table 53 provides calculated economizer #1 outdoor air fractions using **Equation 6** and **Equation 9** without compressors operating and at 95F using **Equation 1** with compressors operating for the 7.5-ton RTU2 at 55F OAT and approximately 400 scfm/ton (equations are described in **Section 4.1.4**). The difference between using **Equation 6** and **Equation 9** was 0 +/- 0.5% at 55F OAT with no compressors operating. The difference between measuring OAF at 55F with no compressors operating and 95F with both compressors operating was 1.7 +/- 1.3%. It is difficult to measure the average mixed air temperature with a shielded thermocouple array due to airflow variations at each sensor. Without discrete airflow measurements each sensor measurement must be equally weighted. Calculating outdoor airflow fractions at 55F with no compressors operating provide a more accurate measurement of the average mixed air temperature and outdoor air fraction. The fan adds heat to the mixed air but does not increase the humidity ratio. Based on accurate measurements of drybulb and wetbulb temperatures, the humidity ratio can be calculated for return, outdoor, and mixed air. Properly accounting for fan heat using to calculate the mixed air drybulb temperature and humidity ratios noted previously, provides all information required to calculate the return, outdoor, and mixed-air enthalpies to calculate the outdoor air fraction using **Equation 6**. A reasonable estimate of the outdoor air fraction can also be calculated using **Equation 1** with both compressors operating with at least a 20F drybulb temperature difference between outdoor air and return air. In the laboratory the mixed-air temperature was measured with an array consisting of 22 shielded-drybulb temperature sensors located upstream of the air filter next to the evaporator inlet. Therefore, fan heat is not included in the calculations.

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[115] The "code tester" is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92). Standard Methods for Laboratory Airflow Measurement.

Table 53: Economizer #1 OAF Calculated using Equations 6 and 9 at 55F OAT without Compressors and Eq. 1 at 95F with Compressors for 7.5-ton TXV RTU2 and ~400 scfm/ton

Description	Test	Evap Airflow scfm/ ton	Eq. 6 Calc OAF_e at 55F %	Eq. 9 Calc OAF_t at 55F %	Test	Evap Airflow scfm/ ton	Eq. 1 Calc OAF_m at 95F %
No Economizer	T2-TRN-100A-55-NE	400	5.7	7.0	3T-75629575-NE3-Retest	383	2.5
Closed	T2-TRN-100A-55-CE-DM-2	392	12.1	12.4	T2-TRN-95-CE-DM	391	8.7
10% Open (2.8V)	T2-TRN-100A-55-1E-DM	394	11.6	11.9	T2-TRN-95-1ER-DM	392	8.9
20% Open (3.6V)	T2-TRN-100A-55-2E-DM	397	15.5	15.9	T2-TRN-95-2ER-DM	392	11.4
30% Open (4.4V)	T2-TRN-100A-55-3E-DM	400	19.4	19.7	T2-TRN-95-3ER-DM	395	20.9
1F (5.1V)	T2-TRN-100A-55-1F-DM	402	23.5	23.6	T2-TRN-95-1FER-DM	395	22.4
2F (6V)	T2-TRN-100A-55-2F-DM	404	31.1	30.8	T2-TRN-95-2FER-DM	394	30.9
3F (6.9V)	T2-TRN-100A-55-3F-DM	405	39.7	39.3	T2-TRN-95-3FER-DM	394	41.9
100% Open (10V)	T2-TRN-100A-55-OE-DM	402	72.7	70.9	T2-TRN-95-OER-DM	380	68.4

Table 54 provides the calculated economizer #0 outdoor air fraction using **Equation 6** at 55F OAT and 400 scfm/ton for the 7.5-ton RTU2 with OEM analog controller/actuator and unsealed and sealed perimeter. For sealed tests, the economizer hood was removed and the gap between the economizer and the cabinet was sealed with tape. In the field, UL-181 waterproof tape must be used to seal the perimeter. With no economizer installed the unsealed OAF was 5.7% and the sealed OAF was 3.1%. The sealed no economizer test has tape on panels to measure cabinet leakage. Sealing the perimeter reduces unintended OAF by 2.1 to 2.4%. Economizer #0 could only be opened slightly beyond 2-fingers with the factory analog controller and actuator installed. Therefore, a digital economizer controller and actuator were installed to test OAF at other damper positions. Unless otherwise noted, the OAF is calculated using **Equation 6** for all other tests.

Table 54: Economizer #0 OAF Calculated using Equation 6 at 55F OAT for RTU2 with OEM Analog Controller/Actuator, Unsealed and Sealed Perimeter and ~400 scfm/ton

Description	Unsealed Perimeter Test	Evap Airflow scfm/ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-100A-55-NE	400	5.7	T2-TRN-100A-55-NE-SC	392	3.1*	2.7
Closed	T2-55-CE-DMM	392	17.9	T2-55-TCE-DMM	390	15.5	2.4
1-Finger 0.7” open	T2-55-1ER-DMM	396	25.3	T2-55-T1ER-DMM	397	23.2	2.1
2-Fingers 1.2” open	T2-55-2ER-DMM	401	29.4				
Maximum Open	T2-55-OER-DMM	403	30.3				

Table 55 provides the calculated economizer #1 outdoor air fraction using **Equation 6** at 55F OAT and 400 scfm/ton for the 7.5-ton RTU2 with retrofit digital controller/actuator and unsealed and sealed perimeter. The OEM economizer was not shipped with a relief damper which is an

optional part. The optional relief damper provides 5.9% OAF. Sealing the economizer perimeter reduces unintended outdoor air leakage by 2.8 to 4.2%. The digital controller opens the damper 42% farther than the analog controller (72.7% versus 30.3% OAF) and provides accurate control when setting minimum positions. Tests for economizer #0 and economizer #1 indicate some of the problems faced by technicians when diagnosing and setting up analog or digital controllers to establish damper positions to meet minimum outdoor air ventilation requirements for acceptable indoor air quality per local building codes.[116] For building occupancies requiring 15% outdoor per ASHRAE 62.1 technicians would generally set the minimum damper position at 2-fingers open or 6V. For economizer #1, this would provide 31.1% outdoor air (with perimeter unsealed) or 107% more than required and reduce cooling and heating efficiency. Setting minimum position at 20% open (3.6V) will provide 15.5% outdoor air and reduce overventilation by 50% compared to 2-fingers open.

Table 55: Economizer #1 OAF Calculated using Eq. 6 at 55F for 7.5-ton TXV RTU2 with Digital Controller/Actuator, Unsealed and Sealed Perimeter and ~400 scfm/ton

Description	Unsealed Perimeter Test	Evap Airflow scfm/ ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-100A-55-NE	400	5.7	T2-TRN-100A-55-NE-SC	392	3.1*	2.7
Relief Dampers	T2-TRN-100A-55-RD	397	5.9				
Closed (2.0V)	T2-TRN-100A-55-CE-DM-2	392	12.1	T2-TRN-100A-55-TCE-DM	390	8.2	3.9
10% Open (2.8V)	T2-TRN-100A-55-1E-DM	394	11.6	T2-TRN-100A-55-T1E-DM	391	8.4	3.2
20% Open (3.6V)	T2-TRN-100A-55-2E-DM	397	15.5	T2-TRN-100A-55-T2E-DM	394	11.7	3.9
30% Open (4.4V)	T2-TRN-100A-55-3E-DM	400	19.4	T2-TRN-100A-55-T3E-DM	398	16.6	2.8
1F (5.1V)	T2-TRN-100A-55-1F-DM	402	23.5	T2-TRN-55-T1FER-DM	395	20.1	3.4
2F (6V)	T2-TRN-100A-55-2F-DM	404	31.1	T2-TRN-55-T2FER-DM	398	28.3	2.8
3F (6.9V)	T2-TRN-100A-55-3F-DM	405	39.7	T2-TRN-55-T3FER-DM	402	35.5	4.2
100% Open (10V)	T2-TRN-100A-55-OE-DM	402	72.7				

The economizer actuator damper position fraction (i.e., incremental volts open divided by total potentiometer voltage) does not provide a one-to-one relationship with the outdoor air fraction. **Figure 24** shows the correlation between outdoor air fraction and damper position with unsealed and sealed perimeter (R^2 = 0.999). The average difference is 3.4 +/- 0.3% between unsealed and sealed OAF. Sealing the perimeter reduces unintended outdoor airflow and improves cooling and heating efficiency. Establishing the most efficient minimum damper position to meet ASHRAE 62.1 requirements is important for health, comfort, and energy efficiency. The outdoor air fraction provided by each economizer will vary as a function of damper position.

[116] ANSI/ASHRAE 2010. ANSI/ASHRAE 62.1-2010. Standard Ventilation for Acceptable Indoor Air Quality.

Figure 24: Outdoor Air Fraction versus Economizer #1 Damper Position for 7.5-ton TXV RTU2 with Unsealed and Sealed Perimeter and ~400 scfm/ton Total Airflow

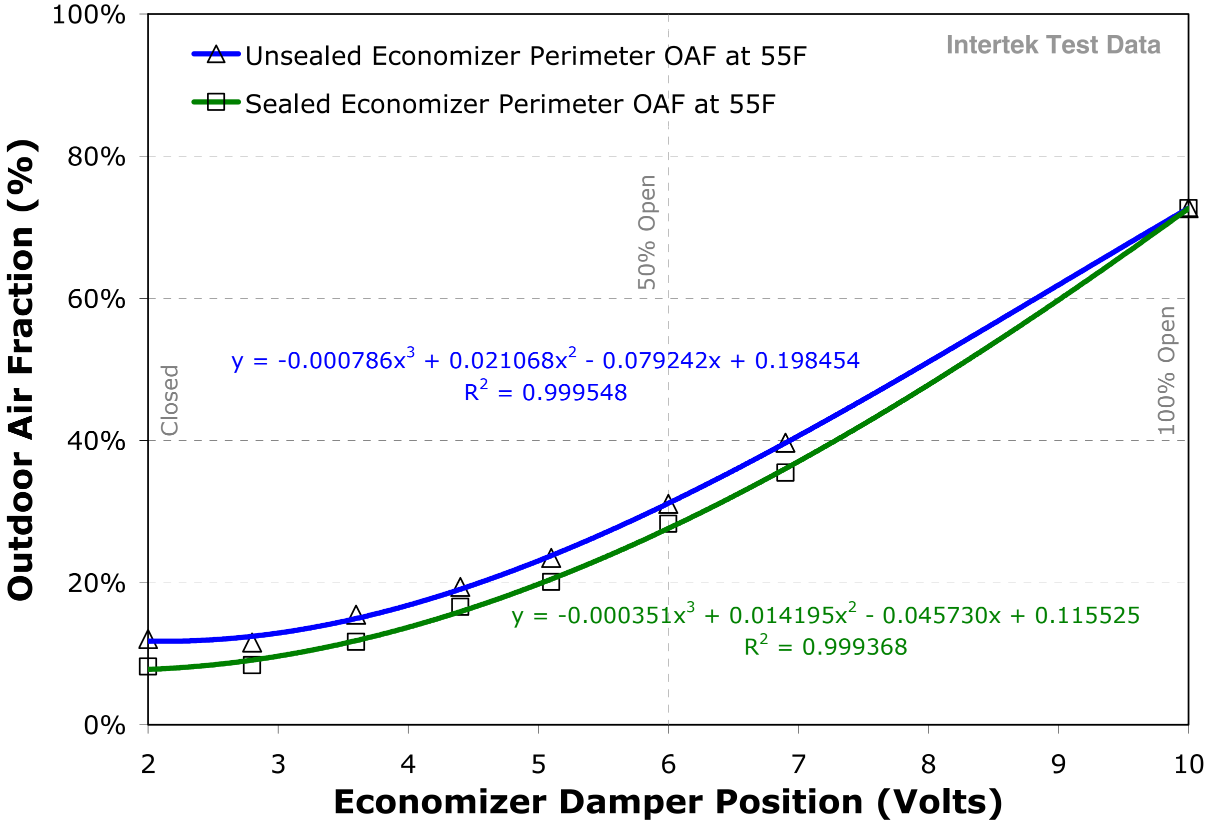


Table 56 provides the economizer #2 outdoor air fraction calculated using **Equation 6** at 55F OAT and 360 scfm/ton for RTU 2 with digital controller/actuator and unsealed and sealed perimeter. Economizer #2 had a factory-installed digital economizer controller and built-in relief dampers. The digital economizer controller was used to open the economizer 2.8, 3.6, 4.4, and 10V corresponding to 10, 20, 30, and 100% open. In the unsealed closed position economizer #2 provides 145% more outdoor air than economizer #1. In the 100% open position it provides 78.9% OAF or 7.4% more outdoor air than economizer #1. For the unsealed 10 to 30% open positions economizer #2 provides 125 to 173% more outdoor air than economizer #1. Sealing the economizer perimeter reduced unintended OAF by 12.6 to 14%.

Table 56: Economizer #2 OAF Calculated using Eq. 6 at 55F OAT for 7.5-ton TXV RTU2 with Digital Controller/Actuator, Unsealed and Sealed Perimeter and ~360 scfm/ton

Description	Unsealed Test	Evap Airflow scfm/ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-100A-55-NE	400	5.7	T2-TRN-100A-55-NE-SC	392	3.1*	2.7
Closed (2.0V)	T2-CAN-55-CE-DM	365	29.7	T2-CAN-55-TCE-DM	348	17.1	12.6
10% Open (2.8V)	T2-CAN-55-1ER-DM	362	31.5	T2-CAN-55-T1ER-DM	346	18.4	13.1
20% Open (3.6V)	T2-CAN-55-2ER-DM	358	34.9	T2-CAN-55-T2ER-DM	339	22.2	12.7
30% Open (4.4V)	T2-CAN-55-3ER-DM	354	46.1	T2-CAN-55-T3ER-DM	335	32.1	14.0
100% Open (10V)	T2-CAN-55-OER-DM	369	78.1	T2-CAN-55-TOER-DM	360	79.3	-1.2

Table 57 provides the calculated economizer #1 outdoor air fraction using **Equation 6** at 55F OAT and 260 scfm/ton for RTU 2 with digital controller/actuator and unsealed and sealed perimeter. With no economizer installed the unsealed OAF is 7.4% and the sealed OAF is 4.1%*. The relief dampers provide 9.5% outdoor airflow. Sealing the gap between the economizer and the cabinet reduced unintended OAF by 1.2 to 2.7%.

Table 57: Economizer #1 OAF Calculated using Eq. 6 at 55F for 7.5-ton TXV RTU2 with Digital Controller/Actuator, Unsealed and Sealed Perimeter and ~260 scfm/ton

Description	Unsealed Test	Evap Airflow scfm/ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-63A-55-NE-DM	261	7.4	T2-TRN-63A-55-NE-DM-SC	257	4.1*	3.3
Relief Dampers	T2-TRN-63A-55-RD-DM	260	9.5				
Closed (2.0V)	T2-TRN-63A-55-CE-DM-2	257	12.3	T2-TRN-63A-55-TCE-DM-2	257	9.8	2.5
10% Open (2.8V)	T2-TRN-63A-55-1E-DM-2	257	12.5	T2-TRN-63A-55-T1E-DM	257	11.3	1.2
20% Open (3.6V)	T2-TRN-63A-55-2E-DM-2	259	15.4	T2-TRN-63A-55-T2E-DM-2	257	12.6	2.7
30% Open (4.4V)	T2-TRN-63A-55-3E-DM-2	262	19.9	T2-TRN-63A-55-T3E-DM-2	260	17.4	2.6
1F (5.1V)	T2-TRN-63A-55-1F-DM-2	262	24.4				
2F (6V)	T2-TRN-63A-55-2F-DM-2	264	31.6				
3F (6.9V)	T2-TRN-63A-55-3F-DM-2	265	40.1				
100% Open (10V)	T2-TRN-63A-55-OE-DM-2	259	75.3				

Table 58 provides the economizer #1 outdoor air fraction calculated using **Equation 6** at 55F OAT and 310 scfm/ton for RTU 2 with digital controller/actuator and unsealed and sealed perimeter. With no economizer installed the unsealed OAF is 7.0% and the sealed OAF is 4.2%. The relief dampers provide 7.2% outdoor air. Sealing the gap between the economizer and the cabinet reduced unintended OAF by 2.2 to 2.9%.

Table 58: Economizer #1 OAF Calculated using Eq. 6 at 55F for 7.5-ton TXV RTU2 with Digital Controller/Actuator, Unsealed and Sealed Perimeter and ~310 cfm/ton

Description	Unsealed Test	Evap Airflow scfm/ ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-75A-55-NE-DM	312	7.0	T2-TRN-75A-55-NE-DM-SC	308	4.2*	2.8
Relief Dampers	T2-TRN-75A-55-RD-DM	312	7.2				
Closed	T2-TRN-75A-55-CE-DM-2	309	12.2	T2-TRN-75A-55-TCE-DM-2	307	9.2	2.9
10% Open (2.8V)	T2-TRN-75A-55-1E-DM	310	12.8	T2-TRN-75A-55-T1E-DM	307	10.0	2.9
20% Open (3.6V)	T2-TRN-75A-55-2E-DM-2	312	15.3	T2-TRN-75A-55-T2E-DM-2	308	12.5	2.8
30% Open (4.4V)	T2-TRN-75A-55-3E-DM-2	315	19.8	T2-TRN-75A-55-T3E-DM-2	313	17.6	2.2
1F (5.1V)	T2-TRN-75A-55-1F-DM-2	316	24.1				
2F (6V)	T2-TRN-75A-55-2F-DM-2	317	31.4				
3F (6.9V)	T2-TRN-75A-55-3F-DM-2	320	40.6				
100 Open (10V)	T2-TRN-75A-55-OE-DM-2	314	74.8				

Table 59 provides the economizer #1 outdoor air fraction calculated using **Equation 6** at 55F OAT and 360 scfm/ton for RTU 2 with digital controller/actuator and unsealed and sealed perimeter. With no economizer installed the unsealed OAF was 6.1% and the sealed OAF was 3.4%*. The relief dampers provide 6.3% outdoor air. Sealing the gap between the economizer and cabinet reduced unintended OAF by 2.6 to 3.1%.

Table 59: Economizer #1 OAF Calculated using Eq. 6 at 55F for 7.5-ton TXV RTU2 with Digital Controller/Actuator Retrofit, Unsealed and Sealed Perimeter and ~360 cfm/ton

Description	Unsealed Test	Evap Airflow scfm/ ton	Eq. 6 Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-88A-55-NE-DM	358	6.1	T2-TRN-88A-55-NE-DM-SC	354	3.4*	2.7
Relief Dampers	T2-TRN-88A-55-RD-DM	357	6.3				
Closed (2.0V)	T2-TRN-88A-55-CE-DM-2	355	12.0	T2-TRN-88A-55-TCE-DM	354	8.9	3.1
10% Open (2.8V)	T2-TRN-88A-55-1ER-DM	355	12.1	T2-TRN-88A-55-T1ER-DM	352	9.6	2.6
20% Open (3.6V)	T2-TRN-88A-55-2E-DM-2	361	15.4	T2-TRN-88A-55-T2E-DM-2	359	12.6	2.8
30% Open (4.4V)	T2-TRN-88A-55-3E-DM-2	363	19.8	T2-TRN-88A-55-T3E-DM-2	363	17.2	2.7
1F (5.1V)	T2-TRN-88A-55-1F-DM-2	364	24.3				
2F (6V)	T2-TRN-88A-55-2F-DM-2	366	31.4				
3F (6.9V)	T2-TRN-88A-55-3F-DM-2	371	39.9				
100 Open (10V)	T2-TRN-88A-55-OE-DM-2	366	73.6				

Table 60 provides economizer #1 outdoor air fraction calculated using **Equation 6** at 55F OAT and 450 scfm/ton for RTU 2 with digital controller/actuator and unsealed and sealed perimeter. With no economizer installed the unsealed OAF is 6.6% and the sealed OAF is 3.3%*. The relief

dampers provide 7.6% outdoor air. Sealing the gap between the economizer and the cabinet reduced unintended OAF by 2.1 to 2.4%.

Table 60: Economizer #1 OAF Calculated using Eq. 6 at 55F for 7.5-ton TXV RTU2 with Digital Controller/Actuator, Unsealed and Sealed Perimeter and ~450 cfm/ton

Description	Unsealed Test	Evap Airflow scfm/ ton	Eq. Calc Unsealed OAF_e at 55F %	Sealed Perimeter Test	Evap Airflow scfm/ ton	Eq. 6 Calc Sealed Perimeter OAF_e at 55F %	ΔOAF %
No Economizer	T2-TRN-110A-55-NE-DM-2	444	6.6	T2-TRN-110A-55-NE-DM-SC	442	3.3*	3.3
Relief Dampers	T2-TRN-110A-55-RD-DM	443	7.6				
Closed (2.0V)	T2-TRN-110A-55-CE-DM-2	438	12.1	T2-TRN-110A-55-TCE-DM	444	10.0	2.1
10% Open (2.8V)	T2-TRN-110A-55-1E-DM	447	12.6	T2-TRN-110A-55-T1E-DM	446	10.2	2.4
20% Open (3.6V)	T2-TRN-110A-55-2E-DM	450	16.0	T2-TRN-110A-55-T2E-DM	446	13.6	2.4
30% Open (4.4V)	T2-TRN-110A-55-3E-DM	452	20.4	T2-TRN-110A-55-T3E-DM	449	18.3	2.1
1F (5.1V)	T2-TRN-110A-55-1F-DM-2	447	24.2				
2F (6V)	T2-TRN-110A-55-2F-DM-2	450	31.2				
3F (6.9V)	T2-TRN-110A-55-3F-DM-2	453	39.0				
100 Open (10V)	T2-TRN-110A-55-OE-DM-2	452	74.9				

4.2.5 Economizer 95F Efficiency Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the application efficiency impact of economizer outdoor air ventilation for the 7.5-ton TXV RTU2 with economizer #1 and #2 installed with outdoor air dampers closed, partially open, and 100% open and the economizer perimeter unsealed and sealed with tape. Tests were performed with factory charge and outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. **Table 61** provides economizer #1 outdoor air ventilation impacts versus damper position with unsealed and sealed perimeter and 400 scfm/ton airflow at 95F OAT. With no economizer installed the total EER* was 8.8 and the sensible EER*s was 7.4. With economizer #1 installed and dampers closed the total EER* was 7.6 and sensible EER*s was 6.7. The reduction in efficiency with economizer #1 installed and closed dampers was 13.5% for total EER* and 9% for sensible EER*s compared to no economizer. With closed dampers the economizer #1 efficiency was 31% less than the AHRI EER rating of 11.0 and 17% less than the sensible EER at the AHRI test conditions. Opening economizer dampers per the outdoor air leakage tests performed at 55F in the previous section significantly reduced efficiency. The minimum tested application efficiency was 3.8 EER* for unsealed and 3.9 EER* for sealed perimeter which are 53 to 66% less than the AHRI rating and 49 to 57% less than the application efficiency with no economizer installed. If a building requires 15% outdoor air per ASHRAE 62.1, then economizer #1 would provide 15.5% OAF with minimum damper position of 20% open (3.6V) and unsealed perimeter. If a technician set the minimum damper position at 2-fingers open, the economizer #1 would provide 31.1% OAF or 100% more outdoor ventilation than 20% open. The overventilation at 2-fingers open would reduce EER*s to 5.7 EER*s or 13.6% less efficient than 6.6 EER*s at 20% open. Providing adequate outdoor ventilation air is as important as providing comfortable indoor temperature control. The reduction in efficiency due to overventilation beyond minimum requirements represents an important energy efficiency opportunity for space cooling and heating.

Table 61: Economizer #1 Outdoor Air Ventilation Impacts versus Damper Position Sealed and Unsealed Perimeter for 7.5-ton TXV RTU2 and 400 scfm/ton at 95F OAT

Description	Test	Unsealed Total EER*	Unsealed Sensible EER*s	Test	Sealed Perimeter Total EER*	Sealed Perimeter Sensible EER*s
No Economizer	3T-75629575-NE3-Retest	8.8	7.4	NA	NA	NA
Closed	T2-TRN-95-CE-DM	7.6	6.7	T2-TRN-95-TCE-DM	8.0	6.9
10% Open (2.8V)	T2-TRN-95-1ER-DM	7.6	6.7	T2-TRN-95-T1ER-DM	7.9	6.8
20% Open (3.6V)	T2-TRN-95-2ER-DM	7.2	6.6	T2-TRN-95-T2ER-DM	7.5	6.7
30% Open (4.4V)	T2-TRN-95-3ER-DM	6.6	6.2	T2-TRN-95-T3ER-DM	7.0	6.5
1F (5.1V)	T2-TRN-95-1FER-DM	6.1	6.1	T2-TRN-95-T1FER-DM	6.5	6.3
2F (6V)	T2-TRN-95-2FER-DM	5.7	5.7	T2-TRN-95-T2FER-DM	5.8	5.8
3F (6.9V)	T2-TRN-95-3FER-DM	5.3	5.3	T2-TRN-95-T3FER-DM	5.4	5.4
100% Open (10V)	T2-TRN-95-OER-DM	3.8	3.8	T2-TRN-95-TOER-DM	3.9	3.9

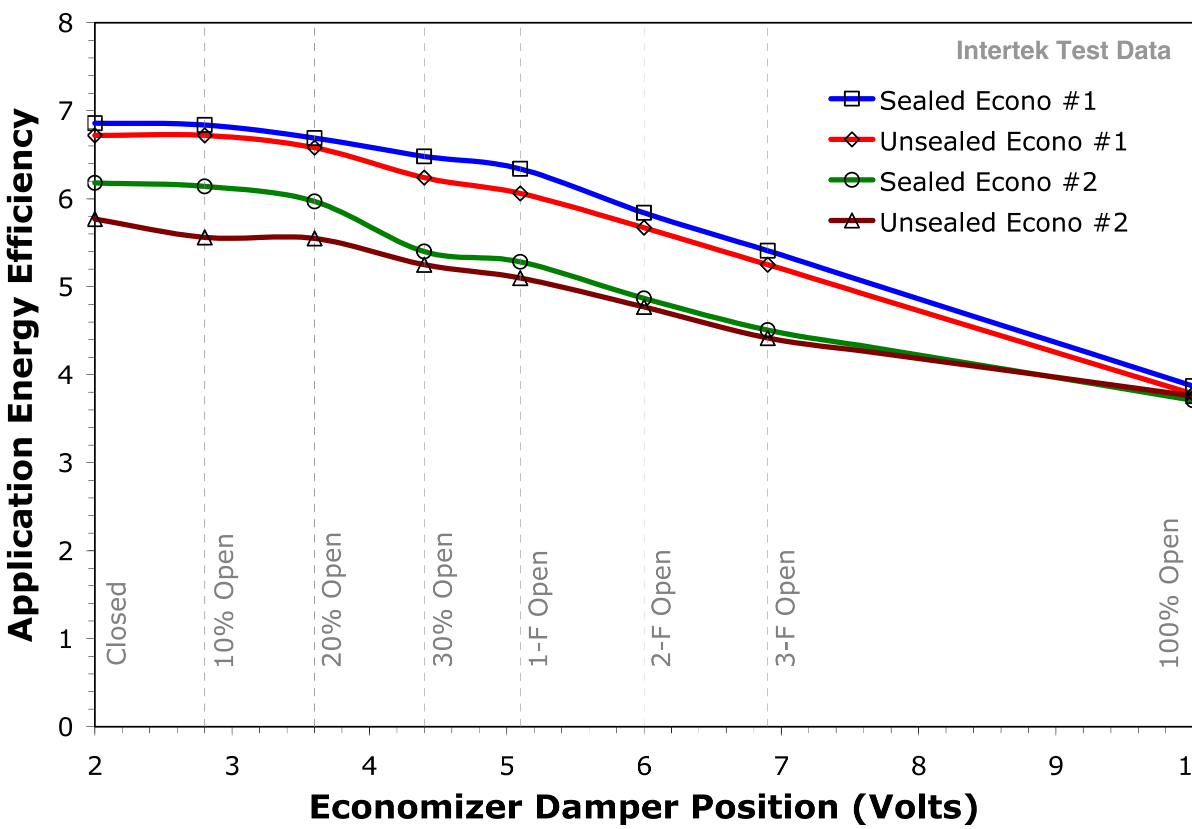
Table 62 provides economizer #2 outdoor air ventilation impacts versus damper position with unsealed and sealed perimeter and 370 scfm/ton at 95F OAT. With economizer #2 installed and dampers closed the total EER* was 5.9 and sensible EER*s was 5.8. The reduction in efficiency economizer #2 installed and closed dampers was 33% for total EER* and 22% for sensible EER*s. With closed dampers the economizer #2 efficiency was 46% less than the AHRI EER rating of 11.0 and 29% less than the sensible EER for the AHRI test conditions. The minimum tested efficiency was 3.8 EER* for unsealed and 3.7 EER* for sealed perimeter which are 54 to 66% less than the AHRI rating and 50 to 58% less than the efficiency with no economizer installed.

Table 62: Economizer #2 Outdoor Air Ventilation Impacts versus Damper Position Sealed and Unsealed Perimeter for 7.5-ton TXV RTU2 and 370 scfm/ton at 95F OAT

Description	Test	Unsealed Total EER*	Unsealed Sensible EER*s	Test	Sealed Perimeter Total EER*	Sealed Perimeter Sensible EER*s
No Economizer	3T-75629575-NE3-Retest	8.8	7.4	NA	NA	NA
Closed (2.0V)	T2-CAN-95-CE-DM	5.9	5.8	T2-CAN-95-TCE-DM	7.2	6.2
10% Open (2.8V)	T2-CAN-95-1ER-DM	5.6	5.6	T2-CAN-95-T1ER-DM	7.1	6.1
20% Open (3.6V)	T2-CAN-95-2ER-DM	5.7	5.6	T2-CAN-95-T2ER-DM	6.7	6.0
30% Open (4.4V)	T2-CAN-95-3ER-DM	5.3	5.3	T2-CAN-95-T3ER-DM	5.6	5.4
100% Open (10V)	T2-CAN-95-OER-DM	3.8	3.8	T2-CAN-95-TOER-DM	3.8	3.7

Figure 25 shows the decrease in application sensible energy efficiency of RTU2 as dampers are opened from fully closed (2V) to fully open (10V) with economizer #1 and #2 installed. The mandated outdoor ventilation rates for most building occupancies range from 6 to 10% for offices, 22% for retail, 33% for auditoriums and schools, 40% for restaurants and health clubs, and 53% or more for cafeterias and sports arenas. Ventilation rates for unoccupied spaces should be optimized to save energy. While overventilation is in fact a system load, most of the load could be avoided with optimal minimum economizer damper positions and eliminating unintended outdoor air leakage. The impact on energy efficiency due to overventilation loads is realistic. Field observations found approximately 50% of units with economizers not working properly or dampers stuck 10 to 100% open with Molex plugs or other objects stuck between damper blades. The efficiency decrease represents realistic efficiency due to excessive ventilation loads (2 to 5 times greater than ASHRAE 62.1) when OAT is 95F or higher.

Figure 25: Application Sensible Energy Efficiency versus Damper Position for 7.5-ton TXV RTU2 with Economizer #1 and #2 Sealed and Unsealed Perimeter at 95F OAT



Overventilation and unintended outdoor airflow are common maintenance faults on all commercial buildings. Reducing overventilation can have a significant impact on thermal comfort, HVAC efficiency, and energy use. The manufacturer provides tables of information for "troubleshooting" faults with the cooling and heating system. Too much outdoor air is identified as a possible cause of inadequate heating. The manufacturer lists the following faults for inadequate heating: 1) dirty air filter, 2) gas input to unit too low, 3) unit undersized, 4) restricted airflow, 5) blower speed too low, 6) limit-switch causes main burners to cycle, and 7) too much outdoor air. Technicians can check and correct dirty air filters, restricted airflow, and blower speed (fan belt tension/alignment, pulley and motor sheave). Technicians can also check gas input pressure and burner limit switch. Too much outdoor air can be checked and corrected by adjusting the economizer minimum damper position.

4.2.6 Airflow Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of airflow faults on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. Airflow test results for standard static pressure (2-HP blower motor) are provided in **Table 63** including diagnostics for manufacturer refrigerant charge protocols (delta discharge pressure, suction pressure and superheat). The sensible efficiency (EER*s) is greatest at 349 scfm/ton. Total efficiency (EER*) is greatest at lowest airflow (i.e., 252 scfm/ton) and declines as airflow increases. Discharge pressure (ΔDP) passes the manufacturer refrigerant charge protocol for all tests. Suction pressure (ΔSP) passes for airflows greater than or equal to 349 scfm/ton. The circuit 2 delta superheat (ΔSH) passes all tests, but circuit 1 ΔSH fails across all airflows tested. The sensible cooling capacities for all tests are at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load at 95F and highlighted in green. For standard static pressure, the CEC protocol ΔTS is 3.7F at 252 scfm/ton airflow indicating low airflow and the sensible cooling capacity is 10% less than non-faulted 100% airflow tests and both cells are highlighted in red. The CEC ΔTS values for all other tests are between -3F and +3F and highlighted in green.

Table 63: Airflow Fault Tests at Standard Static (2-HP Motor) Pressure for 7.5-ton TXV RTU2 with Economizer #1 at 95F OAT

Test	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	ESP IWC	Fan Power W	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-1875-95-CF-C	-10/-9	-10/-10	6/5	3.7	252	0.27	750	7,680	8.7	49,035	6.38
T2-2250-95-CF	-8/-7	-7/-7	6/5	0.9	304	0.38	970	7,919	8.2	51,622	6.52
T2-2625-95-CF	-6/-5	-4/-5	7/4	-0.3	349	0.51	1,290	8,298	8.0	55,581	6.70
T2-3000-95-CF	-4/-3	-4/-2	6/3	-1.8	388	0.61	1,590	8,645	7.6	57,015	6.60
T2-3240-95-CF	-3/-2	-2/-1	7/3	-2.8	435	0.77	2,140	9,217	6.9	59,658	6.47

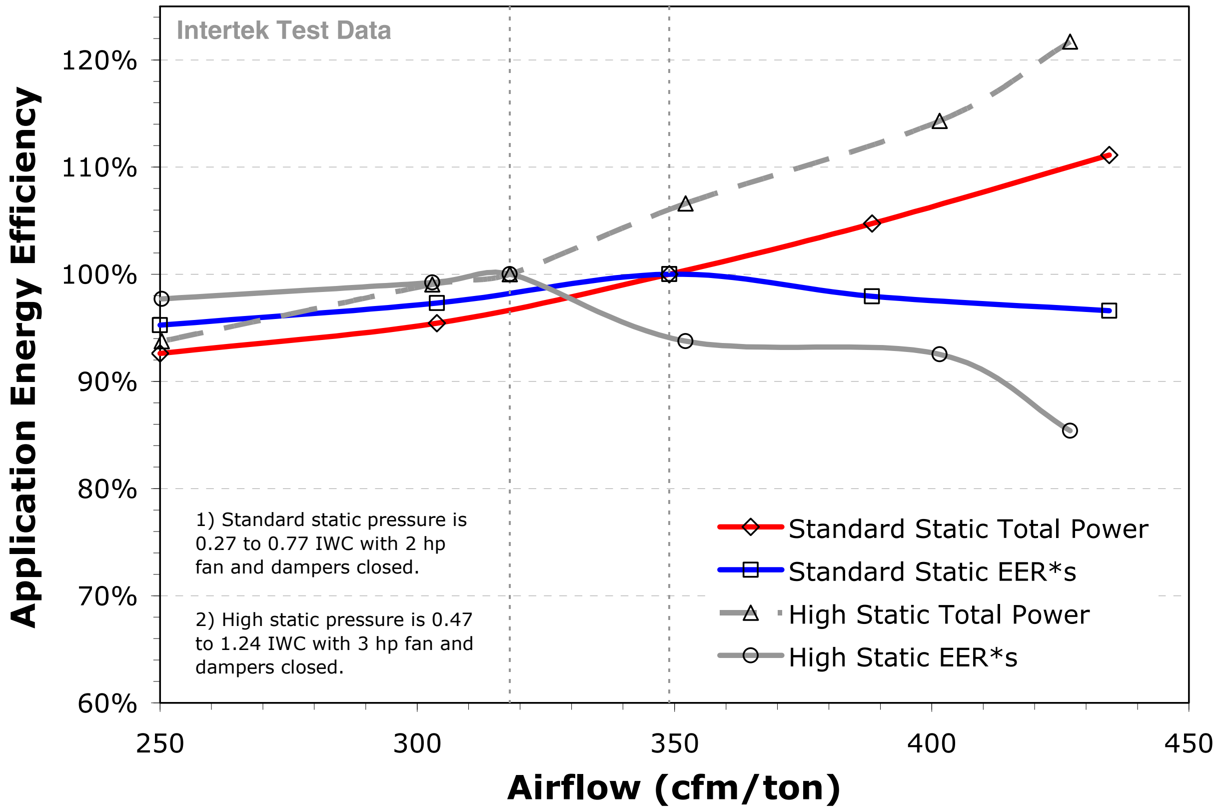
Airflow test results for high static pressure (3-HP blower motor) are provided in **Table 64**. The sensible efficiency (EER*s) is greatest at 318 scfm/ton. Total efficiency (EER*) is greatest at lowest airflow (i.e., 250 scfm/ton) and declines as airflow increases. Delta discharge pressure (ΔDP) passes manufacturer refrigerant charge protocol for all tests. Delta suction pressure (ΔSP) passes for airflows greater than or equal to 352 scfm/ton. Circuit 2 "delta superheat" (ΔSH) passes all tests, but circuit 1 ΔSH is outside tolerances for all tests. The sensible cooling capacities for all tests are at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load at 95F highlighted in green. For high static pressure, the CEC ΔTS is 3.5F at 250 scfm/ton airflow indicating low airflow and the sensible cooling capacity is 10% less than non-faulted 100% airflow tests and both cells are highlighted in red. The CEC ΔTS is -3.9F at 427 scfm/ton airflow test indicating low sensible cooling capacity (highlighted in yellow), but the capacity is acceptable and highlighted in green indicating a "false alarm." All other ΔTS tests are between -3F and +3F and highlighted in green with acceptable sensible cooling capacities highlighted in green.

Table 64: Airflow Fault Tests at High Static (3-HP Motor) Pressure for 7.5-ton TXV RTU2 with Economizer #1 at 95F OAT

Test	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ton	ESP IWC	Fan Power W	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-HS-1875-95-CF-C	-10/-9	-10/-10	6/4	3.5	250	0.47	850	7,778	8.4	48,347	6.22
T2-HS-2250-95-CF	-10/-9	-10/-10	6/5	1.0	303	0.67	1,240	8,217	8.0	51,883	6.31
T2-HS-2384-95-CF	-6/-5	-6/-5	6/3	0.4	318	0.68	1,300	8,296	7.9	52,786	6.36
T2-HS-2625-95-CF	-5/-4	-5/-4	6/4	-1.4	352	0.91	1,840	8,844	7.0	52,764	5.97
T2-HS-3000-95-CF-C	-4/-2	-3/-1	6/3	-2.7	402	1.08	2,430	9,484	6.5	55,839	5.89
T2-HS-3200-95-CF-C	-3/-2	-2/0	6/3	-3.9	427	1.24	3,010	10,097	5.6	54,859	5.43

Figure 26 shows the application sensible energy efficiency increasing by 2 to 5% with a 21 to 28% decrease in airflow and efficiency decreasing by 3 to 15% with a 25 to 34% increase in airflow. The maximum standard-static pressure efficiency is 350 cfm/ton and maximum high-static efficiency is 320 cfm/ton.

Figure 26: Application Energy Efficiency versus Airflow at Standard and High Static Pressure for 7.5-ton TXV RTU2 with Economizer #1 at 95F OAT



Low airflow below 250 cfm/ton impacts refrigerant charge diagnostics and may cause coil icing when combined with dirty air filters as shown in **Figure 13**. Coil icing is influenced by a combination of low airflow due to filter blockage or low refrigerant charge plus humid return air. These tests demonstrate that adjusting fan speed and airflow at the unit causes less impact on efficiency than increasing ESP with the code tester to emulate faults in the HVAC duct system.

4.2.7 Refrigerant Charge Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of refrigerant charge faults on the application efficiency (EER*) of RTU2 with economizer #1 installed and perimeter unsealed, dampers closed, and airflow at 356 scfm/ton. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. With dampers closed the unsealed outdoor air leakage was 12% per **Table 59**. Tests were performed with factory charge varying from 60 to 140% of factory charge. The circuit-specific manufacturer refrigerant charge diagnostic protocols include delta discharge pressure (ΔDP), suction pressure (ΔSP), and superheat (ΔSH). . The manufacturer tolerances are +/-10 psig for ΔDP, +/-5 psig for ΔSP and +/-5F for ΔSH. The

CEC protocols include delta temperature split (ΔTS) to evaluate airflow and sensible cooling capacity. The CEC ΔTS tolerances are +/-3F.

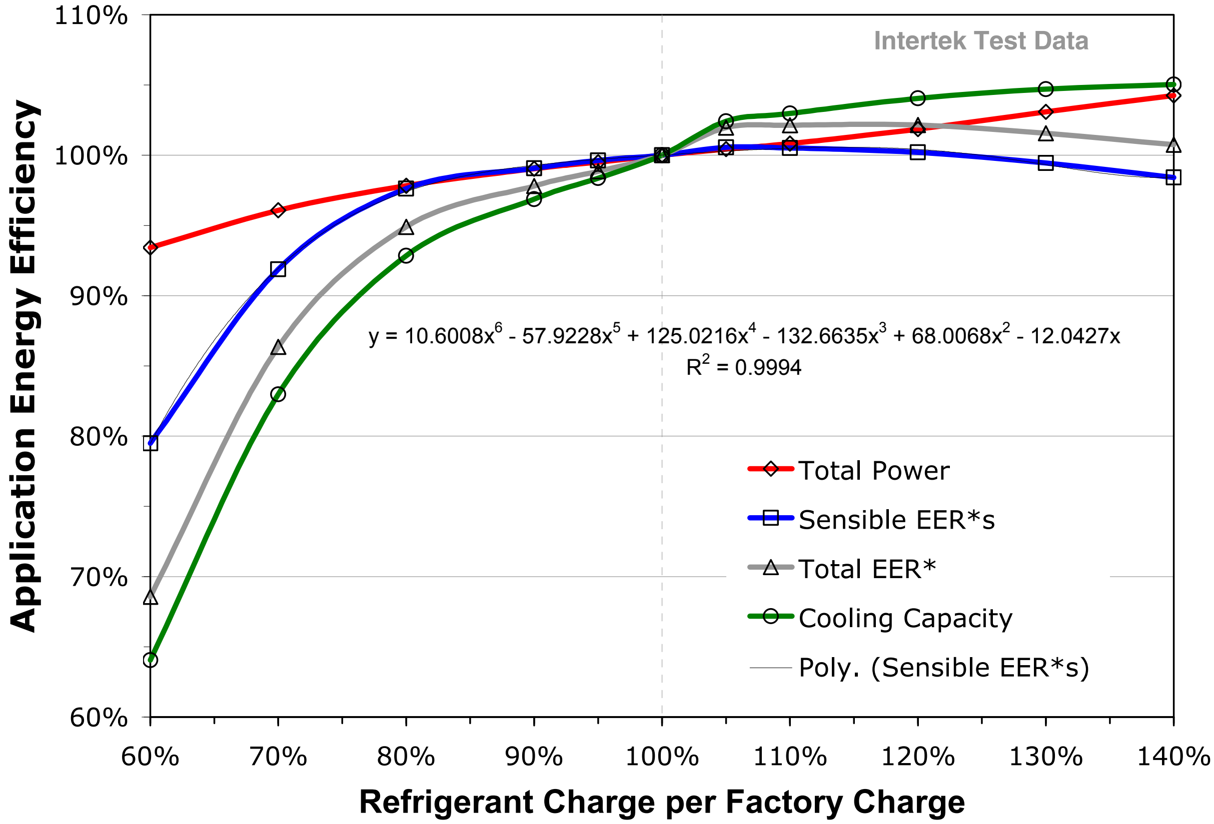
Table 65 provides the refrigerant charge fault impacts and fault detection diagnostics (FDD) at 95F OAT. Efficiency impacts are normalized per 100% factory charge. Application sensible efficiency was maximized from 100 to 110% of factory charge. Undercharging refrigerant by 10 to 40% reduced EER*s by 1 to 20%. Overcharging refrigerant by 10 to 40% reduced EER*s by 0 to 1%. The manufacturer protocols provided 58% accuracy by correctly diagnosing factory charge and 20 to 40% undercharge. Both circuits fail all manufacturer protocols at 30 to 40% undercharge, and both circuits fail discharge pressure and superheat protocols at 20% undercharge. The CEC temperature split protocols provided 100% accuracy by correctly diagnosing proper airflow and cooling capacity for all tests from 30% undercharge to 40% overcharge and low capacity for 40% undercharge. The sensible cooling capacities for all tests except one are at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load at 95F highlighted in green. The 60% charge test was less than the ACCA Manual N sensible cooling capacity and highlighted in red. At 40% overcharge the C1 DP was 256 psig and the C2 DP was 273 psig. The manufacturer does not provide high- or low-pressure cut-out limits.

Table 65: Refrigerant Charge Fault Impacts with Equal Charge per Circuit for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 356 scfm/ton at 95F OAT

Test	C1/C2 Refrig Chg %	Mfr C1/C2 ΔDP	Mfr C1/C2 ΔSP	Mfr C1/C2 ΔSH	CEC Protocol ΔTS	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*
Undercharge										
T2-RC6-60C1-60C2-90A-95-CE	60/60	-30/-29	-25/-18	40/35	-5.0	7,722	42,944	5.6	42,340	5.5
T2-RC5-70C1-70C2-90A-95-CE	70/70	-19/-19	-14/-7	30/25	-2.4	7,942	55,633	7.0	50,338	6.3
T2-RC4-80C1-80C2-90A-95-CE	80/80	-12/-12	-8/-4	23/10	-1.1	8,087	62,250	7.7	54,459	6.7
T2-RC3-90C1-90C2-90A-95-CE	90/90	-7/-8	-5/-3	10/5	-0.6	8,187	64,951	7.9	55,947	6.8
T2-RC2-95C1-95C2-90A-95-CE	95/95	-6/-7	-5/-4	7/4	-0.4	8,225	65,955	8.0	56,523	6.9
Factory Charge										
T2-RCB1-100C1-100C2-90A-95-CE-5	100/100	-5/-5	-5/-4	5/4	-0.3	8,266	67,047	8.1	57,018	6.9
Overcharge										
T2-RC7-105C1-105C2-90A-95-CE	105/105	-4/-2	-5/-4	4/4	0.0	8,303	68,668	8.3	57,595	6.9
T2-RC8-110C1-110C2-90A-95-CE	110/110	-3/0	-5/-4	4/3	0.1	8,335	69,048	8.3	57,793	6.9
T2-RC9-120C1-120C2-90A-95-CE	120/120	-2/6	-6/-4	4/3	0.2	8,420	69,766	8.3	58,201	6.9
T2-RC10-130C1-130C2-90A-95-CE	130/130	1/13	-6/-4	4/2	0.3	8,521	70,199	8.2	58,449	6.9
T2-RC11-140C1-140C2-90A-95-CE	140/140	3/19	-6/-4	3/2	0.3	8,618	70,424	8.2	58,510	6.8

Figure 27 shows the application energy efficiency impacts versus refrigerant charge per factory charge with equal percentage charge per circuit for the 7.5-ton TXV RTU2 at 95F OAT. Total and EER*s are maximized at 100 to 110% of factory charge. For 5 to 40% undercharge EER*s decreased by up to 21% and capacity decreased by 26 to 36%. For overcharge EER*s increased by 0.6% or decreased by 1.6% and cooling capacity increased by 2.6 to 5%.

Figure 27: Application Energy Efficiency versus Refrigerant Charge per Factory Charge (equal percentage per circuit) for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 356 scfm/ton at 95F OAT



The polynomial regression equation curve-fit of sensible energy efficiency (EER^*_s) versus refrigerant charge per factory charge for the 7.5-ton TXV RTU2 at 95F is shown in **Figure 27**. **Equation 11** and **Equation 16** can be used to calculate EER*s impacts associated with refrigerant charge adjustments based on recovery and weigh-out of refrigerant charge and reported charge adjustment.

Equation 16 $$y = 10.601^6 - 57.923^5 + 125.022x^4 - 132.664x^3 + 68.007x^2 - 12.043x$$

Where,

y = EER*s impact at refrigerant charge per factory charge ratio (dimensionless)

x = refrigerant charge per factory charge ratio (dimensionless)

Figure 28 shows the sight glass refrigerant charge FDD with white bubbles in circuit 1 at 90% of factory charge and no bubbles in circuit 2 with 100% factory charge. Liquid line sight glasses provide non-intrusive refrigerant charge FDD for undercharged conditions.

Figure 28: Sight Glass FDD Shows White Bubbles in C1 with 90% Factory Charge

Undercharging can cause icing of the evaporator coil by reducing the evaporator saturation temperature below freezing causing water condensing on the coil to freeze into ice. Coil icing can reduce airflow and decrease efficiency even more. Icing of the coil was avoided while performing undercharge tests by operating the fan only (no compressors) in between tests, checking evaporator coil to make sure no ice was present and condensate pan was dry. For this unit, overcharging produced no efficiency improvement. Overcharging causes liquid refrigerant to flood the compressor during normal operation and start-up and dilute oil causing inadequate bearing lubrication and premature failure. Procedures for troubleshooting and servicing air conditioning systems are provided in technician training text books.[117] The most common problems are high or low discharge or suction pressure or continuous compressor operation. These problems are caused by a number of faults including: 1) dirty air filter, 2) blocked evaporator/condenser, 3) low cooling capacity or excessive outdoor air, 4) insufficient evaporator airflow, 5) refrigerant restriction, 6) non-condensables, 7) thermostat defective/set too low, 8) low line voltage (faulty contactor/transformer), 9) defective compressor/overload, or 10) refrigerant over/undercharge. Prior to adjusting refrigerant charge, technicians need to check and correct all other faults on the list. If none of the other faults are present and problem still exists, then refrigerant charge FDD and adjustments might be necessary.

[117] Tomczyk, J. 1995. Troubleshooting and Servicing Modern Air Conditioning and Refrigeration Systems. ESCO Press. Mt. Prospect, Ill.: Educational Standards Corporation.

4.2.8 Condenser Blockage Fault Tests for 7.5-ton TXV RTU2

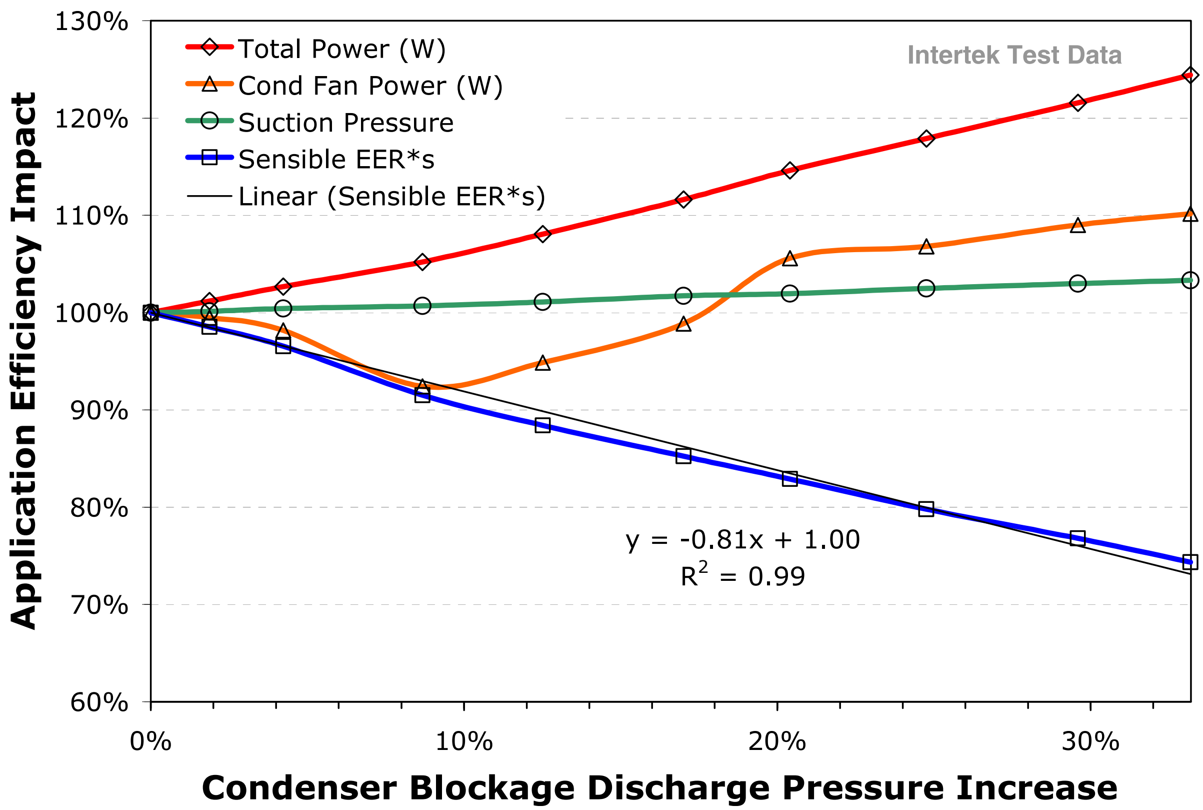
Laboratory tests were performed to evaluate the impact of condenser blockage faults on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed, economizer perimeter unsealed, and airflow of 360 scfm/ton. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. The condenser coil was blocked on the outside of the coil with plastic corrugated cardboard used to ship condensers (to block but not damage fins). Coil blockage was increased incrementally to increase discharge pressure by across both refrigerant circuits. Condenser blockage test results are provided in **Table 66**. Diagnostic tests results are provided for manufacturer refrigerant charge protocols (discharge pressure ΔDP, suction pressure ΔSP, and superheat ΔSH), CEC refrigerant charge protocols (superheat ΔSH), and CEC temperature split (ΔTS) protocols. The manufacturer ΔSP refrigerant charge protocols correctly diagnosed proper charge for all tests and for ΔDP and ΔSH from 0 to 10% condenser coil blockage. For 30 to 80% coil blockage the manufacturer ΔDP protocols misdiagnosed coil blockage as overcharge and ΔSH misdiagnosed undercharge. The CEC temperature split (ΔTS) protocol correctly diagnosed proper airflow and proper capacity for all coil blockage tests. Condenser coil blockage decreased the application sensible capacity by 1 to 7%, and all tests had capacities greater than 105% of the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 66: Condenser Blockage Fault Impacts for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 360 scfm/ton at 95F OAT

Test	Average Discharge Pressure %	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Cond. Fan Power W	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-MFB3-100C1-100C2-90A-95-CE-2	0.0	-6/-5	-5/-4	5/4	-1.2	702	8,247	8.1	57,429	7.0
T2-MF45-100C1-100C2-90A-CB05-CCE	1.9	-2/-1	-4/-4	5/3	-0.4	698	8,345	8.0	57,256	6.9
T2-MF46-100C1-100C2-90A-CB10-CCE	4.2	4/5	-4/-3	5/3	-0.5	689	8,467	7.8	56,928	6.7
T2-MF47-100C1-100C2-90A-CB20-CCE	8.7	14/17	-4/-3	7/4	-0.9	648	8,676	7.3	55,289	6.4
T2-MF48-100C1-100C2-90A-CB30-CCE	12.5	24/27	-3/-3	7/6	-1.0	666	8,913	7.1	54,875	6.2
T2-MF49-100C1-100C2-90A-CB40-CCE	17.0	35/38	-3/-3	6/6	-1.0	694	9,205	6.8	54,641	5.9
T2-MF50-100C1-100C2-90A-CB50-CCE	20.4	43/47	-2/-3	6/6	-1.0	741	9,452	6.6	54,569	5.8
T2-MF51-100C1-100C2-90A-CB60-CCE	24.8	53/58	-2/-2	6/6	-1.2	749	9,723	6.3	54,029	5.6
T2-MF52-100C1-100C2-90A-CB70-CCE	29.6	65/71	-2/-2	6/6	-1.4	765	10,026	6.0	53,614	5.3
T2-MF53-100C1-100C2-90A-CB80-CCE	33.2	74/80	-1/-2	6/6	-1.5	773	10,261	5.7	53,134	5.2

Figure 29 shows the application energy efficiency impacts versus discharge pressure increase due to condenser coil blockage at 95 OAT. Total EER* and sensible EER*s were maximized with no coil blockage. Discharge pressure increased by 2 to 30%, total power increased by 1 to 24%, and sensible efficiency decreased by 1 to 26%.

Figure 29: Application Efficiency Impacts versus Discharge Pressure Increase due to Condenser Blockage for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 360 scfm/ton at 95F OAT



Equation 17 is the regression equation shown in **Figure 29**. **Equation 17** can be used to calculate the EER*s impact associated with blocked/clean discharge pressure ratio increase at constant OAT due to condenser coil blockage for the 7.5-ton TXV unit.

Equation 17 $y_c = -0.81x_p + 1$

Where,

y_c = EER*s impact of condenser coil blockage based on discharge pressure ratio increase (dimensionless)

$x_p = \frac{DP_b}{DP_c} - 1$ = discharge pressure (DP) ratio increase due to condenser coil blockage (dimensionless)

While this manufacturer does not provide "troubleshooting" procedures to diagnose condenser blockage faults, procedures discussed for the 7.5-ton non-TXV unit could be used to distinguish condenser blockage from other faults. The most common problem is excessive head pressure caused by the following faults: 1) dirty air filter, 2) dirty condenser coil, 3) refrigerant overcharge, 4) air in system (non-condensables), and 5) condenser air restricted or short-cycling. Technicians can easily check and correct dirty air filters and dirty (or blocked) condenser. If these corrections eliminate excessive head pressure, then there is no reason to adjust refrigerant charge or check for non-condensables.

4.2.9 Evaporator Blockage Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of evaporator coil blockage faults on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. The evaporator coil was blocked with plastic corrugated cardboard on the upstream side next to the air filter. The inlet area was blocked 30 and 50% to reduce evaporator airflow by 8 to 18%. Preliminary tests were performed without code tester installed before each coil blockage test to match total static pressure with the code tester installed.

Evaporator coil blockage test results are provided in **Table 67**. Diagnostic tests results are provided for manufacturer refrigerant charge protocols (discharge pressure ΔDP, suction pressure ΔSP, and superheat ΔSH), CEC refrigerant charge protocols (superheat ΔSH), and CEC temperature split (ΔTS) protocols. For the baseline test, the manufacturer ΔDP and ΔSP protocols diagnose proper refrigerant charge, but ΔSH misdiagnosed slight undercharge for circuit 1. At 30% evaporator coil blockage both circuits pass ΔDP and C1 barely fails ΔSP and ΔSH. At 50% evaporator coil blockage, ΔDP and ΔSP misdiagnose undercharge and ΔSH correctly diagnoses proper charge. The CEC ΔTS protocol correctly diagnosed proper airflow for and sensible cooling capacity for the baseline and 30% blockage which have sensible cooling capacities at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load and highlighted in green. The CEC ΔTS protocol misdiagnosed capacity for 50% evaporator blockage where capacity is less than the ACCA Manual N cooling load and highlighted in red.

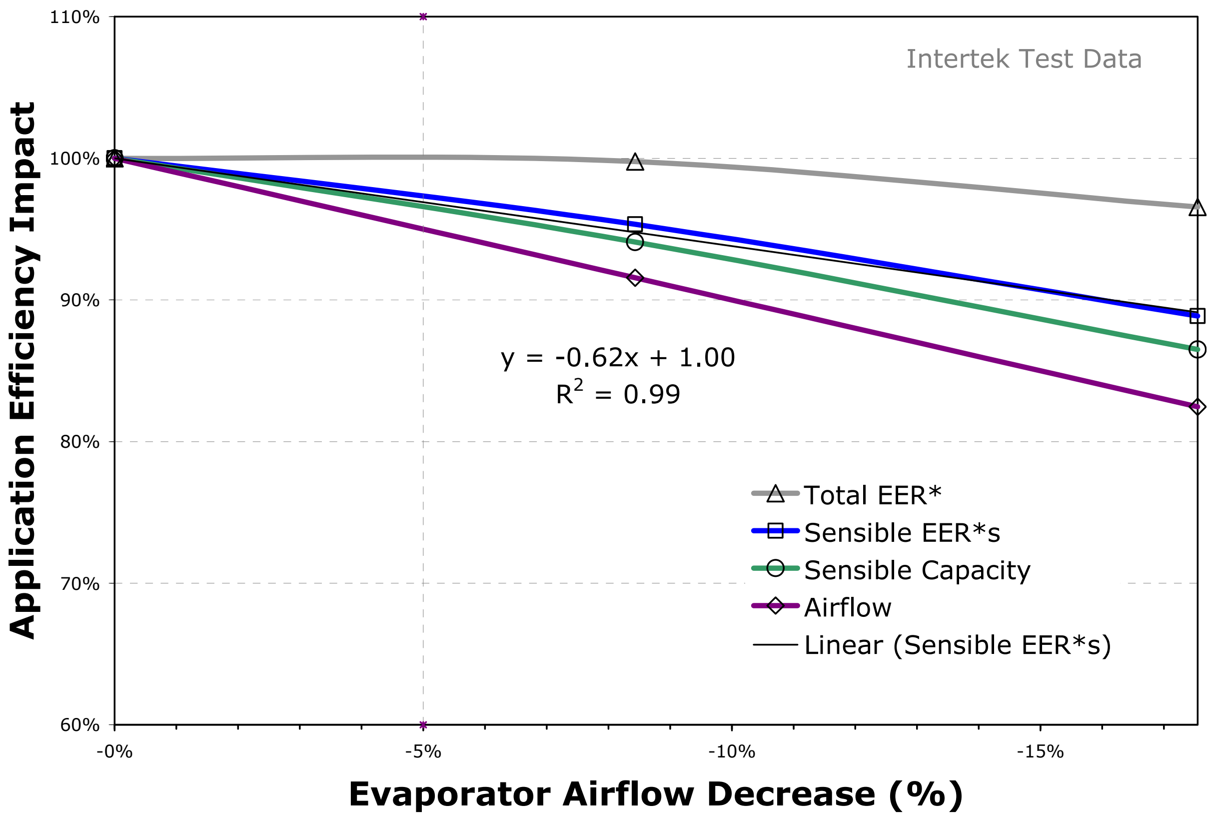
Table 67: Evaporator Coil Blockage Fault Impacts for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed at 95F OAT

Test	Airflow Decrease %	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Airflow scfm/ Ton	Suction Press C1/C2 psig	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-CBASE-3000-95-CE	0.0%	-8/-7	-3/-3	6/5	-2.0	397	76/78	7.66	57,427	6.69
T2-E-3000-95-CE-B30%	8.4%	-9/-8	-5/-6	6/5	-1.6	363	74/75	7.64	54,027	6.38
T2-E-3000-95-CE-B50%	17.5%	-12/-11	-9/-10	5/5	-1.3	327	70/71	7.39	49,675	5.94

Figure 30 shows the energy efficiency impacts versus evaporator airflow decrease. Total EER* and EER*s are maximized with no blockage, and efficiency decreased by 5 to 11% as evaporator

airflow decreased from by 8 to 18%. Low airflow reduced EER*s by 0.2% and total efficiency by 7% (see **Table 63**). Evaporator blockage lowered EER*s more than total efficiency while low airflow only lowered total efficiency with a very small change in EER*s.

Figure 30: Application Efficiency Impacts versus Evaporator Airflow Decrease due to Coil Blockage for 7.5-ton TXV RTU2 with Economizer #1 and Damper Closed



Equation 18 is the regression equation shown in **Figure 30**. **Equation 18** can be used to calculate the EER*s impact associated with dirty/clean airflow ratio due to evaporator coil blockage for the 7.5-ton TXV unit.

Equation 18 $y_e = -0.62x_a + 1$

Where,

y_e = EER*s impact of evaporator coil blockage based on airflow ratio decrease (dimensionless)

$x_a = \frac{cfm_b}{cfm_c} - 1$ = airflow ratio decrease due to evaporator coil blockage (dimensionless)

While this manufacturer does not provide "troubleshooting" procedures to diagnose evaporator blockage faults, procedures discussed for the 7.5-ton non-TXV unit could be used to diagnose evaporator blockage. The most common problem is low suction pressure caused by the following faults: 1) dirty air filter and evaporator coil, 2) low refrigerant charge, 3) metering device or low-side restriction, 4) insufficient evaporator airflow, 5) temperature too low in conditioned space, or 6) filter drier restriction. Technicians can easily check and correct dirty air filter and evaporator coil. If these maintenance procedures eliminate low suction pressure faults, then there is no reason for additional FDD or correction. The manufacturer ΔSP is outside tolerances at 30 and 50% blockage and ΔDP is outside tolerances at 50% blockage, but both protocols pass with no blockage. The CEC ΔTS protocol misdetected the 50% evaporator coil blockage fault even though sensible cooling capacity and airflow were 10% less than the unblocked baseline test. The misdetection was caused by low airflow increasing temperature split, and coil blockage reducing evaporator heat transfer surface area. These tests indicate the importance of technicians following systematic procedures of checking and correcting obvious maintenance faults such as cleaning the evaporator coil and installing clean air filters before performing FDD services.

4.2.10 Restriction Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of refrigerant line restrictions on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed, economizer perimeter unsealed, and airflow of 400 scfm/ton. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In order to emulate liquid line restriction faults a "service" valve was installed upstream of the liquid line driers on each circuit. The service valve can be partially closed to cause a suction pressure reduction and liquid line refrigerant temperature reduction to emulate a restriction at the liquid line drier or expansion device.[118] All restriction tests were performed with the circuit 1 (C1) service valve partially closed. Refrigerant line restriction tests imposed a 15 to 28 psig suction pressure drop and 15 to 20F liquid temperature drop across the restriction. Unrestricted tests have a 1F temperature increase. The unsealed economizer closed damper outdoor airflow rate for all baseline and restriction tests was 12.1%.

Restriction test results are provided in **Table 68**. Diagnostic test results are provided for manufacturer refrigerant charge protocols (discharge pressure ΔDP, suction pressure ΔSP, and superheat ΔSH) and CEC temperature split (ΔTS) protocols. The C1 restriction reduced suction pressure by 19 to 39% and evaporator saturation temperature by 22 to 51% depending on OAT. The C1 restriction reduced EER*s by 11 to 15% and total application efficiency by 8 to 14%. For the baseline tests at 75F and 95F, the manufacturer ΔDP and ΔSP correctly diagnosed refrigerant charge for both circuits and ΔSH diagnosed proper charge for C2 and misdiagnosed undercharge for C1. At 115F, the manufacturer ΔDP and ΔSH protocols misdiagnosed undercharge for both circuits and ΔSP misdiagnosed undercharge for C2. For the C1 restriction tests the manufacturer

[118] Restriction "service" valve turns closed were recorded for each test.

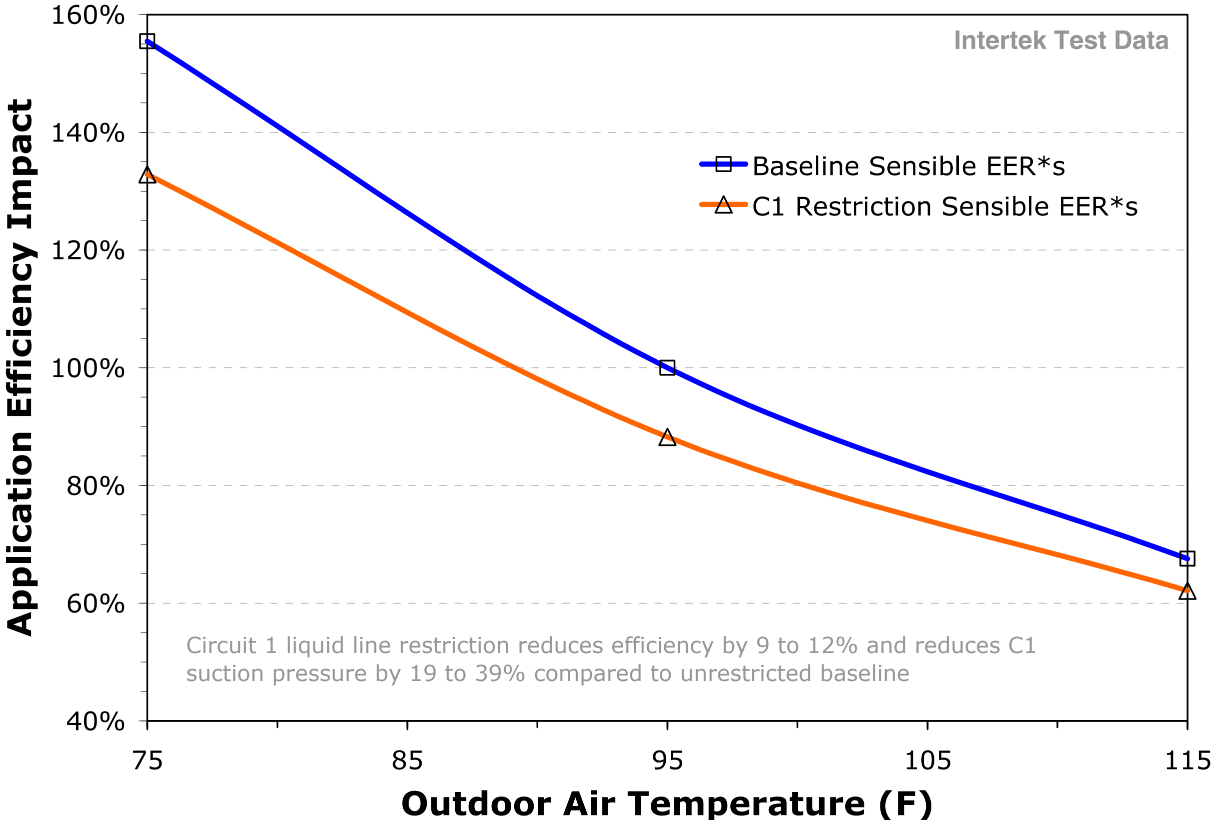
protocols misdiagnosed the restriction as undercharged. The C1 EST was below freezing at 75 and 95F OAT and 11F below C2 at 115F. For the baseline tests, the CEC ΔTS protocol diagnosed proper airflow and sensible cooling capacity at 75F and 95F highlighted in green and low capacity at 115F highlighted in yellow. For the C1 restriction tests the CEC ΔTS protocol diagnosed low cooling capacity at 95F and 115F and proper airflow at 75F highlighted in yellow. At 75F OAT, the CEC ΔTS protocol diagnosed proper airflow and misdiagnosed low cooling capacity. At 75F and 95F OAT, the baseline tests have sensible cooling capacities at least 105% of the ACCA Manual N sensible cooling load highlighted in green. At 115F OAT, the sensible cooling capacity was less than the 61,132 Btuh ACCA Manual N and is highlighted in red. All C1 restriction tests have sensible capacities 10% less than baseline capacities highlighted in red.

Table 68: Restriction Fault Impacts for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 400 scfm/ton Total Airflow at 75, 95, and 115F OAT

Test	OAT F	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Filter Drier C1/C2 ΔT	EST C1/C2 F	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
No Restriction											
T2-RBASE-3000-75-CE	75	-2/-4	2/0	6/-3	0.8	0/-1	43/42	7,274	11.9	67,164	9.2
T2-CBASE-3000-95-CE	95	-7/-7	-3/-3	6/5	-2.0	0/0	45/46	8,586	7.7	57,427	6.7
T2-RBASE-3000-115-CE	115	-15/-11	-2/-8	7/7	-4.0	0/-1	46/48	10,128	5.2	50,172	5.0
C1 Restriction											
T2-R-3000-75-CE	75	-19/-8	-26/-1	37/-3	-2.0	15/-1	21/42	7,062	10.2	57,445	8.1
T2-R-3000-95-CE	95	-20/-10	-24/-4	39/5	-3.8	18/-1	29/46	8,425	6.8	51,341	6.1
T2-R-3000-115-CE	115	-23/-13	-21/-6	35/7	-5.5	20/-1	36/47	10,060	4.8	45,413	4.5

Figure 31 shows the application efficiency impact versus OAT with the circuit 1 restriction and 400 scfm/ton airflow and economizer #1 and dampers closed. The restriction causes EER*s to decrease by 12% at 75F, 9% at 95F, and 9% at 115F.

Figure 31: Application Efficiency Impacts versus OAT with Circuit 1 Restriction for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed and 400 scfm/ton Total Airflow



Restrictions are caused by moisture, copper particles, flux/brazing residue, and particulates inside the system when installed or opened for repair. Oil in new refrigerant systems doesn't remain clean very long, especially in R410A systems with POE oils that have powerful solvent effects. Oil in the system quickly combines with moisture, acids, metal particles and other contaminants to produce sludge which plugs the filter drier or gets stuck on expansion devices. Contaminants causing restrictions can damage the compressor, clog metering devices, or make the metering device function improperly. Liquid line filter driers remove moisture, acid, and particulates (<10 microns) to prevent restrictions on field-charged split systems.

While this manufacturer does not provide "troubleshooting" procedures to diagnose restrictions, procedures discussed for the 7.5-ton non-TXV unit can be used to distinguish restrictions from five other faults including refrigerant undercharge. Refrigerant restrictions are identified as a possible cause of two problems: 1) head pressure too low, and 2) suction pressure too low. These problems are caused by the following faults: 1) dirty air filter, 2) insufficient evaporator airflow, 3) thermostat set to low, 4) compressor valves leaking, 5) restriction in liquid tube/filter drier/metering device, and 6) refrigerant undercharge. Technicians can easily check and correct dirty air filter, insufficient evaporator airflow, and thermostat set too low. If the system simultaneously has low discharge pressure and high suction pressure the cause is leaky compressor valves, worn compressor rings, or leaky oil separator. These problems require compressor replacement. The remaining faults are refrigerant undercharge or restriction in liquid tube/ filter drier/metering device. Undercharge can be ruled out by the fact that C1 had suction pressure 30% lower than C2 and C1 EST was below freezing and 2 times lower than C2 EST at 95F OAT (EST was 21F and C2 was 42F). If the liquid line temperature 12 to 24 inches upstream of the TXV entrance is 2 to 3F colder than ambient air, then there is a restriction upstream. If the temperature drop across the filter drier is greater than 3F, then there is a filter drier restriction. **Table 68** shows the C2 filter-drier ΔT is -1F (inlet minus outlet temperature) and the C1 restriction caused a 15F filter-drier ΔT. Clearly, C1 was restricted and not undercharged. The manufacturer's remedy is "recover refrigerant, remove restriction or replace filter drier, evacuate to 500 microns Hg hold for 20 minutes at or below 1000 microns Hg, and weigh in new refrigerant to factory charge."

4.2.11 Non-Condensable Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of non-condensables (NC) on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed, economizer perimeter unsealed, and airflow of 360 scfm/ton.[119] Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In order to emulate non-condensable air and water vapor faults, nitrogen was alternately added to each circuit in the amount of 0.5 ounces per test with a total of 1 ounce added per circuit or 2 ounce total for the two-circuit system. The weight of nitrogen is normalized with respect to the factory charge (oz/oz) so 0.5 ounces represents 0.25% of the total factory charge for both circuits. The unsealed economizer closed damper outdoor airflow rate for all baseline and non-condensable tests was 12.1%.

Non-condensable test results are provided in **Table 69**. Diagnostic test results are provided for manufacturer refrigerant charge protocols (discharge pressure ΔDP, suction pressure ΔSP, and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 32** shows the application efficiency impacts versus amount of non-condensables added to the 7.5-ton TXV RTU2. Non-condensables reduced sensible EER* by 9 to 22%, total EER* by 11 to 25%, sensible cooling capacity by 4 to 7%, and increased total power by 6 to 20%. The manufacturer ΔDP, ΔSP, and ΔSH refrigerant charge protocols correctly diagnosed proper charge for the C1 and C2 for the baseline test. For the non-condensable tests, the manufacturer ΔDP misdiagnosed C1 and C2 as overcharged and ΔSH misdiagnosed C1 and C2 as undercharged while ΔSP diagnosed both circuits as properly charged. Non-condensables cause conflicting refrigerant charge diagnostics. With 0.25 to 1% Nitrogen in one or more circuits, the manufacturer refrigerant charge protocol misdiagnoses non-condensables as an overcharge based on high discharge pressure, correct charge based on suction pressure, and undercharge based high superheat with respect to

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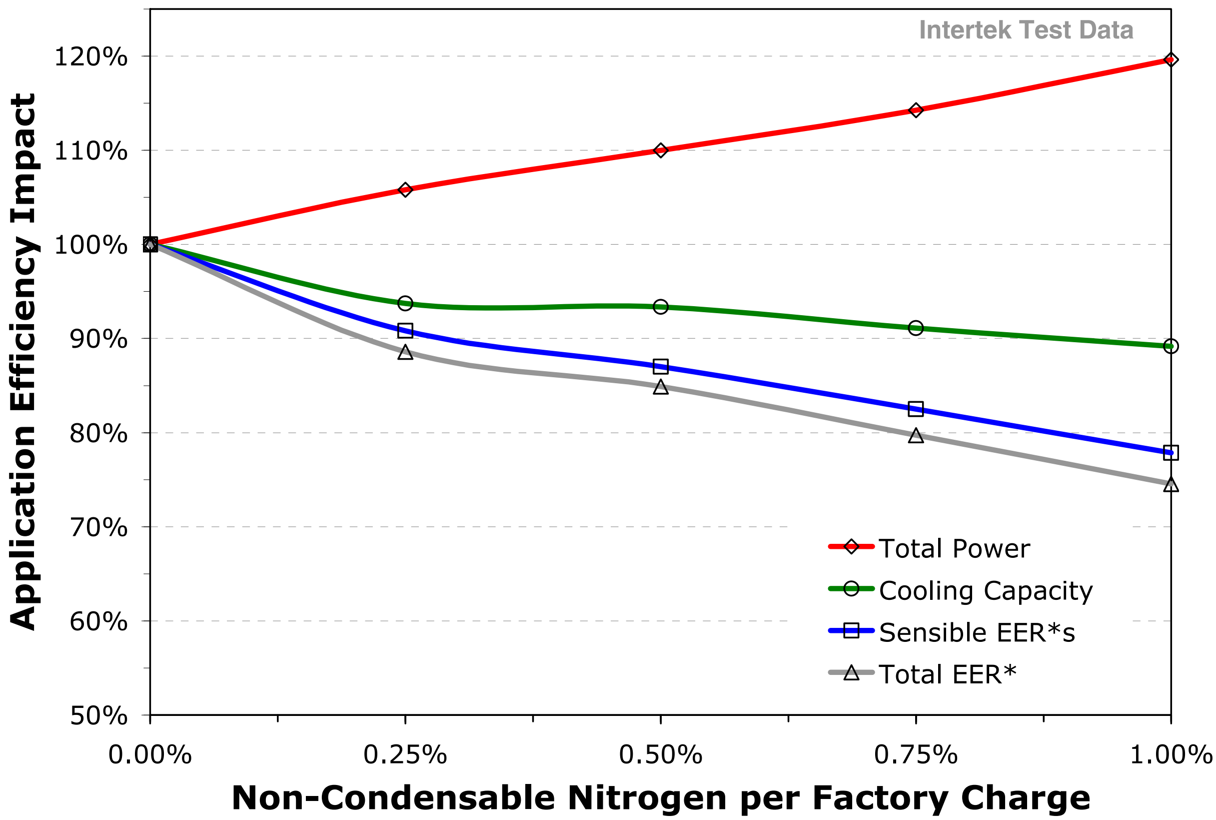
[119] If proper vacuum is not achieved at installation or after being opened for repair, the refrigerant system will be contaminated with non-condensable air and water vapor which can mix with refrigerant oils causing sludge, which can lead to compressor failure. Non-condensables (NC) decrease condenser heat transfer and cooling capacity and increase condenser pressure and compressor power input.

recommended target values. The CEC ΔTS protocols correctly diagnosed all tests with proper airflow and sensible cooling capacity highlighted in green. Non-condensable faults only reduce capacity by 4 to 7% which is insufficient for the CEC ΔTS protocols to diagnose capacity faults. The sensible cooling capacities for all tests are at least 105% of the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green. These tests show how the TXV expansion device can maintain consistent sensible cooling capacity with non-condensables.

Table 69: Non-Condensable Nitrogen Fault Impacts for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed, Factory Charge and 360 scfm/ton at 95F OAT

Test	NC %	Mfr C1/C2 ΔDP	Mfr C1/C2 ΔSP	Mfr C1/C2 ΔSH	CEC ΔTS	Dschg Press C1/C2 psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-MFB3-100C1-100C2-90A-95-CE-3	0.00	-5/-5	-5/-4	5/5	-0.3	248/249	8,244	8.1	57,202	6.9
T2-NC1-CK1-100C1-100C2-90A-CE	0.25	40/-11	-4/-4	21/5	-1.0	293/243	8,721	7.2	54,956	6.3
T2-NC2-CK1-CK2-100C1-100C2-90A-CE	0.50	36/25	-4/1	21/8	-1.1	290/280	9,067	6.9	54,743	6.0
T2-NC3-CK1-100C1-100C2-90A-CE	0.75	65/24	-2/1	20/8	-1.4	318/279	9,419	6.5	53,915	5.7
T2-NC4-CK1-CK2-100C1-100C2-90A-CE	1.00	62/66	-3/5	21/9	-1.6	316/320	9,863	6.1	53,274	5.4

Figure 32: Application Efficiency Impacts versus Non-Condensable Nitrogen for 7.5-ton TXV RTU2 with Economizer #1 Damper Closed, Factory Charge and 360 scfm/ton at 95F



This manufacturer does not provide "troubleshooting" procedures to diagnose non-condensables. Procedures discussed for the 7.5-ton non-TXV unit can be used to diagnose non-condensables from other faults (see **Section 4.1.12**).

4.2.12 Multiple Fault Tests for 7.5-ton TXV RTU2

Laboratory tests were performed to evaluate the impact of multiple faults on the application efficiency (EER*) of RTU2 with economizer #1 installed, dampers closed and 2-fingers open, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. Tests were performed with 85 to 115% refrigerant charge per factory charge, 65 to 110% airflow, 0 to 30% condenser blockage, C1 restriction, and dampers closed or 2-fingers (6V or 50%) open. The outdoor airflow was 12.1% with dampers closed and 31.1% with dampers 2-fingers open at 400 scfm/ton airflow (see **Table 55**). The predicted application efficiency impacts for multiple faults are calculated using **Equation 15**.

The predicted versus measured impacts for multiple faults are shown in **Table 70**. Predicted impacts for multiple faults are calculated using **Equation 15** based on measured single faults impacts. For refrigerant charge (85% and 115%) and condenser coil blockage (30%) the differences between measured and predicted application efficiency impacts are only 1 to 3% (with 12% OAF closed damper). For refrigerant charge (85 to 115%), airflow (65 to 110%), condenser coil blockage (30%), circuit 1 restriction (28% suction pressure reduction), and excess outdoor air (31.1% 2-fingers open) the average difference between measured and predicted application efficiency impacts is 0.3%. For the combined faults of condenser coil blockage plus C1 restriction the difference is larger due to condenser blockage causing increased discharge pressure which forces more refrigerant through the circuit 1 restriction.

Table 70: Measured versus Predicted Multiple Fault Impacts for 7.5-ton TXV RTU2 with Economizer Damper Closed (12% OAF) and 2-Fingers Open (31% OAF), 85-115% Factory Charge, 65-110% Airflow, 0-30% Condenser Blockage, C1 Restriction at 95F OAT

Test	Charge Impact ε_i	Airflow Impact ε_i	Cond Block Impact ε_i	Restrict C1 Impact ε_i	Excess OA Impact ε_i	Predicted Sensible EER*s	Measured Sensible EER*s	Difference Measured vs Predicted $\Delta\varepsilon$ %
T2-MFB3-100C1-100C2-90A-95-CE-2	0.000	0.000	0.000	0.000	0.000	6.96	6.96	0.0%
T2-MF33-85C1-85C2-100A-95-CE	0.012	0.000	0.000	0.000	0.000	6.88	6.74	-2.1%
T2-MFB1-100C1-100C2-100A-95-CE	0.000	0.000	0.000	0.000	0.000	6.96	6.92	-0.6%
T2-MF31-115C1-115C2-100A-95-CE	-0.004	0.000	0.000	0.000	0.000	6.99	6.90	-1.2%
T2-MF15-85C1-85C2-100A-95-CCE	0.012	0.000	0.102	0.000	0.000	6.17	6.17	0.0%
T2-MF0-100C1-100C2-100A-95-CCE	0.000	0.000	0.102	0.000	0.000	6.25	6.26	0.2%
T2-MF13-115C1-115C2-100A-95-CCE	-0.004	0.000	0.102	0.000	0.000	6.28	6.27	-0.1%
T2-MF37-85C1-85C2-90A-95-CE	0.012	-0.006	0.000	0.000	0.000	6.92	6.76	-2.3%
T2-MFB3-100C1-100C2-90A-95-CE-2	0.000	-0.006	0.000	0.000	0.000	7.00	6.96	-0.6%
T2-MF35-115C1-115C2-90A-95-CE	-0.004	-0.006	0.000	0.000	0.000	7.03	6.95	-1.1%
T2-MF19-85C1-85C2-90A-95-CCE	0.012	-0.006	0.102	0.000	0.000	6.21	6.15	-1.0%
T2-MF1-100C1-100C2-90A-95-CCE	0.000	-0.006	0.102	0.000	0.000	6.29	6.22	-1.2%
T2-MF17-115C1-115C2-90A-95-CCE	-0.004	-0.006	0.102	0.000	0.000	6.32	6.21	-1.8%
T2-MF24-85C1-85C2-63A-95-CE	0.012	0.032	0.000	0.000	0.000	6.66	6.60	-0.9%
T2-MF38-100C1-100C2-63A-95-CE	0.000	0.032	0.000	0.000	0.000	6.74	6.70	-0.6%
T2-MF21-115C1-115C2-63A-95-CE	-0.004	0.032	0.000	0.000	0.000	6.77	6.68	-1.2%
T2-MF4-85C1-85C2-63A-95-CCE	0.012	0.032	0.102	0.000	0.000	5.95	6.01	1.1%
T2-MF2-100C1-100C2-63A-95-CCE	0.000	0.032	0.102	0.000	0.000	6.03	6.08	0.8%
T2-MF6-115C1-115C2-63A-95-CCE	-0.004	0.032	0.102	0.000	0.000	6.06	6.04	-0.3%
T2-MF29-85C1-85C2-110A-95-CE	0.012	0.028	0.000	0.000	0.000	6.69	6.41	-4.3%
T2-MF39-100C1-100C2-110A-95-CE	0.000	0.028	0.000	0.000	0.000	6.77	6.73	-0.6%
T2-MF26-115C1-115C2-110A-95-CE	-0.004	0.028	0.000	0.000	0.000	6.80	6.65	-2.2%
T2-MF11-85C1-85C2-110A-95-CCE	0.012	0.028	0.102	0.000	0.000	5.98	5.90	-1.2%
T2-MF7-100C1-100C2-110A-95-CCE	0.000	0.028	0.102	0.000	0.000	6.06	6.01	-0.8%
T2-MF8-115C1-115C2-110A-95-CCE	-0.004	0.028	0.102	0.000	0.000	6.08	6.04	-0.8%
T2-MFR10-85C1-85C2-90A-95-CC2F	0.012	0.000	0.102	0.087	0.149	4.53	4.92	8.0%
T2-MFR8-100C1-100C2-90A-95-CC2F	0.000	0.000	0.102	0.087	0.149	4.61	5.21	11.5%
T2-MFR12-115C1-115C2-90A-95-CC2F	-0.004	0.000	0.102	0.087	0.149	4.64	5.33	13.0%
Average								**0.3%**

Multiple fault test results are provided in **Table 71**. Diagnostic test results are provided for manufacturer refrigerant charge protocols (ΔDP, ΔSP, and ΔSH) and CEC ΔTS protocols. The manufacturer protocols correctly diagnosed undercharge for multiple fault tests with 85% factory charge. The manufacturer protocols misdiagnosed multiple fault tests with 30% condenser blockage as being undercharged. Except for the 65% airflow and 30% condenser blockage tests, the manufacturer protocols correctly diagnosed 100% factory charge with 100 to 115% airflow. The CEC ΔTS properly diagnosed low airflow for six tests with 65% airflow highlighted in yellow. The CEC ΔTS misdiagnosed three tests with 85 to 115% factory charge, 110% airflow, and 30% condenser blockage with low capacity highlighted in yellow. For all other tests with 90 to 100% airflow, 85 to 115% factory charge, and 0 to 30% condenser blockage the CEC ΔTS correctly diagnosed sensible cooling capacity. The sensible cooling capacities for all tests are greater than 105% of 45,024 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 71: Multiple Fault Impacts for 7.5-ton TXV RTU2 with Economizer Damper Closed, 85-115% Factory Charge, 65-110% Airflow, and 0-30% Condenser Blockage at 95F OAT

Test	C1/C2 Charge %	Airflow scfm/ton per 400 %	Cond Block Coil %	Mfr C1/C2 ΔDP	Mfr C1/C2 ΔSP	Mfr C1/C2 ΔSH	CEC Protocol ΔTS	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-MF33-85C1-85C2-100A-95-CE	85/85	100		-11/-8	-7/-2	22/18	-2.0	57,580	6.7
T2-MFB1-100C1-100C2-100A-95-CE	100/100	100		-5/-3	-3/-3	5/5	-1.3	59,712	6.9
T2-MF31-115C1-115C2-100A-95-CE	115/115	100		-2/9	-3/-3	4/3	-1.1	60,962	6.9
T2-MF15-85C1-85C2-100A-95-CCE	85/85	100	30	16/18	-6/-2	21/18	-2.4	56,270	6.2
T2-MF0-100C1-100C2-100A-95-CCE	100/100	100	30	25/30	-2/-2	6/6	-1.7	58,645	6.3
T2-MF13-115C1-115C2-100A-95-CCE	115/115	100	30	29/43	-3/-2	4/3	-1.3	59,967	6.3
T2-MF37-85C1-85C2-90A-95-CE	85/85	90		-11/-9	-8/-4	23/20	-1.0	55,559	6.8
T2-MFB2-100C1-100C2-90A-95-CE	100/100	90		-4/-2	-4/-4	5/5	-0.2	57,713	6.9
T2-MF35-115C1-115C2-90A-95-CE	115/115	90		-3/7	-5/-4	4/4	0.1	58,963	6.9
T2-MF19-85C1-85C2-90A-95-CCE	85/85	90	30	18/19	-6/-3	21/19	-1.2	54,422	6.2
T2-MF1-100C1-100C2-90A-95-CCE	100/100	90	30	26/31	-3/-3	6/6	-0.5	56,552	6.2
T2-MF17-115C1-115C2-90A-95-CCE	115/115	90	30	31/44	-4/-3	4/4	-0.2	57,718	6.2
T2-MF24-85C1-85C2-63A-95-CE	85/85	65	0	-14/-14	-10/-9	20/19	3.2	49,876	6.6
T2-MF38-100C1-100C2-63A-95-CE	100/100	65	0	-9/-7	-9/-10	6/7	3.9	51,586	6.7
T2-MF21-115C1-115C2-63A-95-CE	115/115	65	0	-8/1	-9/-10	4/4	4.1	52,203	6.7
T2-MF4-85C1-85C2-63A-95-CCE	85/85	65	30	13/14	-9/-8	20/20	3.0	49,191	6.0
T2-MF2-100C1-100C2-63A-95-CCE	100/100	65	30	20/24	-8/-9	6/8	3.7	50,992	6.1
T2-MF6-115C1-115C2-63A-95-CCE	115/115	65	30	24/37	-9/-9	4/5	4.2	51,917	6.0
T2-MF29-85C1-85C2-110A-95-CE	85/85	110	0	-11/-8	-7/-1	23/18	-3.6	58,331	6.4
T2-MF39-100C1-100C2-110A-95-CE	100/100	110	0	-3/0	-2/-1	7/7	-2.7	62,291	6.7
T2-MF26-115C1-115C2-110A-95-CE	115/115	110	0	-2/10	-2/-1	4/3	-2.5	62,478	6.6
T2-MF11-85C1-85C2-110A-95-CCE	85/85	110	30	18/21	-5/1	22/19	-4.0	57,252	5.9
T2-MF7-100C1-100C2-110A-95-CCE	100/100	110	30	27/32	0/0	7/7	-3.4	59,682	6.0
T2-MF8-115C1-115C2-110A-95-CCE	115/115	110	30	32/45	-1/0	4/4	-3.0	61,256	6.0

Figure 33 shows the measured and predicted application efficiency impacts versus refrigerant charge per factory charge with airflow from 65 to 115% and 0 to 30% condenser blockage (CB). The predicted impacts (black lines) for 65% airflow and 30% condenser blockage are -0.2 to 0.6% lower than measured impacts. Predicted impacts for 110% airflow and 30% condenser blockage are -0.4 to 0.4% lower than measured. EER*s was greatest at 90 to 100% of rated airflow, 100% factory charge and no condenser blockage. Efficiency decreased by 3 to 5% with 85% factory charge and 65% airflow. Efficiency decreased by an additional 7.8 to 10.4% with 30% blocked condenser coil.

Figure 33: Measured and Predicted Multiple Fault Impacts for 7.5-ton TXV RTU2 with Economizer Damper Closed, 85-115% Factory Charge, 65-110% Airflow, and 0-30% Condenser Blockage at 95F OAT

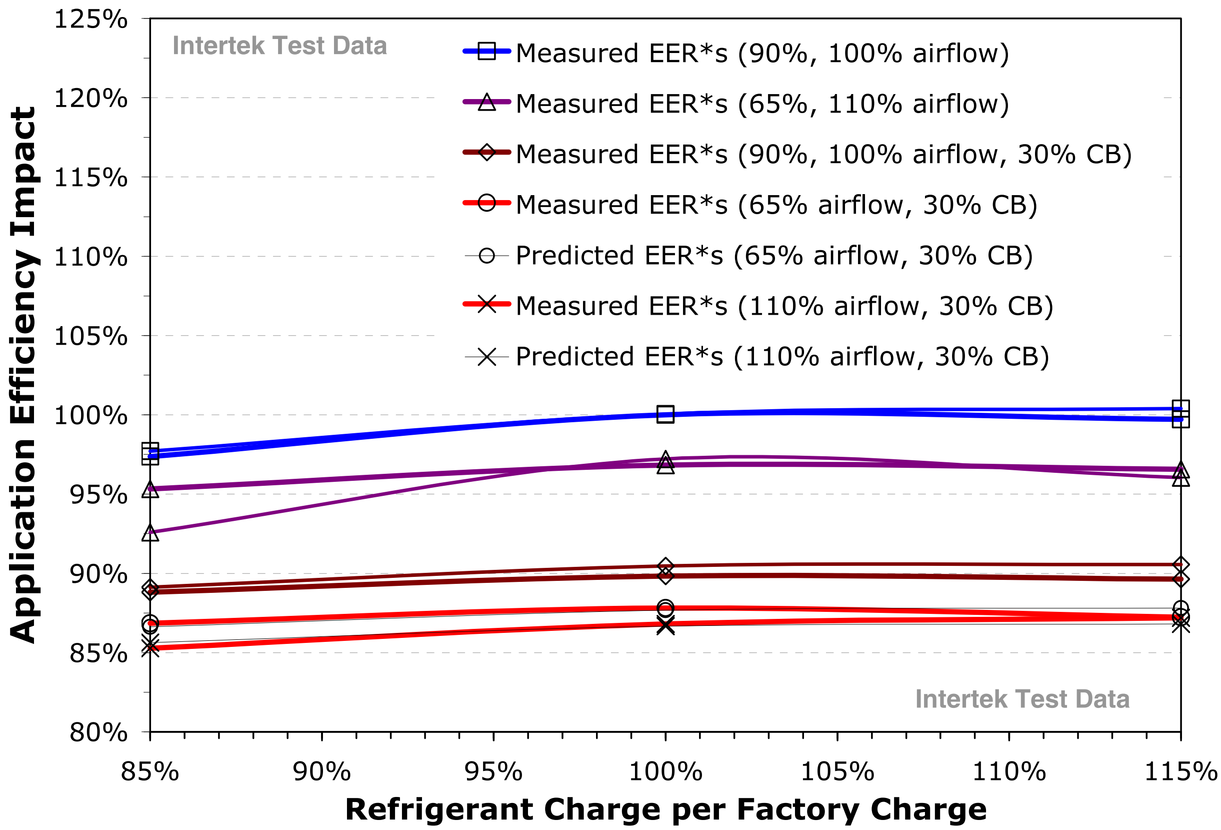


Table 72 provides multiple fault test results for RTU2 with a circuit 1 restriction and economizer damper 2-fingers open causing excess outdoor air (31.1% OAF versus 12.1% OAF with closed dampers). Manufacturer refrigerant charge protocol FDD results are also provided. **Figure 34** shows the measured and predicted application efficiency impacts versus refrigerant charge per factory charge with 0 to 30% condenser blockage (CB), C1 restriction, and economizer damper closed or 2-fingers open. The predicted impacts (black lines) for condenser blockage are 0.5 to 0.9% higher than measured impacts. Predicted impacts for C1 restriction and economizer damper 2-fingers open are 4.6 to 9% lower than measured. These tests indicate that increased discharge

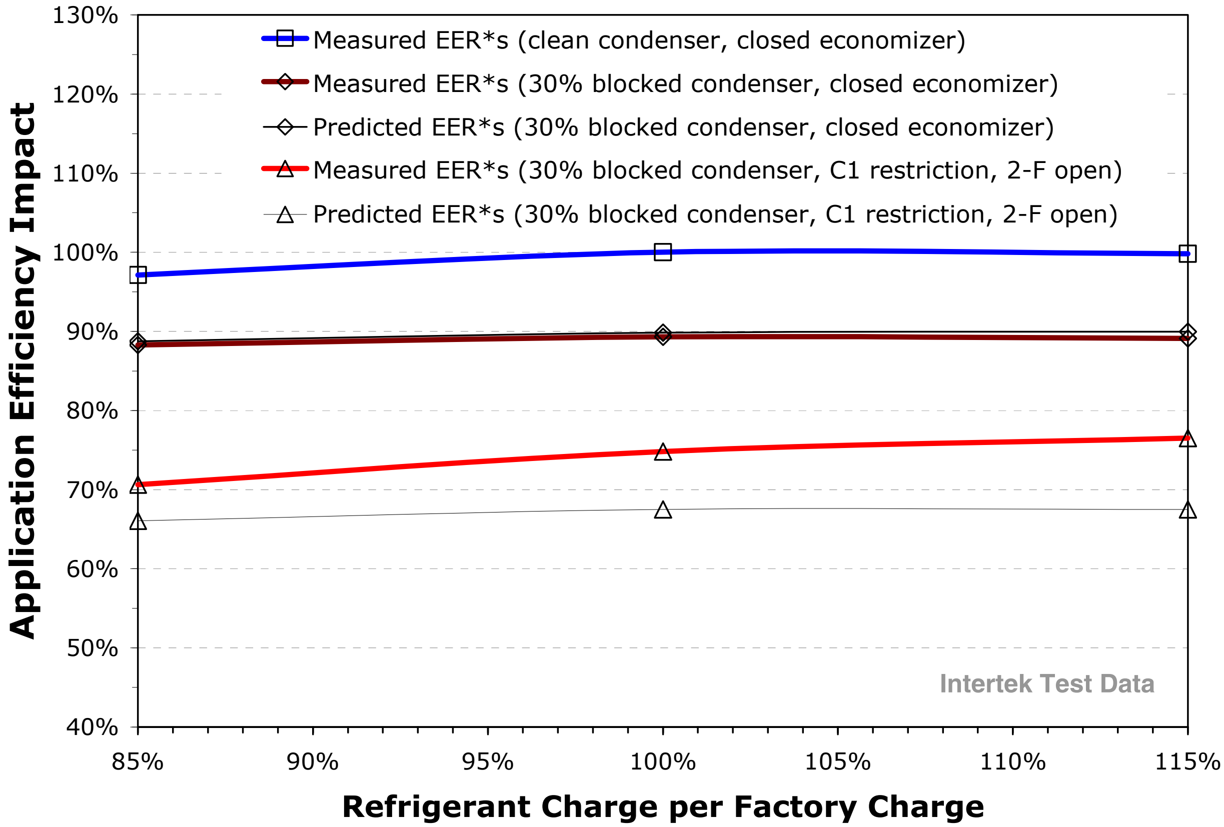
pressure from condenser blockage can mitigate a restriction. However, cleaning the condenser will cause the full impact of the restriction fault to reoccur. The application sensible efficiency (EER*s) is maximized at 100% factory charge and no coil blockage. EER*s decreased by 1.4% with 85% factory charge and decreased by 10 to 12% with 30% condenser blockage and by 23 to 29% with 30% condenser blockage plus circuit 1 restriction plus economizer damper 2-fingers open.

The manufacturer protocols correctly diagnosed proper charge at 100% factory charge with no other faults, but diagnosed C1, C2 or both as undercharged at 85% factory charge. The manufacturer protocols misdiagnosed 115% factory charge as correct but efficiency and capacity were within 5% of non-faulted values. With 30% condenser blockage, the manufacturer protocols diagnosed C1, C2 or both as undercharged, overcharged, or correctly charged at 85 to 115% factory charge. With 30% condenser blockage plus circuit 1 restriction plus economizer damper 2-fingers open, the manufacturer protocols misdiagnosed C1, C2 or both as undercharged, overcharged, or properly charged. These results indicate the importance of following a logical sequence of maintenance procedures before performing FDD. The CEC ΔTS protocols properly diagnosed sensible cooling capacity for all tests. The multiple fault tests with C1 restriction plus 30% condenser coil blockage plus 2-fingers open dampers have sensible cooling capacities 10% less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red or yellow. The other multiple fault test sensible cooling capacities are at least 105% of ACCA Manual N and highlighted in green.

Table 72: Multiple Fault Impacts for 7.5-ton TXV RTU2 with Economizer Dampers Closed or 2-Fingers Open, 85-115% Factory Charge, 0-30% Condenser Blockage, and C1 Restriction at 95F OAT

Test	C1/C2 Charge %	C1 Restrict 2-F Open	Blocked Cond Coil %	Mfr C1/C2 ΔDP	Mfr C1/C2 ΔSP	Mfr C1/C2 ΔSH	CEC Protocol ΔTS	Sensible Cooling Capacity Btuh	Sensible EER*s
T2-MF37-85C1-85C2-90A-95-CE	85/85			-11/-9	-8/-4	23/20	-1.0	55,559	6.8
T2-MFB2-100C1-100C2-90A-95-CE	100/100			-4/-2	-4/-4	5/5	-0.2	57,713	6.9
T2-MF35-115C1-115C2-90A-95-CE	115/115			-3/7	-5/-4	4/4	0.1	55,559	6.9
T2-MF19-85C1-85C2-90A-95-CCE	85/85		30	18/19	-6/-3	21/19	-1.2	54,422	6.2
T2-MF1-100C1-100C2-90A-95-CCE	100/100		30	26/31	-3/-3	6/6	-0.5	56,552	6.2
T2-MF17-115C1-115C2-90A-95-CCE	115/115		30	31/44	-4/-3	4/4	-0.2	57,718	6.2
T2-MFR10-85C1-85C2-90A-95-CC2F	85/85	Yes	30	-3/12	-31/0	37/1	-5.5	42,188	4.9
T2-MFR8-100C1-100C2-90A-95-CC2F	100/100	Yes	30	10/24	-22/0	30/0	-4.4	46,104	5.2
T2-MFR12-115C1-115C2-90A-95-CC2F	115/115	Yes	30	17/38	-17/-1	26/0	-3.6	48,340	5.3

Figure 34: Measured and Predicted Multiple Fault Application Efficiency Impacts for 7.5-ton TXV RTU2 with Economizer Dampers Closed or 2-Fingers Open, 85-115% Factory Charge, 0-30% Condenser Blockage, and C1 Restriction at 95F OAT



Troubleshooting multiple faults using manufacturer procedures will reduce or eliminate “false alarms,” misdetection, and misdiagnosis. For example, multiple faults such as 85% refrigerant charge plus 30% condenser blockage plus C1 restriction plus 2-fingers open dampers does not cause the C1 discharge pressure or C2 suction pressure and superheat to be outside tolerances (test T2-MFR10-85C1-85C2-90A-95-CC2F). However, C1 discharge pressure and C2 suction pressure and superheat are significantly outside manufacturer tolerances and indicate either undercharge or overcharge. Technicians can visually diagnose and correct condenser coil blockage. Diagnosing undercharge from restriction can be performed by measuring the temperature drop across the filter drier. The circuit 1 filter drier temperature drop is 24F indicating a restriction while the circuit 2 filter drier temperature increases by 1F indicating no restriction. Refrigerant must be recovered from circuit 1 to replace the restricted filter drier. Circuit 1 must be evacuated to 500 microns held at or below 1000 microns for 30 minutes and circuit 1 factory charge weighed into the unit. Circuit 2 can be diagnosed and corrected by technician by adding charge to within manufacturer tolerances.

4.3 Test Results for 3-ton Non-TXV Packaged HVAC RTU5

One 3-ton multiple fixed-orifice non-TXV packaged HVAC model (RTU5) was tested in the laboratory per the ANSI/AHRI 210/240 test procedure. RTU5 uses R22 refrigerant and was shipped with a 0.75 horsepower (HP) blower motor, forward-curved centrifugal blower wheel with 1” wide blades and 10” diameter x 10” width. RTU5 has one compressor and is equipped with multiple fixed-orifice expansion valves on the header of the evaporator inlet. The unit was shipped from the factory with motor sheave set to 3 turns out from the maximum fan speed setting, 4.5 inch diameter fan pulley. The manufacturer installation, start-up, and service instructions indicate that the motor sheave is typically set from the factory to 3 turns to provide 890 revolutions per minute (RPM) fan speed.[120] The unit was tested in the horizontal configuration.

4.3.1 Out-of-Box Tests for 3-ton non-TXV RTU5

The 3-ton non-TXV RTU5 was tested in the “out-of-the-box” as-purchased condition in the horizontal configuration. **Table 73** provides the out-of-box tests for RTU5 with 0.15 IWC ESP and 95F outdoor conditions, and 80F DB and 67F WB indoor conditions. The initial tests were performed at 919 to 945 rpm with the motor sheave at 3 turns, but the airflow was 33% greater than the 450 scfm/ton maximum airflow specified in ANSI/AHRI 210/240. The first two EER* tests were 6% to 9% less than the rated 11.2 EER (C-ONE and C-ONE-OB). The motor sheave was adjusted to 5.5 turns to achieve 756 rpm and approximately 1,200 scfm at 0.15 IWC ESP per ANSI/AHRI Standard 210/240. The cabinet panel joints were sealed with tape to reduce outdoor air leakage for AHRI tests. The third EER* test was 4% less than the rated 11.2 EER (C-ONE-5TA-OB). Intertek technicians recovered and weighed the refrigerant charge from the unit and found 5.3 lbs or 4% less than the manufacturer factory charge of 5.5 lbs. The unit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the 5.5 lbs factory charge. For test C-22A-ONE with factory charge, the total EER* was 10.9 and only 3% less than the published AHRI 11.2 EER rating. The cooling capacity was 35,648 Btuh or 1% less than the published 36,000 Btu/hr AHRI rating.

[120] Carrier 2003. 48HJ004-007 Single-Package Rooftop Heating/Cooling Standard and Low NOx Units. Installation, Start-up, and Service Instructions. Form 48HJ-22SI. Fig. 56 – Cooling Charging Charts. http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48hj-22si.pdf.

Table 73: Out-of-Box Tests for 3-ton non-TXV RTU5 with 4.5” Diameter Blower Pulley and without Economizer

Test	Charge %	Fan Turn	Fan RPM	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER*	Tested EER*	ΔEER*	Tested Sensible EER*s
C-ONE	96	3.0	945	0.15	600	480	3,510	36,746	11.2	10.5	-6%	8.5
C-ONE-OB	96	3.0	919	0.15	603	468	3,442	34,950	11.2	10.2	-9%	8.3
C-ONE-5TA-OB	96	5.5	756	0.15	431	293	3,248	34,801	11.2	10.7	-4%	7.9
C-22A-ONE	100	5.5	757	0.15	430	296	3,274	35,648	11.2	10.9	-3%	8.0

4.3.2 AHRI Verification Tests for 3-ton non-TXV RTU5

The ANSI/AHRI Standard 210/240 verification tests were performed to evaluate rated performance and provide the DEER DMQC team with part-load cycling data for developing building energy simulations. **Table 74** provides ANSI/AHRI EER and SEER verification cycling test data for the 3-ton non-TXV unit (RTU5). The measured EER at 95F OAT was 10.97 and the measured EER at 82F OAT was 12.96. The average SEER was 12.33 and the average cyclic degradation coefficient (C_d) was 0.113.[121] The EER was within 2% of the published AHRI rating, the cooling capacity was within 0.04% of the published AHRI rating, and the SEER was within 95% of the published AHRI rating. All measured values were within ANSI/AHRI tolerances.[122]

[121] Cycling degradation (CD) coefficient measures the efficiency loss due to cycling of units as determined in Appendices C and D of ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment Standard 210/240. Air-Conditioning Heating and Refrigeration Institute.

[122] ANSI/AHRI STANDARD 210/240-2008, “6.5 Tolerances. To comply with this standard, measured test results shall not be less than 95% of Published Ratings for performance ratios and capacities.”

Table 74: ANSI/AHRI EER and SEER Verification and Cycling Tests for 3-ton non-TXV RTU5 without Economizer

Test	ESP IWC	Airflow scfm/ ton	Rated Cooling Capacity Btuh	Rated EER	Rated SEER	Total Cooling Capacity Btuh	Tested EER	Tested SEER	C_d
C-22A	0.15	428	36,000	11.2	13	35,987	10.97		
C-22B	0.16	430	39,750	14	13	37,323	12.96		
C-22C-D #2								12.38	0.186
C-22C-D #3								12.36	0.092
C-22C-D #4								12.31	0.101
C-22C-D #5								12.35	0.094
C-22C-D #6								12.31	0.101
C-22C-D #7								12.27	0.106
Average								**12.33**	**0.113**

SEER is calculated from EER_B and cycling tests used to measure C_d per **Equation 19**.[123]

Equation 19 $$SEER = EER_B \times (1 - 0.5 \times C_d)$$

Where,

EER_B = energy efficiency rating at 82F OAT and indoor conditions of 80F DB and 67F WB (Btuh/Watt)

C_d = cyclic degradation coefficient which is the lower of tested value or default of 0.25. If measured C_d is less than zero, then C_d is set to zero (dimensionless)

C_d is based on cyclic EER_C and EER_D tests performed at 82F OAT and dry-coil indoor conditions of 80F DB and 57F WB.

4.3.3 Manufacturer Refrigerant Charge Diagnostics for 3-ton non-TXV RTU5

The unit-specific manufacturer refrigerant charge diagnostic protocols for the 3-ton non-TXV unit are based on suction temperature (ST) as a function of outdoor drybulb (DB) temperature (i.e., condenser entering air) and suction pressure (SP).[124] The manufacturer refrigerant charge

123 Per ANSI/AHRI STANDARD 210/240-2008.

124 Carrier 2003. 48HJ004-007 Single-Package Rooftop Heating/Cooling Standard and Low NOx Units. Installation, Start-up, and Service Instructions. Form 48HJ-22SI. Fig. 46 – Cooling Charging Charts. http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/48hj-22si.pdf.

ST tolerances are +/-5F.[125] The manufacturer does not provide superheat target values, airflow diagnostic protocols, or liquid pressure ports so subcooling cannot be evaluated. The CEC superheat (ΔSH) protocols are used to evaluate refrigerant charge and temperature split (ΔTS) are used to evaluate airflow and sensible cooling capacity faults based on test results for the 3-ton non-TXV unit.[126] For information about the CEC protocols see **Section 2.1.3**. The laboratory tests provide sensible cooling minus ventilation loads which are included in the measurements. Sensible cooling capacity test results are diagnosed using ACCA Manual N sensible cooling load design values minus ventilation loads provided in **Table 2** and described in **Section 2.2.1**.

Table 75 shows manufacturer and CEC refrigerant charge and airflow diagnostics for RTU5. For the 96% factory charge test (C-ONE-5TA-OB), the manufacturer ΔST and CEC ΔSH protocols diagnosed undercharge correctly highlighted in red. For 100% charge tests (C-22A-ONE, C-22A, C-22B) the manufacturer and CEC protocols misdiagnose “false alarm” undercharge, but EER and cooling capacity are within published AHRI ratings. The CEC ΔTS diagnosed proper airflow and cooling capacity highlighted in green. The sensible cooling capacities for all tests are at least 105% of the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 75: Manufacturer and CEC Refrigerant Charge and Airflow Diagnostics for 3-ton non-TXV RTU5 without Economizer at 95F and 82F Outdoor Temperature

Test	Charge %	OAT (F)	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Fan Power W	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-ONE-5TA-OB	96	95	15	15	-1.2	431	293	3,248	34,801	10.7	25,697	7.9
C-22A-ONE	100	95	11	12	-0.8	430	296	3,274	35,648	10.9	26,268	8.0
C-22A	100	95	10	11	-0.7	428	292	3,279	35,987	11.0	26,354	8.0
C-22B	100	82	13	8	-0.4	430	293	2,880	37,323	13.0	26,890	9.3

Table 76 shows manufacturer and CEC refrigerant charge and airflow diagnostics for five tests performed on the 3-ton non-TXV unit with economizer installed, damper closed, 367 to 372 scfm/ton airflow at 95F OAT and 75F DB and 62F WB return air temperature. Refrigerant charge for the five tests ranges from 60 to 140% of factory charge. With factory charge (C-95-3-FC-C), the EER* was 6.4 or 42% less than published the AHRI 11 rating and the EER*s was 6.0. The EER* is lower due to typical static pressure and economizer installed with closed damper

[125] Manufacturer refrigerant charge protocol for undercharge: ΔST > 5F. Manufacturer refrigerant charge protocol for overcharge: ΔST <-5F. Manufacturer protocol for correct charge: -5F ≤ ΔST ≤ 5F.

[126] California Energy Commission (CEC). 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. RA3.2 Procedures for Determining Refrigerant Charge for Split System Space Cooling Systems Without a Charge Indicator Display. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

which provides 23.5% OAF. The 10% overcharge test C-95-3+10-C, total EER* was 6.4 and sensible EER*s was 5.9 or 2% less. The 40% overcharge test C-95-3+40-C total EER* was 6.0 or 6% less than factory charge and sensible EER*s was 5.8 or 4% less than factory charge. The 10% undercharge test C-95-3-10-C, total EER* was 6.2 or 3% less than factory charge and sensible EER*s was 5.9 or 2% less. The 40% undercharge test C-95-3-40-C, total EER* was 4.3 or 33% less than factory charge and sensible EER*s was 4.6 or 23% less than factory charge.

The manufacturer ΔST protocol correctly diagnoses over- and undercharge tests, but provides a "false alarm" undercharge for 100% factory charge. The CEC ΔSH correctly diagnoses the 60 and 90% undercharge tests, provides a "false alarm" undercharge for factory charge, and incorrectly diagnoses the 110 and 140% overcharge tests as correct when they are not. The CEC ΔTS protocols correctly diagnose the 10% overcharge test with proper airflow and capacity highlighted in green. The CEC ΔTS correctly diagnoses all other tests as low capacity (ΔTS < -3F) highlighted in yellow. The 10% overcharge sensible cooling is correct since the total cooling capacity is greater than 105% of the 19,097 Btuh ACCA Manual N sensible cooling load and is highlighted in green. The 60% charge test sensible cooling capacity is less than the 19,097 Btuh ACCA Manual N sensible cooling load and is highlighted in red. All other tests have sensible cooling capacities greater than ACCA Manual N highlighted in yellow.

Table 76: Manufacturer and CEC Refrigerant Charge and Airflow Diagnostics for 3-ton non-TXV RTU5 with Economizer #5 and Closed Damper at 95F Outdoor Temperature

Test	Charge %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ Ton	Fan Power W	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-95-3-40-C	60	46	31	-8.0	379	372	3,112	13,412	4.3	14,371	4.6
C-95-3-10-C	90	15	16	-3.8	373	369	3,282	20,478	6.2	19,376	5.9
C-95-3-FC-C	100	7	10	-3.4	372	370	3,336	21,288	6.4	19,903	6.0
C-95-3+10-C	110	-8	-3	-3.0	368	367	3,405	21,774	6.4	20,231	5.9
C-95-3+40-C	140	-9	-4	-3.3	367	367	3,426	20,707	6.0	19,797	5.8

4.3.4 Economizer Outdoor Airflow Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate outdoor airflow, overventilation, and unintended outdoor air leakage on the 3-ton non-TXV RTU5 with economizer #5 installed. Tests were performed with unsealed economizer perimeter and with tape to seal around the perimeter of the economizer where it attaches to the cabinet. Tests were performed with the evaporator fan blower motor on, both compressors off, outdoor conditions of 55F and indoor conditions of 75F DB and 62F WB. The outdoor, return, and mixed air drybulb temperatures were measured using resistance temperature detector (RTD) sensors in the outdoor, return, and supply air samplers. The outdoor air temperature entering the economizer was also measured using an array of 6 thermocouple sensors installed in the economizer inlet. The volumetric flow rate of air was measured using the Code Tester.[127] For tests with blower fan and compressors operating, the mixed air temperature entering the evaporator was measured with an array of 22 shielded-drybulb temperature sensors located on the air filter inlet adjacent to the evaporator.

Table 77 provides calculated outdoor air fractions at 55F OAT using **Equation 6** and **Equation 9** with no compressors operating and at 95F using **Equation 1** with compressors operating (equations are described in **Section 4.1.4**). The difference was -1.3 +/- 0.4% between using **Equation 6** at 55F OAT and using **Equation 9** at 55F OAT. The difference was -1.4 +/- 3.1% between using **Equation 6** at 55F and no compressors operating and **Equation 1** at 95F with compressor operating. With no economizer installed the outdoor air leakage varied from 2 to 8.4%. With perimeter unsealed and dampers from closed to fully open, the OAF ranged from 23.5 to 68% depending on OAT.

Table 77: Economizer #5 Outdoor Air Fractions Calculated using Equations 6 and 9 at 55F OAT without Compressors and Equation 1 at 95F OAT with Compressors for RTU5

Description	Test	Evap Airflow scfm/ton	Equation 6 Calc OAF at 55F OAF_e %	Equation 9 Calc OAF at 55F OAF_t %	Test	Evap Airflow scfm/ton	Equation 1 Calc OAF at 95F OAF_m %
No Economizer	C-55-NE	339	7.1	8.4	C-MF-75629575-NE3J	359	2.0
Closed	C-55-CE-DM	385	23.5	24.6	C-MF-75629575-E3J	372	23.6
1F (5.1V)	C-55-1ER-DM	381	32.6	33.5	C-MF-75629575-1E3J	370	39.3
2F (6V)	C-55-2ER-DM	388	40.0	40.8	C-MF-75629575-2E3J	373	42.8
3F (6.9V)	C-55-3ER-DM	386	52.4	54.3	C-MF-75629575-3E3J	378	54.2
100% Open (10V)	C-55-OER-DM	372	66.3	68.0	C-MF-75629575-OE3J	363	61.8

Figure 35 shows unsealed economizer #5 installed on RTU5. **Figure 36** shows the same economizer with tape to seal unintended perimeter leakage. Tests were performed with tape to seal the economizer perimeter. In the field, UL-181 waterproof tape should be used to seal the perimeter. These tests were performed to measure the impact of reducing unintended outdoor air leakage. All other sealed perimeter tests only have tape on the economizer perimeter where it connects to the unit. Technicians generally use their fingers to set damper positions where 1-finger is assumed to be open 10%, 2-fingers 20% and 3-fingers 30%. Using fingers to set minimum damper positions causes variations in the opening depending on finger size and placement with respect to the damper and frame. Finger diameters are as follows: 1-Finger is 0.7 inch (1.8 cm), 2-fingers is 1.3 inches (3.3 cm), and 3-fingers is 2 inches (5.1 cm).

Table 78 provides calculated OAF for economizer #5 using **Equation 6** for the 3-ton non-TXV unit at 55F OAT with unsealed and sealed perimeter. With no economizer the OAF was 7.1%. With perimeter unsealed and dampers from closed to fully open, the OAF ranged from 23.5 to

[127] The "code tester" is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92). Standard Methods for Laboratory Airflow Measurement.

66.3% at 55F OAT. Sealing the perimeter (under the hood) reduced unintended outdoor air leakage by 1.8 to 9.5% with more reduction when dampers are closed or 1-finger open.

Figure 35: Unsealed Economizer Perimeter for 3-ton non-TXV RTU5 (wires showing through gap)

Figure 36: Sealed Economizer for 3-ton non-TXV RTU5 with Tape around Perimeter

Table 78: Economizer #5 Outdoor Air Fractions Calculated using Equation 6 at 55F OAT and ~375 scfm/ton for 3-ton non-TXV RTU5 with Unsealed and Sealed Perimeter

Description	Unsealed Test	Evap Airflow scfm/ton	Equation 6 Unsealed Calc OAF_e at 55F %	Sealed Test	Evap Airflow scfm/ton	Eq. 6 Sealed Perimeter Calc OAF_e at 55F %	ΔOAF %
No Economizer	C-55-NE	339	7.1	C-55-NES	330	4.1	3.0
Closed (2.0V)	C-55-CE-DM	385	23.5	C-55-TCE-DM	366	14.0	9.5
1F (5.1V)	C-55-1ER-DM	381	32.6	C-55-T1ER-DM	383	26.6	6.0
2F (6V)	C-55-2ER-DM	388	40.0	C-55-T2ER-DM	384	35.0	5.0
3F (6.9V)	C-55-3ER-DM	386	52.4	C-55-T3ER-DM	379	50.6	1.8
100% Open (10V)	C-55-OER-DM	372	66.3	C-55-TOER-DM	374	65.8	0.5

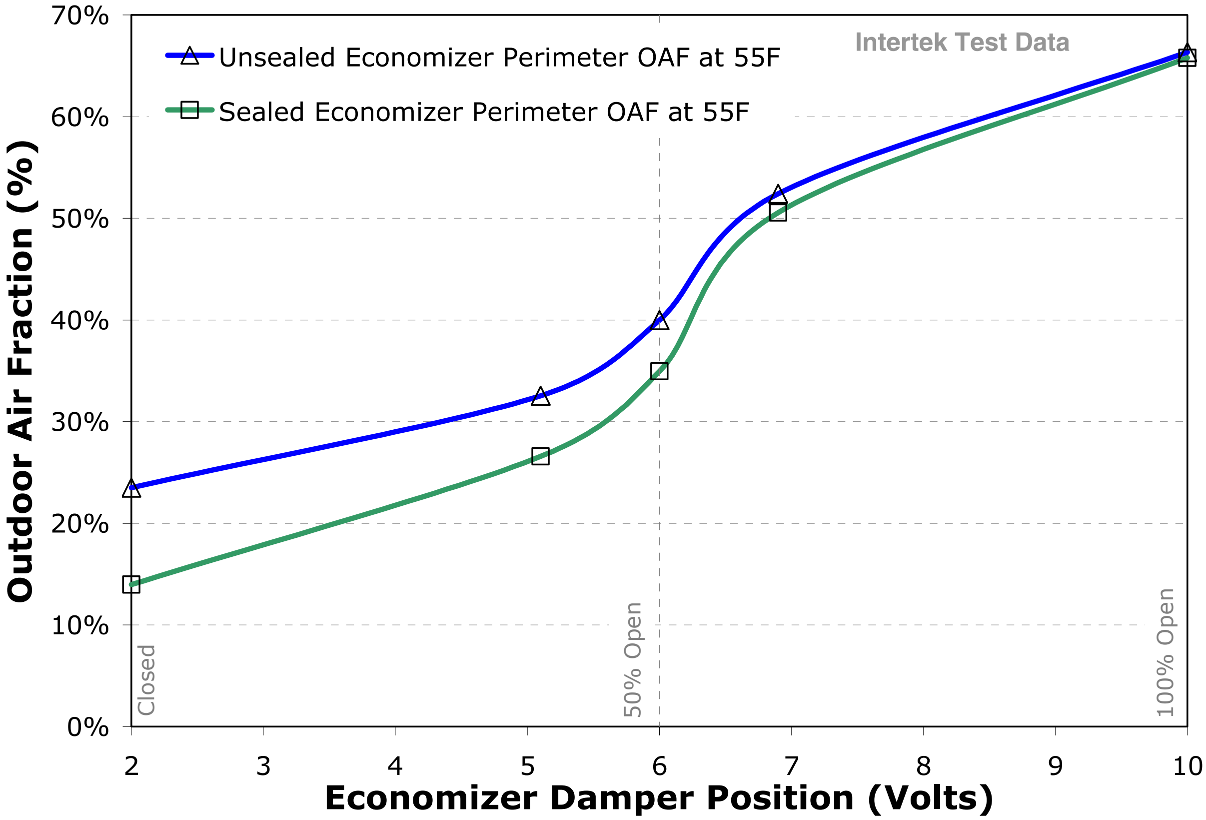
Table 79 provides calculated outdoor air fractions using **Equation 6** at 55F OAT for RTU5 with Economizer #5 installed, 0 to 50% evaporator coil blockage and unsealed economizer perimeter. Evaporator coil blockage reduced airflow by 1 to 9% and the OAF by 3 to 6%.

Table 79: Second Tests – Economizer #5 Outdoor Air Fractions Calculated using Equation 6 at 55F for the 3-ton non-TXV RTU5 with 0 to 50% Evaporator Coil Blockage

Description	Unsealed Test	Equation 6 Calc OAF_e at 55F %	OAF_e Impact %	Evap Airflow scfm/ton	Evap Airflow Impact %
Closed 0% Evaporator Blockage	C-55-NE	28.1	0%	365.6	0%
Closed 5% Evaporator Blockage	C-EB5-55-CE	27.4	-3%	361.5	-1%
Closed 10% Evaporator Blockage	C-EB10-55-CE	26.9	-4%	360.7	-1%
Closed 20% Evaporator Blockage	C-EB20-55-CE	26.6	-5%	355.6	-3%
Closed 35% Evaporator Blockage	C-EB35-55-CE	26.6	-6%	345.8	-5%
Closed 50% Evaporator Blockage	C-EB50-55-CE	26.9	-5%	331.1	-9%

Figure 37 shows the outdoor air fraction versus damper position with unsealed and sealed perimeter for RTU5 with economizer #5. The average difference between unsealed and sealed is 4.6 +/- 2.6%. Sealing the perimeter (under the hood) reduced unintended outdoor airflow and improved cooling and heating efficiency. Reducing overventilation and establishing the most efficient minimum damper position is important for health, comfort, and energy efficiency. Outdoor airflow provided by each economizer varies and most manufacturers do not provide accurate outdoor airflow data as a function of damper position.

Figure 37: Outdoor Air Fraction versus Economizer #5 Damper Position for 3-ton non-TXV RTU5 with Unsealed and Sealed Perimeter at 375 scfm/ton Total Airflow



4.3.5 Economizer 95F Efficiency Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate the application efficiency impact of economizer outdoor airflow for the 3-ton non-TXV RTU5 with economizer #5 installed and outdoor air dampers closed, partially open, and 100% open and the economizer perimeter unsealed and sealed with tape. Tests were performed with factory charge and outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. **Table 80** provides economizer #5 outdoor air ventilation impacts versus damper position and 360 scfm/ton airflow at 95F OAT. At 100% open dampers the ESP was 0.4 IWC and with closed dampers the ESP was 0.6 IWC. At 100% open position the ESP was 0.41 IWC. ISP ranged from -0.03 (100% open) to -0.18 IWC (closed). With no economizer installed the total EER* was 9.2 and the sensible EER*s was 7.2. With economizer #5 installed and dampers closed the total EER* was 6.4 and sensible EER*s was 6.0. The outdoor airflow (OA) ventilation load can have a significant impact on cooling and heating efficiency especially when the minimum damper position is more open than necessary to meet the ASHRAE 62.1 minimum ventilation requirement. The reduction in efficiency with economizer #5 installed and closed dampers was 31% for total EER* and 17% for sensible EER*s. With closed dampers the economizer #5 efficiency was 42% less than the AHRI EER rating of 11.0 and 26% less than the sensible EER at the AHRI test conditions. Opening economizer dampers per the outdoor air leakage tests performed at 55F in the previous section significantly reduced efficiency. The minimum tested application sensible efficiency was 3.1 EER*s for unsealed and 3.2 EER*s for sealed perimeter which are 61% less than the AHRI rating and 57% less than the efficiency with no economizer installed. The -0.1 unsealed total EER* was negative due to outdoor airflow supplying more latent load than the evaporator can remove. If a building requires 15% outdoor air per ASHRAE 62.1, then economizer #5 would provide 14% OAF with closed damper perimeter sealed and 23.5% OAF with closed damper perimeter unsealed. If a technician set the minimum damper position at 2-fingers open, economizer #5 would provide 40% OAF or 2 times more outdoor ventilation than closed. The overventilation at 2-fingers open (unsealed perimeter) would reduce EER*s to 4.9 EER*s or 18% less efficient than 6 EER*s at closed position with unsealed perimeter. The 2-finger open minimum damper position would reduce total efficiency to 3.9 EER* or 39% less than 6.4 total EER* with closed damper. If the building required 20% OA, then technicians could close the damper to reduce OAF from 40 to 23.5% and increase sensible EER*s from 4.9 to 6.0 and save 18.3%. Providing adequate outdoor ventilation air is as important as providing comfortable indoor temperature control. The reduction in efficiency due to overventilation beyond minimum requirements represents an important energy efficiency opportunity for space cooling and heating.

Table 80: Economizer #5 Outdoor Air Ventilation Impacts versus Damper Position Unsealed and Sealed Perimeter for 3-ton non-TXV RTU5 and 360 scfm/ton at 95F

Description	Test	Unsealed Total EER*	Unsealed Sensible EER*s	Test	Sealed Perimeter Total EER*	Sealed Perimeter Sensible EER*s
No Economizer	C-MF-75629575-NE3J	9.2	7.2	NA		
Closed	C-MF-75629575-E3J	6.4	6.0	C-75629575-TE3	7.9	6.6
1F (5.1V)	C-MF-75629575-1E3J	5.0	5.3	C-75629575-T1E3	6.1	5.8
2F (6V)	C-MF-75629575-2E3J	3.9	4.9	C-75629575-T2E3-1	4.6	5.2
3F (6.9V)	C-MF-75629575-3E3J	2.0	4.1	C-75629575-T3E3	2.3	4.2
100% Open (10V)	C-MF-75629575-OE3J	-0.1	3.1	C-75629575-TOE3	0.1	3.2

Table 81 provides economizer #5 outdoor air ventilation impacts and FDD versus damper position with sealed and unsealed perimeter frame and 360 to 370 scfm/ton airflow at 95F OAT. With 100% open dampers, the ESP was 0.4 and with dampers closed the ESP was 0.58 IWC. Opening the dampers from closed to 1–finger reduces EER*s by 12%. Closed position provides 23.5% outdoor air and increases capacity and efficiency by 11 to 48%. Closed damper with sealed economizer perimeter provides 14% outdoor air and improves EER*s to 6.6 EER* or 10% compared to 6.0 EER* unsealed perimeter.

For all tests except sealed perimeter 100% open, the manufacture ΔST and CEC ΔSH protocols misdiagnosed “false alarm” undercharge. The CEC ΔTS protocol indicates low cooling capacity with negative temperature split from -3.9 to -11F.[128] The CEC ΔTS protocol correctly diagnoses low sensible cooling capacity for 89% of tests highlighted in red or yellow. Excess outdoor air causes inadequate cooling capacity. The remedy is to adjust minimum damper position to provide just enough outdoor to meet ASHRAE 62.1 ventilation requirements. Tests with no economizer installed, closed dampers with or without sealed perimeter frame, and 1-finger open with sealed perimeter frame have sensible cooling capacities greater than the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in green or yellow. All other tests do not meet ACCA Manual N due to overventilation highlighted in red.

Table 81: Economizer #5 Outdoor Air Ventilation Impacts and FDD versus Damper Position Unsealed and Sealed Perimeter for 3-ton non-TXV RTU5 at 360 scfm/ton at 95F

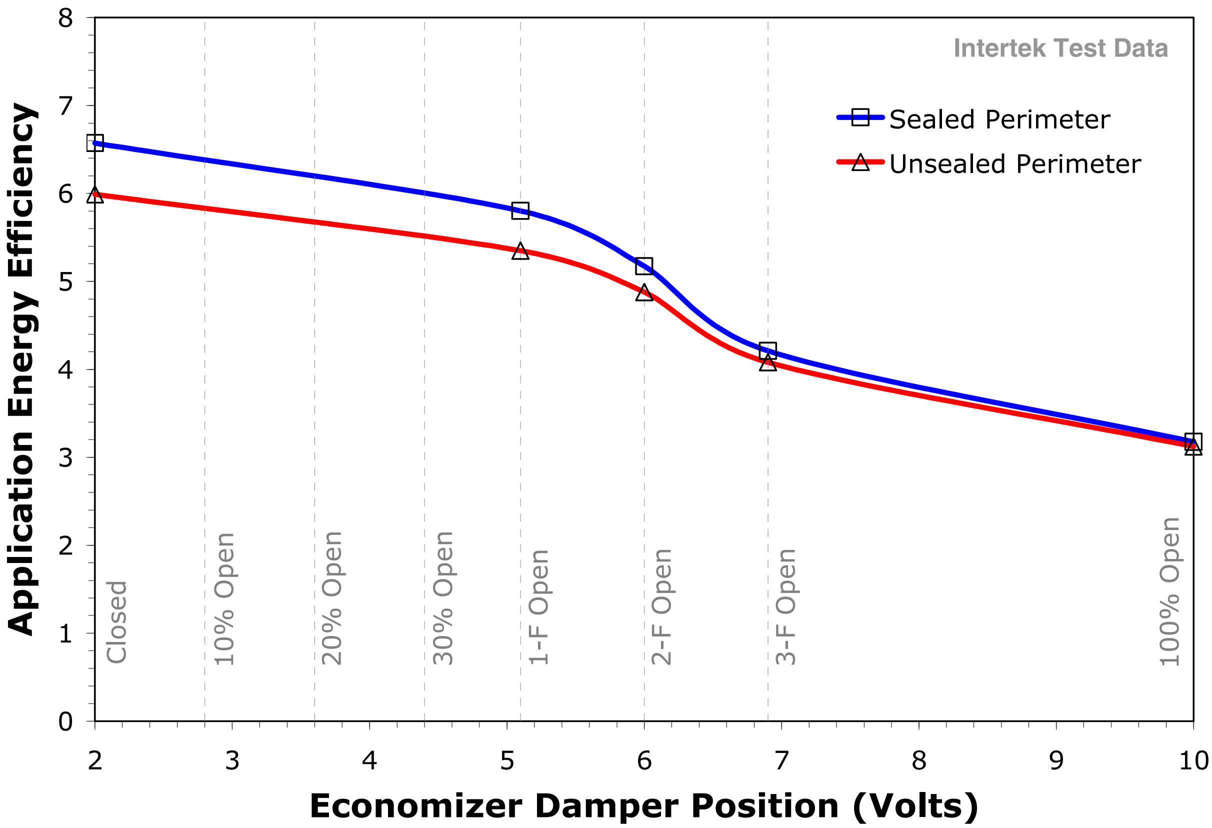
Description	Test	OAF %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
No Economizer	C-MF-75629575-NE3J	4.5	8.4	12.0	0.2	3,288	30,234	9.2	23,656	7.2
Unsealed										
Closed (2.0V)	C-MF-75629575-E3J	23.5	9.6	12.0	-3.4	3,326	21,176	6.4	19,926	6.0
1F (5.1V)	C-MF-75629575-1E3J	32.6	9.5	12.0	-4.9	3,338	16,677	5.0	17,856	5.3
2F (6V)	C-MF-75629575-2E3J	40.0	9.9	11.0	-6.2	3,343	13,014	3.9	16,314	4.9
3F (6.9V)	C-MF-75629575-3E3J	52.4	10.0	9.0	-8.3	3,350	6,681	2.0	13,667	4.1
100% Open (10V)	C-MF-75629575-OE3J	66.3	8.9	5.0	-10.5	3,342	-223	-0.1	10,435	3.1
Sealed										
Closed (2.0V)	C-75629575-TE3	14.0	8.9	12.0	-1.5	3,310	25,995	7.9	21,752	6.6
1F (5.1V)	C-75629575-T1E3	26.6	9.5	12.0	-3.5	3,326	20,427	6.1	19,299	5.8
2F (6V)	C-75629575-T2E3-1	35.0	9.9	11.0	-5.3	3,330	15,419	4.6	17,224	5.2
3F (6.9V)	C-75629575-T3E3	50.6	10.1	9.0	-7.9	3,346	7,822	2.3	14,084	4.2
100% Open (10V)	C-75629575-TOE3	65.8	8.7	4.0	-10.2	3,332	470	0.1	10,582	3.2

Figure 38 shows the decline in the application sensible energy efficiency for the 3-ton non-TXV RTU5 as dampers are opened from fully closed (2V) to fully open (10V) with economizer #5 installed. The mandated outdoor ventilation rates for most building occupancies range from 6 to 10% for offices, 22% for retail, 33% for auditoriums and schools, 40% for restaurants and health clubs, and 53% or more for cafeterias and sports arenas. Ventilation rates for unoccupied spaces can be minimized to save energy. While outdoor ventilation is a system load, most of the load could be avoided if the optimal minimum damper position is established. Field observations found approximately 50% of units with economizers not working properly or dampers stuck 10

[128] Temperature split tests are based on well-mixed return and supply drybulb and wetbulb temperatures ignoring outdoor air mixing with return air which increases the evaporator inlet drybulb and wetbulb temperatures which are difficult for technicians to accurately measure in the field.

to 100% open with Molex plugs or other objects stuck between damper blades. If a commercial building space required 14% OAF and the dampers were set at 1-finger open, the building would receive 32.6% OA. Sealing the economizer perimeter and closing the damper would reduce OAF from 32.6 to 14% or 57%, and increase EER*s from 5.3 to 6.0 EER*s and save 19.7%.

Figure 38: Application Sensible Energy Efficiency versus Damper Position for 3-ton non-TXV RTU5 with Economizer #5 Sealed and Unsealed Perimeter at 95F OAT



Overventilation and unintended outdoor airflow are common maintenance faults on all commercial buildings. Reducing overventilation can have a significant impact on thermal comfort, HVAC efficiency, and energy use. The manufacturer provides tables of information for "troubleshooting" faults with the cooling and heating system. Too much outdoor air is identified as a possible cause of inadequate heating. The manufacturer lists the following faults for inadequate heating: 1) dirty air filter, 2) gas input to unit too low, 3) unit undersized, 4) restricted airflow, 5) blower speed too low, 6) limit-switch causes main burners to cycle, and 7) too much outdoor air. Technicians can check and correct dirty air filters, restricted airflow, and blower speed (fan belt tension/alignment, pulley and motor sheave). Technicians can also check gas input pressure and burner limit switch. Too much outdoor air can be checked and corrected by adjusting the economizer minimum damper position.

4.3.6 Refrigerant Charge Fault Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate the impact of refrigerant charge faults on the application energy efficiency of RTU5 with economizer #5 installed and perimeter unsealed, dampers closed, and airflow at 375 scfm/ton. Tests were performed at outdoor temperatures of 95F and return temperatures of 75F DB and 62F WB with factory charge varying from 60 to 140% of factory charge. With economizer #5 dampers closed the outdoor airflow was 23.5% and with dampers 1-finger open the outdoor airflow was 32.6% per **Table 77**. Refrigerant charge was added or removed in increments of 10% of the factory charge for each test. Preliminary measurements were performed without code tester installed for each test setup in order to match total static pressure with the code tester installed.

Refrigerant charge fault test results for RTU5 are provided in **Table 82**. Diagnostic test results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 39** shows the application energy efficiency impacts versus refrigerant charge per factory charge for RTU5 with the economizer damper closed at 95F OAT. Total EER* and EER*s are maximized at 100% to 110% factory charge. Undercharging refrigerant by 10 to 40% reduced EER*s by 0 to 28% and total application efficiency by 2 to 32%. Overcharging by 10 to 40% reduced EER*s by 0 to 3% and total application efficiency by 0 to 5%. **Equation 20** is the regression equation shown in **Figure 39**. **Equation 20** and **Equation 11** can be used to calculate EER*s impacts associated with refrigerant charge adjustments based on recovery and weigh-out of refrigerant charge and reported charge adjustment for the 3-ton non-TXV RTU5.

Equation 20 $$y = -1.9622x^5 + 9.4303x^4 - 16.26x^3 + 11.077x^2 - 1.2851x$$

Where,

y = EER*s impact at refrigerant charge per factory charge ratio (dimensionless)

x = refrigerant charge per factory charge ratio (dimensionless)

With the economizer installed and dampers closed or 1-finger open, the manufacturer ST protocol correctly diagnosed 10 to 40% undercharge and 10 to 40% overcharge. With dampers closed, the CEC SH protocols correctly diagnosed 10 to 40% undercharge. With dampers 1-finger open, the CEC protocols correctly diagnosed 10 to 40% undercharge and 20 to 40% overcharge. The CEC ΔTS protocols correctly diagnosed low capacity for all tests except 10% overcharge with closed dampers, due to 23.5% outdoor airflow with dampers closed and 32.6% with dampers 1-finger open. The manufacturer ST protocol accuracy was 89% and the CEC SH protocol accuracy was 61%. At 40% overcharge DP was 228 psig, and the high pressure cut-out is 428 psig and the reset is 320 psig.[129] All refrigerant charge tests with closed dampers and 90 to 140% factory charge have sensible cooling capacities greater than the 19,097 Btuh ACCA

[129] Carrier 2006. Product Data. WeatherMaster® 48HJ004-028 48HE003-006, Single-Package Rooftop Units, Gas Heating/Electric Cooling, 2 to 25 Nominal Tons. Page 17. http://dms.hvacpartners.com/docs/1009/Public/00/48H-1PD.pdf.

Manual N sensible cooling load highlighted in green or yellow. All other tests with closed dampers or 1-finger open dampers have sensible cooling capacities less than ACCA Manual N due to overventilation highlighted in red. The CEC ΔTS protocol correctly diagnosed low sensible cooling capacity for 94% of tests highlighted in red or yellow.

Table 82: Refrigerant Charge Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed and 1-Finger Open and 375 scfm/ton at 95F OAT

Test	Refrig Charge %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol ΔTS	Airflow scfm/ ton	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
Undercharge Closed										
C-95-3-40-C	60	46	31	-8.0	379	3,112	13,412	4.3	14,371	4.6
C-95-3-30-C	70	34	25	-6.0	376	3,166	16,241	5.1	16,722	5.3
C-95-3-20-C	80	24	20	-4.5	371	3,232	18,904	5.8	18,445	5.7
C-95-3-10-C	90	15	16	-3.8	373	3,282	20,478	6.2	19,376	5.9
Factory Charge Closed										
C-95-3-FC-C	100	7	10	-3.4	372	3,336	21,288	6.4	19,903	6.0
Overcharge Closed										
C-95-3+10-C	110	-8	-3	-3.0	368	3,405	21,774	6.4	20,231	5.9
C-95-3+20-C	120	-9	-4	-3.1	367	3,431	21,436	6.2	20,098	5.9
C-95-3+30-C	130	-9	-4	-3.2	366	3,420	21,040	6.2	19,903	5.8
C-95-3+40-C	140	-9	-4	-3.3	367	3,426	20,707	6.0	19,797	5.8
Undercharge 1-Finger										
C-95-3-40-1	60	34	25	-9.9	379	3,112	8,073	2.6	11,858	3.8
C-95-3-30-1	70	35	24	-7.9	380	3,173	11,184	3.5	14,475	4.6
C-95-3-20-1	80	24	20	-6.4	382	3,242	13,908	4.3	16,420	5.1
C-95-3-10-1	90	15	15	-5.4	376	3,296	15,888	4.8	17,538	5.3
Factory Charge 1-Finger										
C-95-3-FC-1	100	7	10	-5.1	373	3,352	16,373	4.9	17,780	5.3
Overcharge 1-Finger										
C-95-3+10-1	110	-10	-5	-4.4	375	3,428	17,616	5.1	18,714	5.5
C-95-3+20-1	120	-11	-6	-4.5	371	3,466	17,216	5.0	18,437	5.3
C-95-3+30-1	130	-11	-6	-4.6	371	3,465	16,802	4.8	18,262	5.3
C-95-3+40-1	140	-12	-6	-4.8	371	3,473	16,230	4.7	18,021	5.2

Figure 39: Application Efficiency Impacts versus Refrigerant Charge per Factory Charge for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed at 95F OAT

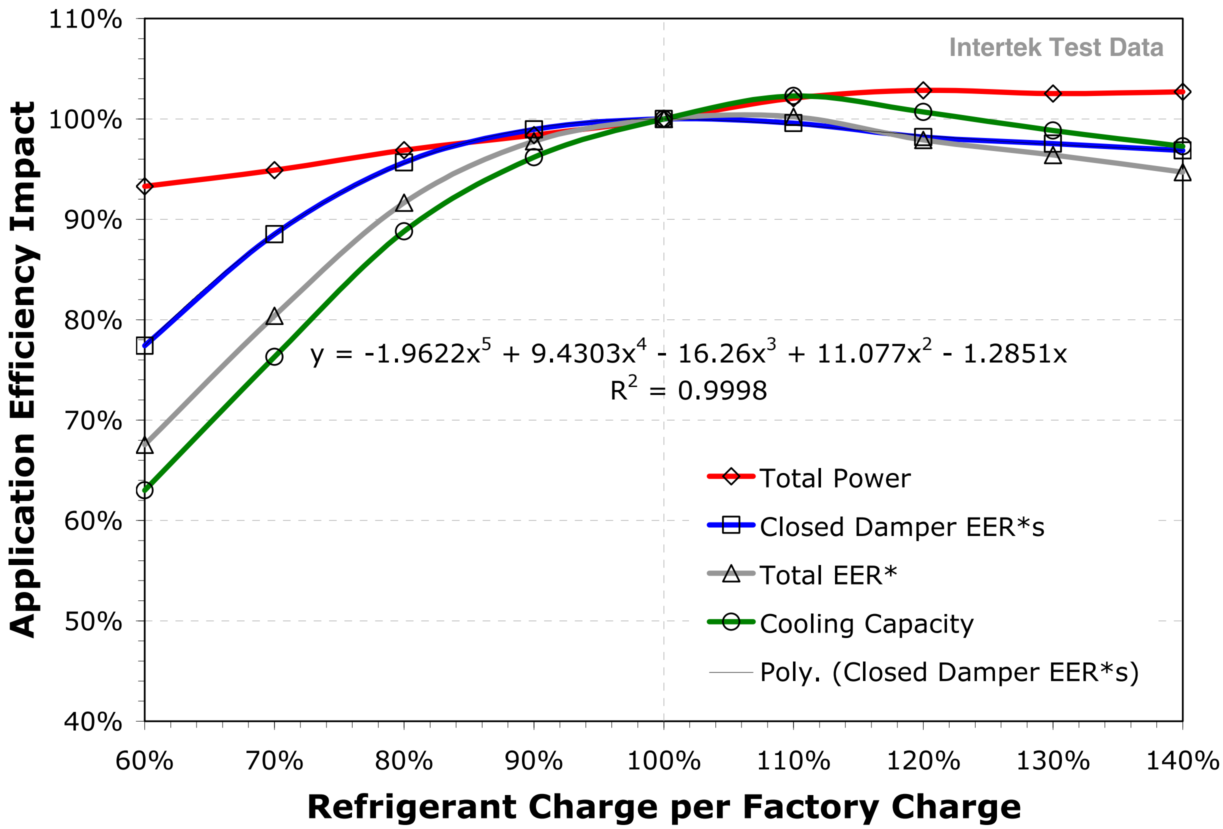
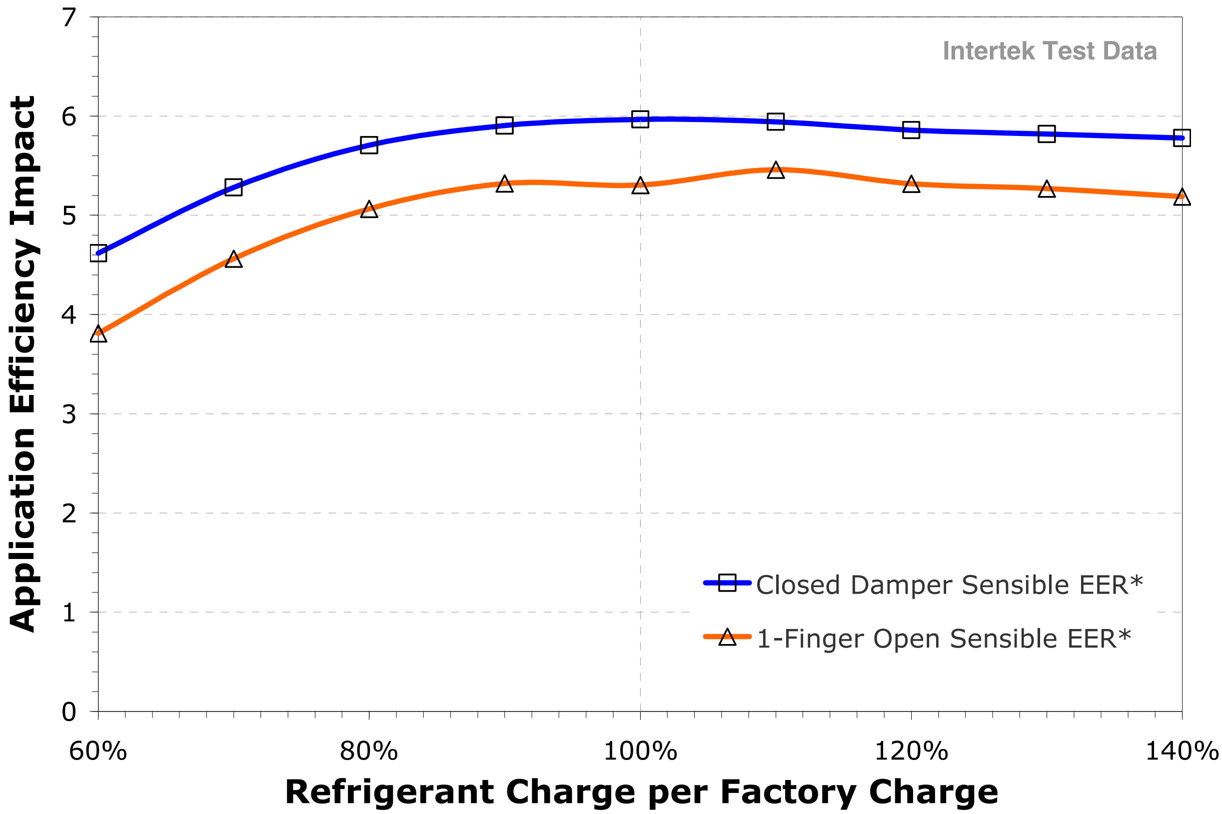


Figure 40 shows the application sensible efficiency impact versus refrigerant charge per factory charge for RTU5 with the economizer damper closed and 1-finger open at 95F OAT. Changing the economizer damper position from closed to 1-finger open increased outdoor airflow from 23.5 to 32.6% and reduced EER*s by 8 to 17% for an average reduction of 11.1 +/- 1.6%. As noted previously, unnecessary ventilation loads can be avoided if optimal minimum damper position is established or if unintended outdoor air leakage is reduced by sealing the economizer perimeter (under the hood) with UL-181 metal tape.

Figure 40: Application Sensible Efficiency Impacts versus Refrigerant Charge per Factory Charge for 3-ton non-TXV RTU5 with Economizer #5 Closed and 1-Finger Open at 95F OAT



Field observations of 35 units found an average difference between recovered and pre-existing refrigerant charge of 15.1 +/- 3.2% corresponding to an average EER*s impact of 8.9 +/- 3.9%. This is 1.9 times greater than the average EER*s impact of 4.6% that could be calculated based on the average difference between recovered (94.4%) and pre-existing charge (79.1%). The average of individual EER*s impacts is 1.9 times larger than the difference between the average recovered and pre-existing charge.

Severe undercharge causes icing of the evaporator coil as evaporator saturation temperature decreases below freezing causing water that condenses on the coil to freeze into ice. Coil icing reduces airflow which decreases efficiency even more. Icing of the coil was avoided while performing undercharge tests by operating unit with fan only (no compressors) in between tests, checking evaporator coil to make sure no ice was present, and condensate pan was dry. Overcharging can cause liquid refrigerant to flood the compressor during normal operation and start-up which dilutes oil causing inadequate bearing lubrication and premature failure.

The manufacturer provides information for “troubleshooting” and diagnosing refrigerant charge faults.[130] Procedures for troubleshooting and servicing air conditioning systems are also provided in technician training text books.[131] The most common problems are high or low discharge or suction pressure or continuous compressor operation. These problems are caused by a number of faults including: 1) dirty air filter, 2) blocked evaporator/condenser, 3) undersized unit or low cooling capacity or excessive outdoor air, 4) insufficient evaporator airflow, 5) refrigerant restriction, 6) non-condensables, 7) thermostat defective/set too low, 8) low line voltage (faulty contactor/transformer), 9) defective compressor/overload, or 10) refrigerant over/undercharge. Prior to adjusting refrigerant charge, technicians need to check and correct all other faults on the list. If none of the other faults are present and problem still exists, then refrigerant charge adjustments might be necessary.

4.3.7 Condenser Blockage Fault Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate the impact of condenser blockage faults on the application efficiency (EER*) of RTU5 with economizer #5 installed, dampers closed, economizer perimeter unsealed, and airflow of ~330 scfm/ton. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge and evaporator airflow of approximately 360 scfm//ton. The condenser coil was blocked on the outside of the coil with plastic corrugated cardboard used to ship condensers (to block but not damage fins). The test setup was based on field measurements of 29 units where dirty condensers were cleaned and the discharge pressure decreased by 1 to 28%. Condenser blockage test results at 95F OAT are provided in **Table 83**. Discharge pressure increases by 1.6 to 29.6% by installing 5 to 80% condenser coil blockage on the inlet side of the condenser coil. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 41** shows the application energy efficiency impacts versus discharge pressure increase due to condenser coil blockage at 95F OAT. Condenser coil blockage reduces EER*s by 1 to 25%, sensible capacity decreases by 0.2 to 8%, and total power increases by 1 to 23% with discharge pressure increase of 30%. **Equation 21** is the regression equation shown in **Figure 41**. **Equation 21** can be used to calculate the EER*s impacts based on the discharge pressure ratio increase at constant OAT due to condenser coil blockage for the 3-ton non-TXV unit.

Equation 21 $y_c = 0.86 x_p + 1.0$

Where,

y_c = EER*s impact of condenser coil blockage based on discharge pressure increase (dimensionless)

[130] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

[131] Tomczyk, J. 1995. Troubleshooting and Servicing Modern Air Conditioning and Refrigeration Systems. ESCO Press. Mt. Prospect, Ill.: Educational Standards Corporation.

x_p = discharge pressure increase due to condenser coil blockage (dimensionless)

Table 84 and **Table 85** provide condenser coil blockage test results at 115F and 82F OAT. Tests at 115F OAT are only provided for 5 to 50% condenser coil blockage due to the high pressure switch turning off the compressor at 60% coil blockage and above.[132] The manufacturer ΔST protocol misdiagnosed overcharge for all tests except 0 to 5% blockage at 95F OAT, no blockage at 115F, and 30 to 80% blockage at 82F OAT. The manufacturer ΔST protocol misdiagnosed undercharge for 0 to 5% blockage at 82F OAT. The CEC ΔSH protocol misdiagnosed overcharge for 0 to 10% blockage at 95F OAT and 40 to 80% blockage at 82F. The CEC ΔSH protocol also misdiagnosed undercharge for no blockage at 82F OAT.

The CEC ΔTS correctly diagnosed low capacity for all tests highlighted in yellow at 95F and 115F OAT. The CEC ΔTS correctly diagnosed low capacity for all tests highlighted in yellow at 95F and 115F OAT. The CEC ΔTS correctly diagnosed proper airflow and sensible capacity for all tests highlighted in green at 82F OAT. All tests at 95F OAT are less than the 45,024 Btuh ACCA Manual N sensible cooling load highlighted in red. All tests at 115F OAT are less than the 61,132 Btuh ACCA Manual N sensible cooling load highlighted in red. All tests at 82F OAT are greater than 105% of the 14,939 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 83: Condenser Blockage Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed and 360 scfm/ton at 95F

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-CB0-95-CE	0.0%	1.9	4.3	-3.9	361	273.4	3,539	5.37	18,411	5.20
C-CB5-95-CE	1.6%	-3.5	-0.1	-4.0	364	277.9	3,583	5.28	18,453	5.15
C-CB10-95-CE	3.7%	-8.5	-4.6	-3.9	363	283.5	3,632	5.23	18,584	5.12
C-CB20-95-CE	7.2%	-10.5	-5.4	-4.1	365	293.2	3,732	4.96	18,421	4.94
C-CB30-95-CE	10.7%	-11.1	-5.3	-4.2	364	302.7	3,832	4.70	18,207	4.75
C-CB40-95-CE	14.7%	-11.3	-5.3	-4.2	355	313.5	3,939	4.49	17,828	4.53
C-CB50-95-CE	18.2%	-12.7	-5.6	-4.5	364	323.3	4,036	4.20	17,793	4.41
C-CB60-95-CE	21.4%	-12.7	-5.2	-4.7	356	332.1	4,120	3.95	17,193	4.17
C-CB70-95-CE	25.5%	-14.1	-6.0	-4.8	360	343.2	4,234	3.73	17,233	4.07
C-CB80-95-CE	29.6%	-14.9	-5.6	-5.1	362	354.5	4,359	3.44	16,940	3.89

132 High-pressure switch opens at 428 psig and closes at 320 psig. Carrier 2006. Product Data. WeatherMaster® 48HJ004-028 48HE003-006, Single-Package Rooftop Units, Gas Heating/Electric Cooling, 2 to 25 Nominal Tons. Page 39. http://dms.hvacpartners.com/docs/1009/Public/00/48H-1PD.pdf.

Figure 41: Application Efficiency Impacts versus Discharge Pressure Increase due to Condenser Blockage for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed and 360 scfm/ton at 95F

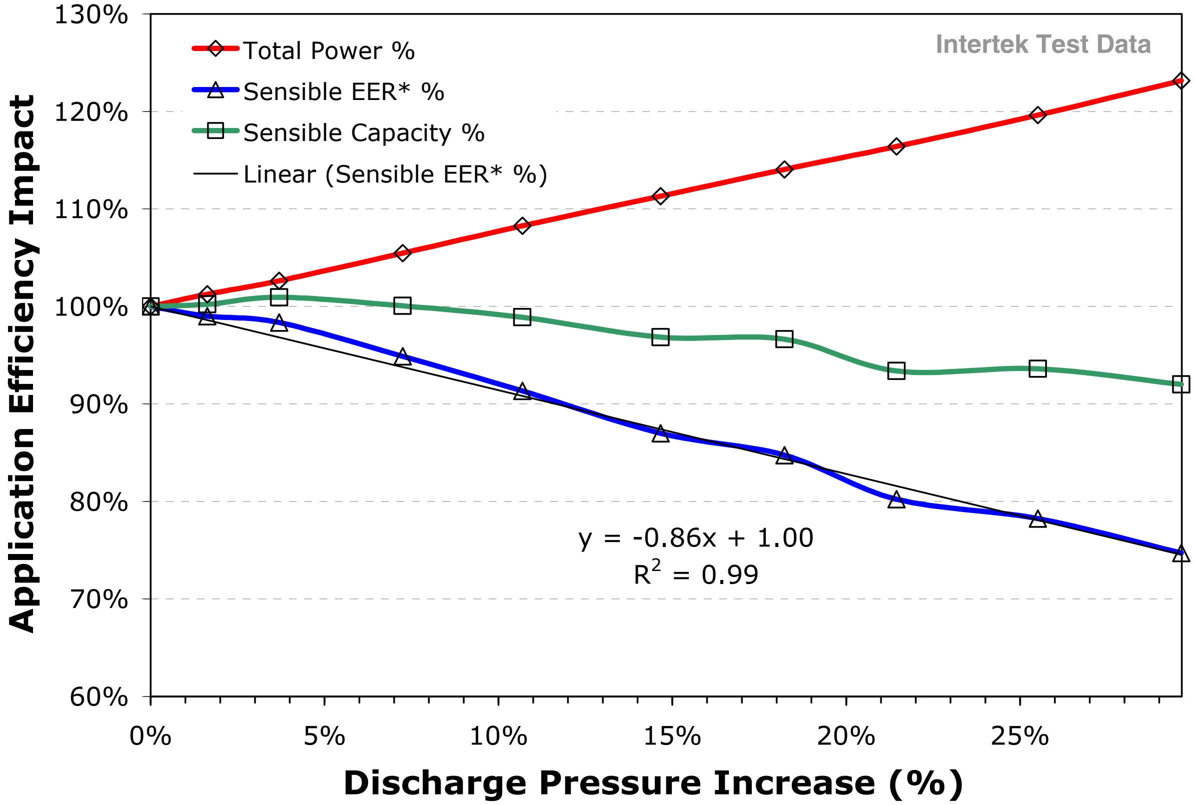


Table 84: Condenser Blockage Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed and 360 scfm/ton at 115F

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-CB0-115-CE	0.0%	-4.9	1.0	-7.5	362	347.7	4,283	2.49	14,018	3.27
C-CB5-115-CE	1.9%	-9.3	1.0	-7.7	362	354.4	4,353	2.33	13,791	3.17
C-CB10-115-CE	3.3%	-5.5	1.3	-7.9	363	359.3	4,405	2.21	13,681	3.11
C-CB20-115-CE	6.2%	-5.8	2.1	-8.1	364	369.2	4,512	2.00	13,363	2.96
C-CB30-115-CE	10.5%	-10.8	2.6	-8.4	362	384.3	4,681	1.78	13,005	2.78
C-CB40-115-CE	13.8%	-12.7	2.0	-8.5	362	395.8	4,818	1.65	12,809	2.66
C-CB50-115-CE	17.7%	-15.5	1.0	-8.9	362	409.2	4,998	1.46	12,397	2.48
C-CB60-115-CE	18.5%	-10.4	1.6	-9.0	360	411.9	5,062	1.33	12,167	2.40

Table 85: Condenser Blockage Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed and 360 scfm/ton at 82F

Test	Average Discharge Pressure %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Discharge Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-CB0-82-CE	0.0%	8.4	6.2	-0.5	361	230.6	3,119	8.92	22,594	7.24
C-CB5-82-CE	2.1%	5.6	4.8	-0.7	367	235.5	3,169	8.81	22,762	7.18
C-CB10-82-CE	4.0%	3.7	3.7	-0.5	366	239.9	3,208	8.73	22,899	7.14
C-CB20-82-CE	8.1%	0.2	1.6	-0.7	364	249.2	3,294	8.42	22,559	6.85
C-CB30-82-CE	12.7%	-7.1	-4.4	-0.8	365	259.8	3,402	8.02	22,488	6.61
C-CB40-82-CE	17.1%	-12.4	-9.0	-0.9	365	270.1	3,506	7.70	22,353	6.38
C-CB50-82-CE	21.6%	-13.5	-9.6	-1.0	364	280.5	3,603	7.36	22,208	6.16
C-CB60-82-CE	22.4%	-13.6	-10.0	-1.0	358	282.2	3,618	7.25	21,746	6.01
C-CB70-82-CE	26.3%	-14.1	-10.0	-1.3	365	291.3	3,709	6.92	21,826	5.88
C-CB80-82-CE	31.1%	-13.6	-9.4	-1.5	360	302.4	3,815	6.52	21,296	5.58

This manufacturer provides “troubleshooting” procedures to diagnose condenser blockage from other faults.[133] The most common problem is excessive head pressure caused by the following faults: 1) dirty air filter, 2) dirty condenser coil, 3) refrigerant overcharge, 4) air in system (non-condensables), and 5) condenser air restricted or short-cycling. Technicians can easily check and correct dirty air filters and dirty (or blocked) condenser. If these corrections eliminate excessive head pressure, then there is no reason to adjust refrigerant charge or check for non-condensables.

4.3.8 Evaporator Blockage Fault Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate the impact of evaporator coil blockage faults on the application efficiency (EER*) of RTU5 with economizer #5 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In order to emulate dirt accumulation the evaporator, the coil was blocked with plastic corrugated cardboard on the upstream side next to the air filter. The inlet area was blocked from 5 to 50% to reduce evaporator airflow by 1 to 13%. Preliminary tests were performed without code tester installed before each coil blockage test to match total static pressure with the code tester installed. Outdoor air leakage was tested at 55F OAT and found to be within 27 +/- 0.4% at 5 to 50% evaporator coil blockage.

Evaporator coil blockage test results are provided in **Table 86** at 95F OAT. Diagnostic test results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature

[133] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 34, Troubleshooting. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. **Figure 42** shows the application energy efficiency impacts versus evaporator airflow decrease due to evaporator coil blockage. Evaporator coil blockage reduced EER*s by 1 to 9% and reduced sensible capacity by 1 to 11%. **Figure 42** provides the regression equation of application sensible energy efficiency (EER*s) versus evaporator airflow decrease caused by blocking the evaporator coil based on laboratory test data for a 3-ton non-TXV Unit at 95F OAT. **Equation 22** is the regression equation shown in **Figure 42**. **Equation 22** can be used to calculate the sensible energy efficiency impact associated with dirty/clean airflow ratio due to evaporator coil blockage for the 3-ton non-TXV unit.

Equation 22 $y_e = -0.744 x_a + 1.0$

Where,

y_e = EER*s impact of evaporator coil blockage based on airflow ratio decrease (dimensionless)

$x_a = \frac{cfm_b}{cfm_c} - 1$ = airflow ratio decrease due to evaporator coil blockage (dimensionless)

Table 87 and **Table 88** show evaporator coil blockage test results at 115F and 82F OAT. At 95F OAT, the manufacturer ΔST protocol correctly diagnosed proper charge for all tests. The CEC ΔSH protocol correctly diagnosed proper charge from 0 to 20% blockage and misdiagnosed a "false alarm" undercharge at 35 to 50% blockage. At 115F OAT, the manufacturer ΔST protocol misdiagnosed a "false alarm" undercharge for all tests except 50% coil blockage. At 115F OAT, the CEC ΔSH protocol correctly diagnosed proper charge for all tests. At 82F OAT, the manufacturer ΔST protocol correctly misdiagnosed a "false alarm" undercharge for all tests. At 82F OAT, the CEC ΔSH protocol misdiagnosed a "false alarm" undercharge for 0 to 10% coil blockage and correctly diagnosed proper charge for 20 to 50% coil blockage. The CEC temperature split ΔTS protocol correctly indicates low capacity for all tests at 95F and 115F. At 82F OAT, the CEC ΔTS protocol misdetected low airflow at 35 and 50% evaporator coil blockage. All tests at 95F and 115F had sensible cooling capacities less than the ACCA Manual N sensible cooling load of 19,097 and 27,240 Btuh highlighted in red. All tests at 82F OAT have sensible cooling capacities greater than the ACCA Manual N sensible cooling load of 15,686 Btuh highlighted in green.

Table 86: Evaporator Coil Blockage Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 and Damper Closed at 95F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Suction Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-EB0-95-CE	0.0%	-1.9	0.0	-4.1	363	78.1	3,549	5.17	18,320	5.16
C-EB5-95-CE	-1.1%	-4.5	-2.4	-4.2	359	78.0	3,533	5.24	18,061	5.11
C-EB10-95-CE	-2.0%	-4.6	-2.8	-4.1	356	77.7	3,534	5.24	17,970	5.08
C-EB20-95-CE	-4.0%	-4.3	-4.2	-4.1	349	76.0	3,509	5.29	17,647	5.03
C-EB35-95-CE	-7.9%	-3.2	-5.1	-4.3	335	74.1	3,488	5.18	16,669	4.78
C-EB50-95-CE	-12.5%	-2.9	-5.7	-3.9	318	73.4	3,477	5.25	16,316	4.69

For these tests, the technician might diagnose low capacity, install clean air filters and clean the evaporator coils. Afterwards, the technician would recheck CEC ΔTS temperature split and diagnose low capacity again with no blockage due to 26.5% outdoor airflow with closed economizer dampers and perimeter unsealed (see **Table 78**). If the technician removed the economizer hood and sealed the gap between the economizer perimeter and the cabinet to reduce unintended outdoor air leakage, this would reduce outdoor airflow to 14% (per **Table 78**). This would reduce unintended outdoor leakage by 9.5%, increase EER*s by 9%, and the temperature split would be 1.5 indicating proper airflow and capacity. This maintenance procedure can be performed with a screw driver, accurate temperature sensors, and UL-181 waterproof tape without hooking up gauges to the refrigerant system. This example is applicable to most packaged HVAC units with overventilation or unintended outdoor air leakage. These tests indicate the importance of technicians following systematic procedures of checking and correcting obvious maintenance faults such as evaporator coil cleaning and installing clean air filters before performing FDD services.

Figure 42: Application Efficiency Impacts versus Evaporator Airflow Decrease due to Coil Blockage for 3-ton non-TXV RTU5 with Economizer #5 and Damper Closed at 95F

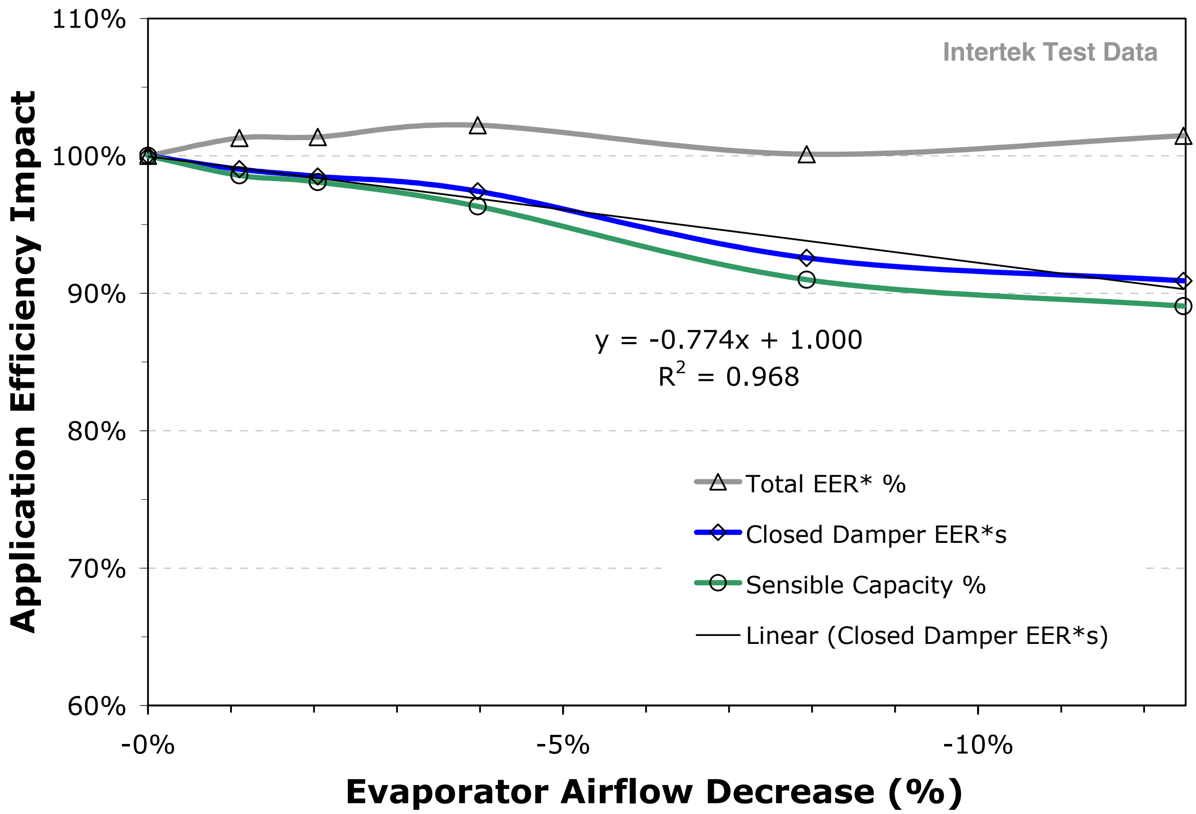


Table 87: Evaporator Coil Blockage Fault Impacts and FDD for 3-ton non-TXV RTU5 with Economizer #5 and Damper Closed at 115F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Suction Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-EB0-115-CE	0.0%	-11.9	0.3	-7.7	356	86.0	4,315	2.23	13,615	3.16
C-EB5-115-CE	-1.0%	-6.7	0.3	-7.7	352	84.8	4,293	2.39	13,471	3.14
C-EB10-115-CE	-1.5%	-5.4	0.5	-7.9	350	83.8	4,276	2.39	13,187	3.08
C-EB20-115-CE	-3.3%	-7.6	0.9	-8.1	344	82.5	4,251	2.28	12,714	2.99
C-EB35-115-CE	-8.4%	-6.0	0.5	-8.2	326	80.9	4,218	2.30	11,925	2.83
C-EB50-115-CE	-14.7%	0.3	0.1	-8.2	304	78.4	4,178	2.37	11,167	2.67

Table 88: Evaporator Coil Blockage Fault Impacts and FDD for 3-ton non-TXV RTU5 with Economizer #5 and Damper Closed at 82F

Test	Airflow %	Mfr Protocol ΔST	CEC Protocol ΔSH	CEC Protocol DTS	Airflow scfm/ton	Suction Press psig	Total Power W	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-EB0-82-CE	0.0%	9.0	7.2	-0.9	364	73.6	3,118	8.79	22,303	7.15
C-EB5-82-CE	-0.8%	8.6	6.1	-1.0	361	72.9	3,114	8.77	22,058	7.08
C-EB10-82-CE	-1.0%	8.6	5.5	-1.1	360	72.4	3,109	8.71	21,849	7.03
C-EB20-82-CE	-3.2%	8.7	4.7	-0.9	352	71.8	3,098	8.69	21,567	6.96
C-EB35-82-CE	-8.2%	8.5	2.2	-0.8	334	70.0	3,072	8.62	20,677	6.73
C-EB50-82-CE	-12.6%	7.6	-1.0	-0.6	318	68.3	3,056	8.50	19,891	6.51

This manufacturer provides “troubleshooting” procedures to diagnose evaporator blockage faults.[134] The most common problem is low suction pressure caused by the following faults: 1) dirty air filter and evaporator coil, 2) low refrigerant charge, 3) metering device or low-side restriction, 4) insufficient evaporator airflow, 5) temperature too low in conditioned space, or 6) filter drier restriction. Technicians can easily check and correct dirty air filter and evaporator coil. If these maintenance procedures eliminate low suction pressure faults, then there is no reason for additional FDD or correction.

4.3.9 Multiple Fault Tests for 3-ton non-TXV RTU5

Laboratory tests were performed to evaluate the impact of multiple faults on the application efficiency (EER*) of RTU5 with economizer #5 installed, dampers closed, and economizer perimeter unsealed. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. Tests were performed with 80 to 120% refrigerant charge per factory charge, 0 to 20% blocked evaporator, and 0 to 30% blocked condenser. Preliminary tests were performed without the code tester installed before each coil blockage test to match total static pressure with the code tester installed. At 55F OAT with economizer dampers closed and no evaporator coil blockage, the outdoor air leakage was 26.5% and with coil blockage the outdoor airflow was 27 +/- 0.4%.

[134] Carrier 1996. Installation, Start-Up and Service Instructions. 48HJD/HJE008-014, 48HJF008-012 Single-Package Rooftop Heating/Cooling Units. Page 33, Troubleshooting Table 16 – Cooling Service Analysis. http://dms.hvacpartners.com/docs/1005/public/0e/48hj-15si.pdf

The predicted application efficiency impacts for multiple faults are calculated using **Equation 14**. Measured versus predicted multiple fault impacts are provided in **Table 89** for damper closed (27% OAF), 80 to 120% refrigerant charge, 0 to 20% evaporator coil blockage, and 0 to 30% condenser coil blockage. The difference between measured versus predicted application efficiency impacts are calculated using **Equation 15**. The average difference is -0.3% indicating predicted multiple fault impacts based on summing individual impacts are slightly less than measured impacts. This is an important finding since HVAC maintenance involves multiple repairs and ex ante savings are typically summed for each repair.

Table 89: Measured versus Predicted Multiple Faults Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed (27% OAF), 80-120% Charge, 0-20% Blocked Evaporator, and 0-30% Blocked Condenser

Test	Charge Impact ε_i	Evap Block Impact ε_i	Cond Block Impact ε_i	Predicted Sensible EER*s	Measured Sensible EER*s	Difference Measured vs Predicted $\Delta\varepsilon$ %
C-R100-EB0-CB0-95CE	0.000	0.000	0.000	5.27	5.27	0.0%
C-R80-EB0-CB0-95CE	0.067	0.000	0.000	4.92	4.92	0.0%
C-R120-EB0-CB0-95CE	0.016	0.000	0.000	5.19	5.19	0.0%
C-CB30-95-CE	0.000	0.000	0.087	4.82	4.75	-1.4%
C-R80-EB0-CB30-95CE	0.067	0.000	0.087	4.47	4.56	2.2%
C-R120-EB0-CB30-95CE	0.016	0.000	0.087	4.73	4.65	-1.8%
C-EB20-95-CE	0.000	0.026	0.000	5.14	5.03	-2.2%
C-R80-EB20-CB0-95CE	0.067	0.026	0.000	4.79	4.76	-0.5%
C-R120-EB20-CB0-95CE	0.016	0.026	0.000	5.05	5.04	-0.2%
C-R100-EB20-CB30-95CE	0.000	0.026	0.087	4.60	4.53	-1.5%
C-R80-EB20-CB30-95CE	0.067	0.026	0.087	4.25	4.36	2.3%
C-R120-EB20-CB30-95CE	0.016	0.026	0.087	4.51	4.48	-0.8%
Average						**-0.3%**

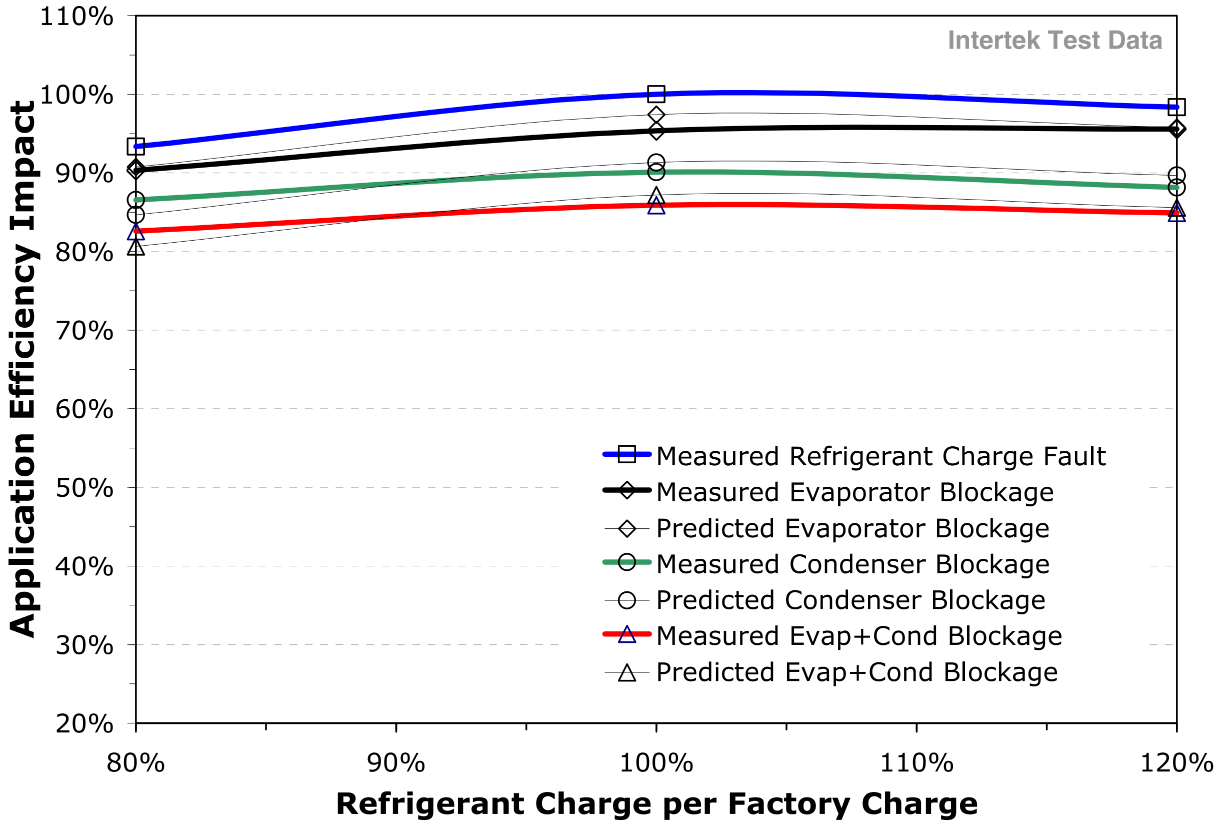
Multiple fault test results are provided in **Table 90**. Diagnostic tests results are provided for manufacturer and CEC refrigerant charge protocols (suction temperature ΔST and superheat ΔSH) and CEC temperature split (ΔTS) protocols. The manufacturer ΔST protocols correctly diagnosed proper charge for 10 of 12 tests. The manufacturer protocol misdiagnosed the non-faulted baseline factory charge (C-R100-EB0-CB0-95CE) and factory charge with 30% condenser blockage (C-CB30-95-CE). The CEC ΔSH protocol correctly diagnosed proper charge for 7 out of 12 tests. The CEC protocol misdiagnosed the correctly charged baseline and misdiagnosed the 120% factory charge tests as being correctly charged. The CEC ΔTS protocol correctly diagnosed all tests highlighted in yellow with low cooling capacity. All sensible cooling capacities are less than the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in red.

Table 90: Multiple Fault Impacts for 3-ton non-TXV RTU5 with Economizer #5 Damper Closed, 80-120% Charge, 0-20% Blocked Evaporator, and 0-30% Blocked Condenser

Test	C1/C2 Charge %	Blocked Evap Coil %	Blocked Cond Coil %	Mfr C1/C2 ΔST	CEC Protocol C1/C2 ΔSH	CEC Protocol ΔTS	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
C-R100-EB0-CB0-95CE	100	0	0	6.0	7.0	-3.7	5.40	18,687	5.27
C-R80-EB0-CB0-95CE	80	0	0	24.0	20.0	-5.5	4.67	16,724	4.92
C-R120-EB0-CB0-95CE	120	0	0	-9.0	-5.0	-3.3	5.35	18,671	5.19
C-R100-EB20-CB30-95CE	100	20	30	-3.0	-2.0	-4.6	4.43	17,323	4.53
C-R80-EB20-CB30-95CE	80	20	30	20.0	18.0	-5.8	4.16	15,937	4.36
C-R120-EB20-CB30-95CE	120	20	30	-8.0	-4.0	-4.6	4.35	17,104	4.48
C-CB30-95-CE	100	0	30	-11.1	-5.3	-4.2	4.70	18,207	4.75
C-R80-EB0-CB30-95CE	80	0	30	20.0	19.0	-5.4	4.23	16,831	4.56
C-R120-EB0-CB30-95CE	120	0	30	-10.0	-4.0	-4.3	4.40	17,996	4.65
C-EB20-95-CE	100	20	0	-4.3	-4.2	-4.1	5.29	17,647	5.03
C-R80-EB20-CB0-95CE	80	20	0	25.0	19.0	-5.7	4.63	16,105	4.76
C-R120-EB20-CB0-95CE	120	20	0	-7.0	-5.0	-4.0	5.16	17,953	5.04

Figure 43 shows the measured and predicted energy efficiency impacts versus varying refrigerant charge per factory charge, evaporator coil blockage, condenser coil blockage, and evaporator plus condenser coil blockage. EER*s was maximized with 120% of factory charge. Efficiency decreased by 7% at 80% of factory charge. Efficiency decreases by an additional 3 to 9% with 20% blocked evaporator or 20% blocked evaporator plus 30% blocked condenser coil. The predicted impacts (gray dashed lines) for evaporator and condenser coil blockage and evaporator plus condenser coil blockage are 0.3% higher than measured impacts. These tests indicate that multiple faults can mitigate the impacts of individual faults.

Figure 43: Measured and Predicted Multiple Fault Application Efficiency Impacts for 3-ton non-TXV RTU5 with Economizer #5 and Damper Closed, 80-120% Charge, 0-20% Blocked Evaporator, and 0-30% Blocked Condenser



Most HVAC systems have multiple faults. Troubleshooting multiple faults using manufacturer procedures will reduce or eliminate "false alarms," misdetection, and misdiagnosis. These examples indicate multiple faults such as undercharge or overcharge plus evaporator and condenser blockage cause FDD protocols to misdiagnose undercharge. Technicians need to visually diagnose and correct evaporator and condenser coil blockage and check overventilation and other more complicated faults before performing refrigerant charge FDD.

4.4 Test Results for 3-ton TXV Packaged HVAC RTU4

One 3-ton multiple fixed-orifice TXV packaged HVAC model (RTU4) was tested in the laboratory per the ANSI/AHRI 210/240 test procedure. The unit uses R22 refrigerant and was shipped with a 1.5 horsepower (HP) blower motor, forward-curved centrifugal blower wheel with 1" wide blades, 11.5" diameter x 9" width. RTU4 has one compressor and is equipped with one thermostatic expansion valve (TXV) on the evaporator inlet. The unit was shipped from the factory with a 5.75 inch diameter fan pulley and the motor sheave set to 2.5 turns out from the maximum fan speed setting. The manufacturer installation, start-up, and service instructions indicate that the motor sheave is typically set from the factory to 2.5 turns to provide between

615 and 920 revolutions per minute (RPM) fan speed.[135] The unit was tested in the horizontal configuration.

4.4.1 Out-of-Box Tests for 3-ton TXV RTU4

The 3-ton TXV RTU4 was tested in the "out-of-the-box" as-purchased condition in the horizontal configuration. **Table 91** provides the out-of-box tests for RTU4 at various fan speeds, pressures, and airflows. Tests were performed at 95F outdoor conditions, and 80F DB and 67F WB indoor conditions. The first test was performed at 808 rpm (2.5 turns) and 662 scfm/ton and the EER* was 10.7 or 4% less than the rated 11.2 EER (L-ONE-SS). The motor sheave was adjusted to 5 turns to reduce fan speed to 664 rpm and the Code Tester was adjusted to increase ESP to 0.30 IWC and achieve 445 scfm/ton. This is slightly less than the 450 scfm/ton ANSI/AHRI 210/240 maximum allowable airflow. For second test (L-ONE-ST) the EER* was 11.4 or 2% greater than the published 11.2 EER rating and within ANSI/AHRI tolerances. The L-ONE-5T total cooling capacity was 35,761 Btuh or 1.8% less than the 36,400 Btu/hr AHRI rating at 95F outdoor and 80F DB and 67F WB indoor conditions and within ANSI/AHRI tolerances.[136] Intertek recovered and weighed the refrigerant charge from the unit and found 6.4 lbs or 9% less than the manufacturer factory charge of 7 lbs. The unit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the 7 lbs factory charge. The cabinet panel joints were sealed with tape to reduce outdoor air leakage. For test L-22A with factory charge, the ESP was increased to 0.36 IWC to achieve 400 scfm/ton rated airflow and the EER* was 11.5 or 2.3% greater than the published 11.2 EER rating. The cooling capacity was 35,337 Btuh or 3% less than the 36,400 Btu/hr AHRI rating.

Table 91: Out-of-Box AHRI Tests for 3-ton TXV RTU4 with 5.75" Diameter Blower Pulley and without Economizer

Test	Charge %	Fan Turn	Fan RPM	ESP IWC	Airflow scfm/ton	Fan W	Total W	Total Cooling Capacity Btuh	Rated EER	Tested EER*	ΔEER*	Tested Sensible EER*s
L-ONE-SS	91	2.5	808	0.25	662	370	3,431	36,612	11.2	10.7	-4%	9.1
L-ONE-5T	91	5	663	0.30	445	360	3,129	35,761	11.2	11.4	2%	8.6
L-22A	100	5	664	0.36	400	330	3,084	35,337	11.2	11.5	3%	8.2

135 Lennox 2009. Lennox Service Literature Unit Information L Series 3 to 6 ton, LGA/LCA LGC/LCC 9822-L12, Revised 01/2009. http://tech.lennoxintl.com/C03e7o14l/xddfVVShbC/9822h.pdf

136 Per ANSI/AHRI STANDARD 210/240-2008, "6.5 Tolerances. To comply with this standard, measured test results shall not be less than 95% of Published Ratings for performance ratios and capacities.."

4.4.2 AHRI Verification Tests for 3-ton TXV RTU4

The ANSI/AHRI Standard 210/240 verification tests were performed to verify rated performance and provide the DEER DMQC team with part-load cycling data for developing building energy simulations. **Table 92** provides ANSI/AHRI EER and SEER verification cycling test data for RTU4. The measured EER at 95F OAT was 11.5 and the measured EER at 82F OAT was 14.0. The EER at 95F was 2.3% higher than the published rating. The average SEER was 13.1 based on three SEER tests and the average cyclic degradation coefficient (C_d) was 0.137.[137] The EER was within 2%, cooling capacity was within 2.9%, and SEER was within 0.5% of the published ratings. All measured values were within ANSI/AHRI tolerances.[138]

Table 92: ANSI/AHRI EER* and SEER* Verification and Cycling Tests for 3-ton TXV RTU4 without Economizer

Test	ESP IWC	Airflow scfm/ton	Rated Cooling Capacity Btuh	Rated EER	Rated SEER	Total Cooling Capacity Btuh	Tested EER	Tested SEER	C_d
L-22A	0.36	400	36,400	11.2		35,337	11.5		
L-22B	0.36	400	36,400		13	38,515	14.0		
L-22C-L22D Cyclic #2					13			13.0	0.145
L-22C-L22D Cyclic #3					13			13.1	0.126
L-22C-L22D Cyclic #4					13			13.1	0.139
Average								**13.1**	**0.137**

4.4.3 Manufacturer Refrigerant Charge Diagnostics for 3-ton TXV RTU4

The unit-specific manufacturer refrigerant charge diagnostics for RTU4 is based on discharge pressure (DP) and suction pressure (SP) as a function of outdoor drybulb (DB) temperature (i.e., condenser entering air) and subcooling (SC).[139] The manufacturer refrigerant charge ΔDP

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137 Cycling degradation (CD) coefficient measures the efficiency loss due to cycling of units as determined in Appendices C and D of ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment Standard 210/240. Air-Conditioning Heating and Refrigeration Institute.

138 Per ANSI/AHRI STANDARD 210/240-2008. 6.5 Tolerances. To comply with this standard, measured test results shall not be less than 95% of Published Ratings for performance ratios and capacities."

139 Lennox 2009. Lennox Service Literature Unit Information L Series 3 to 6 ton, LGA/LCA LGC/LCC 9822-L12, Revised 01/2009. http://tech.lennoxintl.com/C03e7o14l/xddfVVShbC/9822h.pdf. Lennox 2009. Lennox Service Literature Unit Information L Series 3 to 6 ton, LGA/LCA LGC/LCC 9822-L12, Revised 01/2009. http://tech.lennoxintl.com/C03e7o14l/xddfVVShbC/9822h.pdf. Lennox Industries, Inc., 2008. Application and Design Guidelines: Lennox Refrigerant Piping Design and Fabrication Guidelines. One. Corp. 9351−L9.

tolerances are +/-10, ΔSP tolerances are +/-5F, and ΔSC tolerances are +/-1F.[140] The unit does not have recommended superheat target values. The manufacturer does not provide airflow diagnostic protocols. The CEC subcooling (ΔSC) protocols are used to evaluate refrigerant charge and CEC temperature split (ΔTS) protocols are used to evaluate airflow and sensible cooling capacity faults based on test results for the 3-ton TXV unit.[141] For information about the CEC protocols see **Section 2.1.3**. The laboratory tests provide sensible cooling minus ventilation loads which are included in the measurements. Sensible cooling capacity test results are diagnosed using ACCA Manual N sensible cooling load design values minus ventilation loads provided in **Table 2** and described in **Section 2.2.1**.

Table 93 shows manufacturer and CEC refrigerant charge and airflow diagnostic results for the non-TXV RTU4.[142] The manufacturer protocols (ΔDP and ΔSP) misdiagnosed 91% factory charge test (L-ONE-ST), but capacity and efficiency are within 95% of the published AHRI 11.2 rating and 36,400 Btuh cooling capacity. For the 100% charge test the manufacturer protocols correctly diagnosed proper charge (L-22A, L-22B) highlighted in green. At 100% charge the CEC ΔSC protocol misdiagnosed a slight undercharge (ΔSC < -3F) highlighted in red. The CEC ΔTS protocol correctly diagnosed proper airflow and cooling capacity for all tests highlighted in green. The sensible cooling capacities for all tests are greater than 105% of the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 93: Manufacturer and CEC RCA Diagnostics for 3-ton TXV RTU4 without Economizer at 95F and 82F Outdoor Temperature

Test	Charge %	Mfr Protocol ΔDP	Mfr Protocol ΔSP	Mfr Protocol ΔSC	CEC Protocol ΔSC	CEC Protocol ΔTS	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
L-ONE-5T	91	-1	-1	NA	NA	-0.6	3,129	35,761	11.4	26,805	8.6
L-22A	100	-2	-2	-0.6	-3.6	-0.2	3,084	35,337	11.5	25,303	8.2
L-22B	100	0	-2	-0.2	-3.2	1.8	2,746	38,515	14.0	27,455	10.0

[140] Manufacturer refrigerant charge protocol for undercharge: ΔDP < -10 psig, ΔSP <-5 psig, ΔSC <-1F. Manufacturer refrigerant charge protocol for overcharge: ΔDP > 10 psig, ΔSP > 5 psig, ΔSC > 1F. Manufacturer protocol for correct charge: -10 psig ≤ ΔDP ≤ 10 psig, -5 psig ≤ ΔSP ≤ 5 psig, -1F ≤ ΔSC ≤ 1F.

[141] California Energy Commission (CEC). 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. RA3.2 Procedures for Determining Refrigerant Charge for Split System Space Cooling Systems Without a Charge Indicator Display. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

[142] California Energy Commission (CEC). 2008. Reference Appendices for the 2008 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. CEC-400-2008-004-CMF. Appendix RA3 - Residential Field Verification and Diagnostic Test Protocols. Effective January 1 2010. http://www.energy.ca.gov/2008publications/CEC-400-2008-004/CEC-400-2008-004-CMF.PDF

Table 94 provides manufacturer and CEC refrigerant charge and airflow diagnostic tests for the 3-ton TXV RTU4 with economizer #6 installed and closed dampers. The tests were conducted at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. Refrigerant charge for the five tests varied from 60 to 140% of factory charge. The 100% factory charge test L-75629575-E3 total EER* is 31% less than published 11.2 EER AHRI rating due to typical static pressure and economizer installed with closed damper which provides 19.9% outdoor airflow . The 10% overcharge test L-95-3+10-C total EER* and sensible EER*s are roughly the same as factory charge. The 140% overcharge test L-95-3+40-C, total EER* is 2% less than factory charge and EER*s is the same as factory charge. The 10% undercharge test L-95-3-10-C total EER* is -6% less than factory charge and EER*s is 2% less. The 40% undercharge test L-95-3-40-C, total EER* is 40% less than factory charge and EER*s is 35% less than factory charge.

The manufacturer ΔSC protocol correctly diagnosed all charge conditions. The manufacturer ΔDP and ΔSP protocols correctly diagnosed factory charge and +40% overcharge tests and misdiagnosed 10% undercharge and 10 to 40% overcharge as correctly charged. The CEC ΔSC protocol misdiagnosed factory charge as undercharged and 10% overcharge as correctly charged. The CEC ΔTS protocol correctly diagnosed low sensible cooling capacity for 40% undercharge (yellow highlight) which is 29% less than ACCA Manual N (red highlight). The CEC ΔTS correctly diagnosed all other tests (i.e., 379 scfm/ton airflow). These tests indicate no significant issues with the manufacturer protocols in terms of providing consistent and reliable refrigerant charge fault detection diagnostic (FDD) information for units having faults impacting efficiency or capacity by more than 10%. All tests except 60% factory charge have sensible cooling capacities greater than the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in green.

Table 94: Manufacturer and CEC RCA Diagnostics for 3-ton TXV RTU4 with Economizer #6 and Closed Damper at 95F Outdoor Temperature and 380 scfm/ton

Test	Charge %	Mfr Protocol ΔDP	Mfr Protocol ΔSP	Mfr Protocol ΔSC	CEC Protocol ΔSC	CEC Protocol ΔTS	Total Power W	Total Cooling Capacity Btuh	Total EER*	Sensible Cooling Capacity Btuh	Sensible EER*s
L-95-3-40-C	60	-16	-17.5	-7.4	-10.4	-8.4	3,024	14,150	4.7	13,729	4.5
L-95-3-10-C	90	-4.2	-3.6	-5.0	-8.0	-2.8	3,120	22,856	7.3	21,350	6.8
L-75629575-E3	100	-3.0	-4.0	-0.9	-4.0	-2.2	3,124	24,278	7.8	21,875	7.0
L-95-3+10-C	110	-1.3	-4.1	2.0	-1.0	-2.4	3,149	24,112	7.7	22,013	7.0
L-95-3+40-C	140	3.0	-4.3	6.2	3.2	-2.6	3,202	24,276	7.6	22,337	7.0

4.4.4 Economizer Outdoor Airflow Tests for 3-ton TXV RTU4

Laboratory tests were performed to evaluate outdoor airflow, overventilation, and untended outdoor air leakage on the 3-ton TXV RTU4 with economizer #6 installed. In order to measure outdoor airflow, tests were performed on RTU4 with the evaporator fan blower motor on, both compressors off, outdoor conditions of 55F and indoor conditions of 75F DB and 62F WB. Accurate measurements of return temperature, outdoor temperature, and mixed air temperature entering the evaporator or leaving the fan, air pressure (p), airflow (scfm), and fan power are used to calculate the outdoor airflow as a fraction of the total airflow across the evaporator coil. The outdoor, return, and mixed air drybulb temperatures were measured using resistance temperature detector (RTD) sensors in the outdoor, return, and supply air samplers. The outdoor air temperature entering the economizer was also measured using an array of 6 thermocouple sensors installed in the economizer inlet. The volumetric flow rate of air was measured using the Code Tester.[143] For tests with blower fan and compressors operating, the mixed air temperature entering the evaporator was measured with an array of 22 shielded-drybulb temperature sensors located on the air filter inlet adjacent to the evaporator.

Table 95 provides economizer #6 outdoor air fractions calculated using **Equation 6** and **Equation 9** at 55F OAT with no compressors operating and using **Equation 1** at 95F OAT with compressors operating with no economizer installed (equations are described in **Section 4.1.4**). The difference between using **Equation 6** and **Equation 9** is -1.6 +/- 0.18% at 55F OAT with no compressors operating. The mixed-air temperature array was moved out of position from the evaporator inlet during the 95F OAT economizer tests.

Table 95: Economizer #6 Outdoor Air Fractions Calculated using Equations 6 and 9 at 55F without Compressors and using Equation 1 at 95F with Compressors for 3-ton TXV RTU4

Description	Test	Evap Airflow scfm/ton	Equation 6 Calc OAF at 55F OAF_e %	Equation 9 Calc OAF at 55F OAF_t %	Test	Evap Airflow scfm/ton	Equation 1 Calc OAF at 95F OAF_m %
No Economizer					L-75629575-NE3-SS	364	3.9
Closed	L-55-CE-DM	379	19.9	21.9	NA		
1F (5.1V)	L-55-1ER-DM	391	27.8	29.3	NA		
2F (6V)	L-55-2ER-DM	397	32.5	34.0	NA		
3F (6.9V)	L-55-3ER-DM	404	38.2	39.8	NA		
100% Open (10V)	L-55-OER-DM	400	65.1	66.4	NA		

Tests were performed with unsealed economizer perimeter and with tape to seal around the economizer perimeter where it attaches to the cabinet under the hood as shown in **Figure 44**.[144] These tests were performed to measure the impact of reducing unintended outdoor air leakage. Technicians generally use their fingers to set the minimum damper position where 1-finger is assumed to be open 10%, 2-fingers 20%, and 3-fingers 30%. Using fingers to set minimum damper positions causes variations in the opening depending on finger size and placement with

143 The "code tester" is the airflow measuring apparatus described in Section 5.3 Test Chambers (Code Testers), ANSI/ASHRAE 41.2-1987 (RA92). Standard Methods for Laboratory Airflow Measurement.

144 Field procedures for sealing unintended economizer leakage: 1) remove fasteners to remove economizer hood, 2) clean dirt either side of gap where economizer attaches to cabinet, 3) apply UL-181 waterproof tape to seal around entire gap where economizer attaches to cabinet making sure tape does not interfere with damper operation, 4) reinstall economizer hood and secure all fasteners.

respect to the damper and frame. Finger diameters are as follows: 1-Finger is 0.7 inch (1.8 cm), 2-fingers is 1.3 inches (3.3 cm), and 3-fingers is 2 inches (5.1 cm).

Figure 44: Economizer #6 installed on 3-ton TXV Unit with Tape Sealing the Perimeter

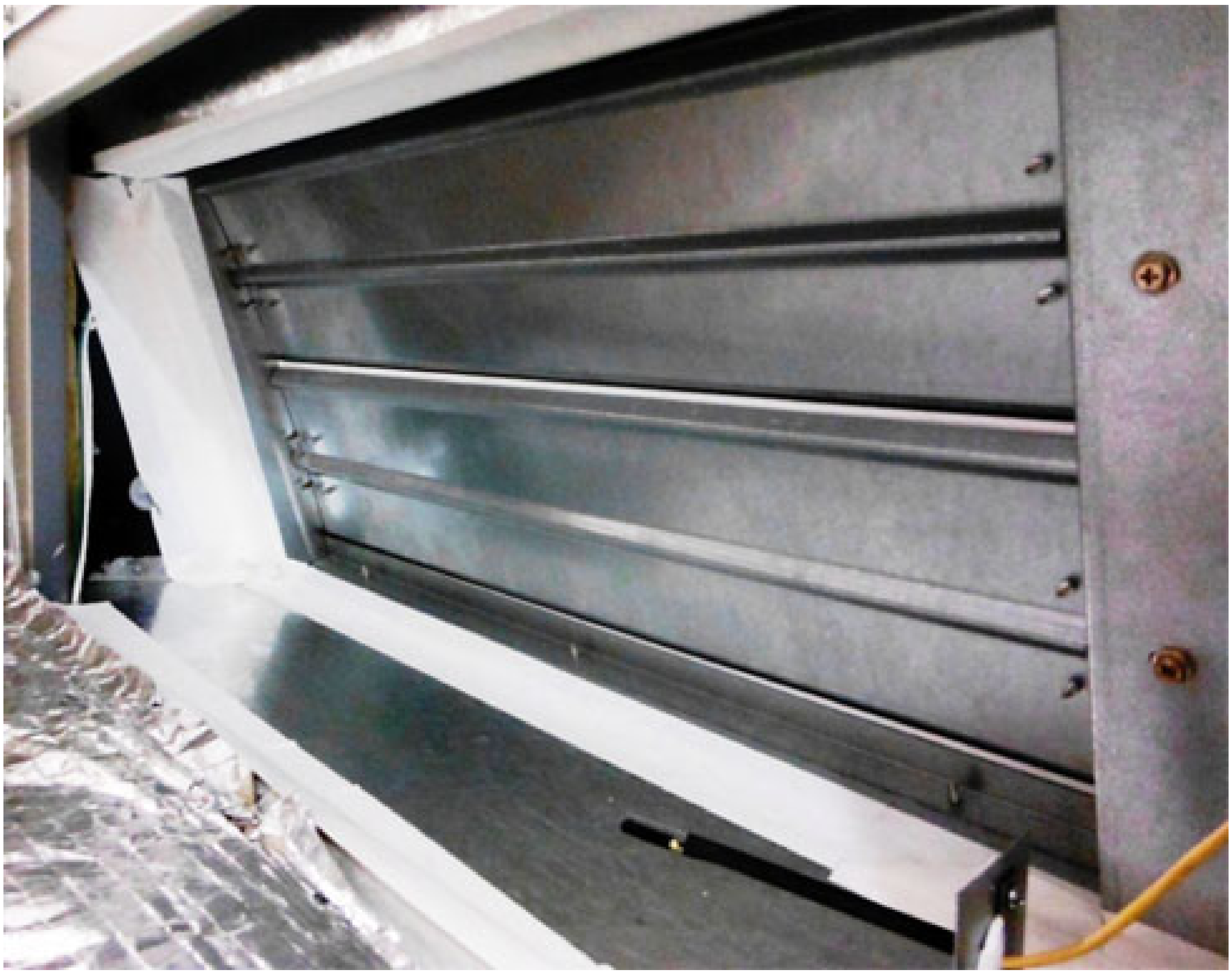


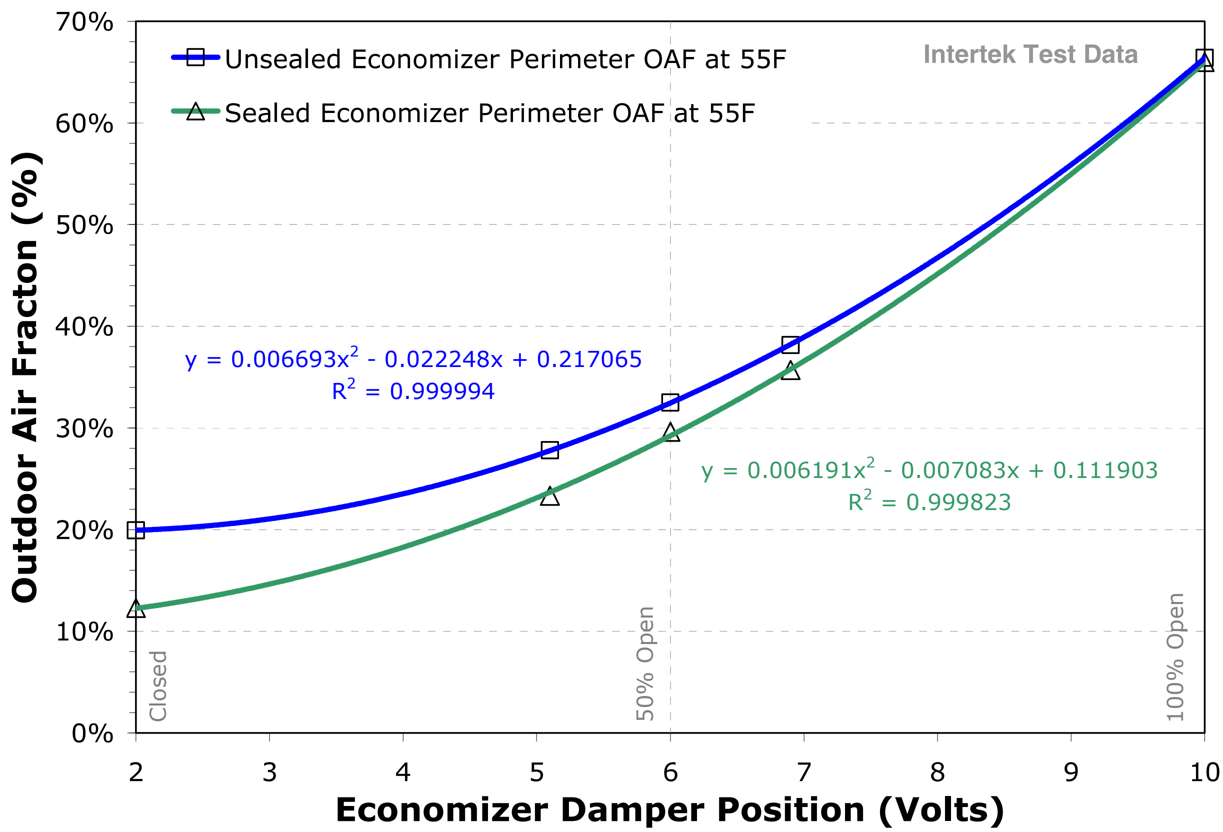
Table 96 provides the economizer #6 outdoor air fractions calculated using **Equation 6** at 55F OAT for the 3-ton TXV RTU4 with unsealed and sealed perimeter. With no economizer installed the OAF was 3.9% at 95F OAT. With perimeter unsealed and dampers from closed to fully open, the OAF ranged from 19.9 to 66.4% at 55F OAT. Sealing the perimeter (under the hood) reduced unintended OAF by 2.5 to 7.6%.

Table 96: Economizer #6 Outdoor Air Fractions Calculated using Equation 6 at 55F for 3-ton TXV RTU4 with Unsealed and Sealed Perimeter

Description	Test	Evap Airflow scfm/ton	Equation 6 Unsealed OAF_e at 55F %	Test	Evap Airflow scfm/ton	Equation 6 Sealed Perimeter OAF_e at 55F %	ΔOAF %
Closed (2.0V)	L-55-CE-DM	379	19.9	L-55-TCE-DM	367	12.3	7.6
1F (5.1V)	L-55-1ER-DM	391	27.8	L-55-T1ER-DM	383	23.3	4.5
2F (6V)	L-55-2ER-DM	397	32.5	L-55-T2ER-DM	392	29.6	2.9
3F (6.9V)	L-55-3ER-DM	404	38.2	L-55-T3ER-DM	401	35.7	2.5
100% Open (10V)	L-55-OER-DM	400	66.4	L-55-TOER-DM	402	66.0	

Figure 45 shows the outdoor air fraction versus damper position with unsealed and sealed perimeter for RTU4 with economizer #6. At the closed position, the difference between unsealed and sealed was 7.6%, and at 1-finger position, the difference was 4.5%.

Figure 45: Outdoor Air Fraction versus Economizer #6 Damper Position for 3-ton TXV RTU4 with Unsealed and Sealed Perimeter at 375 scfm/ton Total Airflow



Reducing unintended outdoor airflow and establishing the most efficient minimum damper position is important for health, comfort, and energy efficiency. Outdoor airflow provided by each economizer will vary and manufacturers typically do not provide outdoor airflow as a function of damper position. Technicians currently have no reliable measurement method to establish optimal damper position and are generally unaware how much energy is wasted by overventilation or unintended outdoor airflow.

4.4.5 Economizer 95F Efficiency Tests for 3-ton TXV RTU4

Laboratory tests were performed to evaluate the application efficiency impact of economizer outdoor air ventilation for the 3-ton TXV RTU4 with economizer #6 installed and outdoor air dampers closed, partially open, and 100% open and the economizer perimeter unsealed and sealed with tape. Tests were performed with factory charge and outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. **Table 97** provides economizer # 6 outdoor air ventilation impacts versus damper position with unsealed and sealed perimeter for the 3-ton RTU4 and 375 scfm/ton airflow at 95F OAT.

Table 97: Economizer #6 Outdoor Air Ventilation Impacts versus Damper Position with Unsealed and Sealed Perimeter for 3-ton TXV RTU4 and 375 scfm/ton at 95F

Description	Test	Unsealed Total EER*	Unsealed Sensible EER*s	Test	Sealed Perimeter Total EER*	Sealed Perimeter Sensible EER*s
No Economizer	L-75629575-NE3-SS	9.1	7.3	NA		
Closed	L-75629575-E3	7.7	7.0	L-75629575-TE3	8.6	7.2
1F (5.1V)	L-75629575-1E3	6.1	6.3	L-75629575-T1E3	6.9	6.6
2F (6V)	L-75629575-2E3	5.1	5.9	L-75629575-T2E3-1	5.8	6.2
3F (6.9V)	L-75629575-3E3	4.1	5.5	L-75629575-T3E3	4.6	5.7
100% Open (10V)	L-75629575-OE3	-0.6	3.6	L-75629575-TOE3	-0.5	3.5

At constant fan speed the ESP declines from 0.68 IWC with closed dampers to 0.51 IWC with fully open dampers. The inlet static pressure (ISP) was -0.02 IWC with fully open dampers and - 0.226 IWC with closed dampers. With no economizer installed the total EER* was 9.1 and the EER*s was 7.4. With economizer #6 installed and dampers closed and perimeter unsealed, the total EER* was 7.7 and EER*s was 7.0. The reduction in efficiency with economizer #6 installed and closed dampers was 15.3% for total EER* and 4% for EER*s. With closed dampers the economizer #6 efficiency was 31% less than the AHRI EER rating of 11.2. Opening dampers per the outdoor air leakage tests performed at 55F in the previous section significantly reduced efficiency. With 100% open dampers, the application sensible efficiency was 3.6 EER* for unsealed and 3.5 EER* for sealed perimeter which are 56% less than the AHRI rating and 51% less than the efficiency with no economizer installed. With 100% open dampers the total EER* was -0.6 unsealed and the EER*s was -0.5 sealed. These EER* values are negative due to outdoor airflow supplying more latent load than the evaporator can remove. For a building requiring 15% outdoor airflow, per ASHRAE 62.1, economizer #6 would provide 23.5% OAF with closed dampers and unsealed perimeter and 15% OAF at 3.1V or 13.8% open and sealed perimeter. Technicians would typically set the minimum damper position at 2-fingers open to achieve 15% OAF ventilation. At 2-fingers open, economizer #6 would provide 32.5% OAF or 2 times more OAF ventilation than at 3.1V with perimeter sealed. The overventilation at 2-fingers open (unsealed perimeter) would reduce EER*s to 5.9 EER*s or 16% less efficient than 7 EER*s at 3.1V with sealed perimeter. Providing adequate outdoor ventilation air is as important as providing comfortable indoor temperature control. The reduction in efficiency due to overventilation beyond minimum requirements represents an important energy efficiency opportunity for space cooling and heating.

Table 98 provides economizer #6 outdoor air ventilation versus damper position with unsealed and sealed perimeter for the 3-ton TXV RTU4 and 364 to 377 scfm/ton airflow at 95F OAT. The ESP was 0.5 IWC for 100% open dampers and 0.68 IWC for closed dampers. With the damper open 1–finger, the sensible cooling capacity and efficiency decreased by 8 to 18%. The manufacture protocols correctly diagnose refrigerant charge and the CEC protocols misdiagnose “false alarm” undercharge. The CEC ΔTS properly diagnoses 88% of tests with low sensible cooling capacity highlighted in yellow (i.e., ΔTS less than -3F) except no economizer and dampers closed which were correctly diagnosed with proper airflow.[145] The sealed 1-finger test sensible capacity highlighted in green was 3% greater than the 19,097 Btuh ACCA Manual N sensible cooling load. All unsealed and sealed tests with dampers from 2-fingers to fully open have sensible cooling capacities less than the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in red.

Excess outdoor air causes inadequate cooling capacity. The remedy is to adjust minimum damper position to meet ASHRAE 62.1 ventilation requirements, but without excess outdoor. Going from 1-finger open to closed position provides 19.7% outdoor air and increases capacity and efficiency by 12%. Closed damper with sealed economizer perimeter provides 12.3% outdoor air and improves EER*s to by 4 to 7.2 EER* versus 7.0 EER* unsealed perimeter.

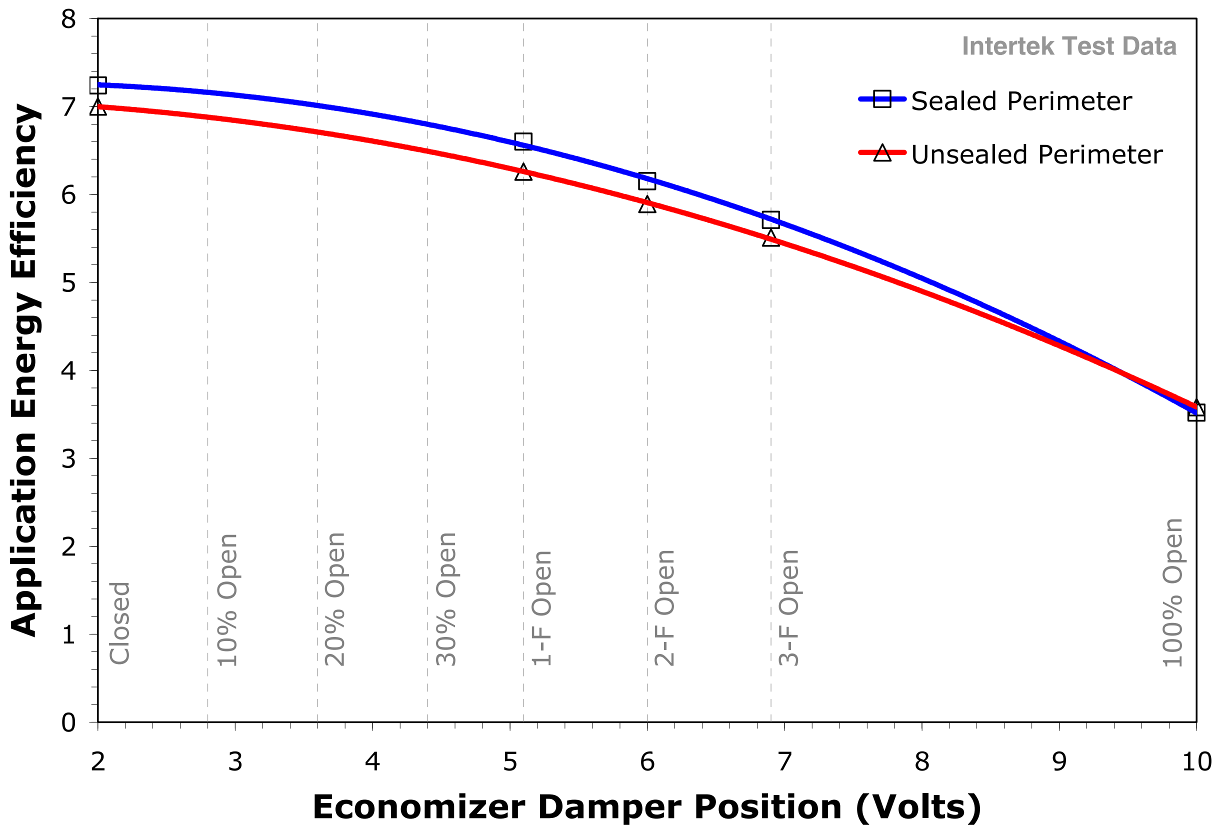
[145] Temperature split tests are based on well-mixed return and supply drybulb and wetbulb temperatures ignoring outdoor air mixing with return air which increases the evaporator inlet drybulb and wetbulb temperatures which are difficult for technicians to accurately measure in the field.

Table 98: Economizer #6 Outdoor Air Ventilation Impacts and FDD versus Damper Position Unsealed and Sealed Perimeter for 3-ton TXV RTU4 and 364-377 scfm/ton at 95F

Description	Test	OAF %	Mfr ΔDP	Mfr ΔSP	Mfr ΔSC	CEC ΔSC	CEC ΔTS	Total Power W	Total Capacity Btuh	Total EER*	Sensible Capacity Btuh	Sensible EER*s
No economizer	L-75629575-NE3-SS	3.7	-5	-10	-0.4	-3.4	-0.9	3,106	28,225	9.1	22,651	7.3
Unsealed												
Closed (2.0V)	L-75629575-E3	19.9	-2.8	-4.0	-0.9	-4.0	-2.2	3,124	24,278	7.8	21,875	7.0
1F (5.1V)	L-75629575-1E3	27.8	-2.5	-3.5	-0.4	-3.4	-4.2	3,128	19,016	6.1	19,581	6.3
2F (6V)	L-75629575-2E3	32.5	-2.4	-3.1	-0.3	-3.3	-5.2	3,131	16,064	5.1	18,441	5.9
3F (6.9V)	L-75629575-3E3	38.2	-2.2	-2.2	-0.2	-3.2	-6.3	3,134	12,767	4.1	17,254	5.5
100% Open	L-75629575-OE3	66.4	-0.4	2.7	-0.4	-3.4	-10.3	3,126	-1,535	-0.5	11,177	3.6
Sealed												
Closed (2.0V)	L-75629575-TE3	12.3	-3.8	-6.6	-0.8	-3.8	-1.2	3,115	26,838	8.6	22,551	7.2
1F (5.1V)	L-75629575-T1E3	23.3	-2.7	-3.9	-0.7	-3.7	-3.3	3,126	21,540	6.9	20,641	6.6
2F (6V)	L-75629575-T2E3-1	29.6	-2.5	-3.5	-0.5	-3.5	-4.5	3,125	18,092	5.8	19,208	6.1
3F (6.9V)	L-75629575-T3E3	35.7	-2.4	-2.8	-0.3	-3.3	-5.7	3,129	14,505	4.6	17,862	5.7
100% Open	L-75629575-TOE3	66.0	-0.5	2.9	-0.5	-3.5	-10.5	3,130	-1,751	-0.6	10,922	3.5

Figure 46 shows the decline in the application sensible energy efficiency for the 3-ton TXV unit as dampers are opened from fully closed (2V) to fully open (10V) with economizer #6 installed. The average EER*s improvement is 4.2 +/- 0.7% from sealing around the economizer perimeter frame under the hood with UL-181 tape. Sealing the perimeter (under the hood) reduced unintended outdoor airflow and improved cooling and heating efficiency. The mandated outdoor ventilation rates for most building occupancies range from 6 to 10% for offices, 22% for retail, 33% for auditoriums and schools, 40% for restaurants and health clubs, and 53% or more for cafeterias and sports arenas. Ventilation rates for unoccupied spaces can be minimized to save energy. While overventilation is in fact a system load, most of the load could be avoided with optimal minimum economizer damper positions and eliminating unintended outdoor air leakage. Field observations found approximately 50% of units with economizers not working properly or dampers stuck 10 to 100% open with Molex plugs or other objects stuck between damper blades.

Figure 46: Application Sensible Efficiency versus Damper Position for 3-ton TXV RTU4 with Sealed and Unsealed Economizer #6 Perimeter at 95F OAT



Overventilation and unintended outdoor airflow are common maintenance faults on all commercial buildings. Reducing overventilation can have a significant impact on thermal comfort, HVAC efficiency, and energy use. The manufacturer provides tables of information for "troubleshooting" faults with the cooling and heating system. Too much outdoor air is identified as a possible cause of inadequate heating. The manufacturer lists the following faults for inadequate heating: 1) dirty air filter, 2) gas input to unit too low, 3) unit undersized, 4) restricted airflow, 5) blower speed too low, 6) limit-switch causes main burners to cycle, and 7) too much outdoor air. Technicians can check and correct dirty air filters, restricted airflow, and blower speed (fan belt tension/alignment, pulley and motor sheave). Technicians can also check gas input pressure and burner limit switch. Too much outdoor air can be checked and corrected by adjusting the economizer minimum damper position.

4.4.6 Refrigerant Charge Fault Tests for 3-ton TXV RTU4

Laboratory tests were performed to evaluate the impact of refrigerant charge faults on the application efficiency (EER*) of RTU4. Tests were performed with factory charge varying from

60 to 140% of factory charge, economizer #6 perimeter unsealed, dampers closed and 1-finger open, and 376 to 407 scfm/ton total evaporator airflow. Tests were performed at outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB. With dampers closed the unsealed outdoor air leakage was 19.9% and with dampers at 1-finger open outdoor air leakage was 27.8% per **Table 95**. The circuit-specific manufacturer refrigerant charge diagnostic protocols include delta discharge pressure (ΔDP), suction pressure (ΔSP), and subcooling (ΔSC). The manufacturer tolerances are +/-10 psig for ΔDP, +/-5 psig for ΔSP and +/-1F for ΔSC. The CEC protocols include delta subcooling (ΔSC) to evaluate refrigerant charge and delta temperature split (ΔTS) to evaluate airflow and sensible cooling capacity. The CEC tolerances are +/-3F for subcooling and temperature split.

Table 99 provides application energy efficiency impacts and fault detection diagnostics (FDD) versus refrigerant charge per factory charge for RTU4. Total and sensible application efficiencies are maximized at 100% factory charge. Undercharging refrigerant by 10 to 40% reduced EER*s by 1 to 41% and overcharging increased EER*s by 0 to 2%. Opening dampers from closed to 1-finger increased outdoor airflow from 20 to 28% and reduced efficiency by 8 to 19%. The manufacturer subcooling protocol provided 100% accuracy but the overall accuracy was 56% due to discharge/suction pressure protocols correctly diagnosing factory charge and 30 to 40% undercharge but not detecting 10 to 20% undercharge and 10 to 40 overcharge. The CEC subcooling protocols provided 56% accuracy by providing correct diagnosis for -10 to -40% undercharge and +40% overcharge, “false alarm” undercharge at factory charge, and missed detection at +10 to +30% overcharge. At 40% overcharge DP was 228 psig. The high pressure cut out is 450 +/- 10 psig and the reset is 300 +/- 20 psig.[146] At -40% undercharge the DP was 209 psig. The low pressure cut out is 140 +/- 10 psig. The closed damper 60 to 70% undercharge tests and the 1-finger open 60 to 80% undercharge tests have sensible cooling capacities less than the 19,097 Btuh ACCA Manual N sensible cooling load highlighted in red. All other charge tests have sensible cooling capacities greater than the ACCA Manual N sensible cooling load highlighted in green or yellow. The CEC temperature split protocols diagnosed 78% of the tests with correct airflow, proper sensible cooling capacity, or low sensible cooling capacity (i.e., ΔTS less than -3F) highlighted in yellow or red.

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[146] Lennox 2009. Lennox Service Literature Unit Information L Series 3 to 6 ton, LGA/LCA LGC/LCC 9822-L12, Revised 01/2009. http://tech.lennoxintl.com/C03e7o14l/xddfVVShbC/9822h.pdf.

Table 99: Refrigerant Charge Fault Impacts for 3-ton TXV RTU4 with Economizer #6 Damper Closed and 1-Finger Open and 376 to 407 scfm/ton at 95F OAT

Test	Refrig Charge %	Mfr Protocol ΔDP	Mfr Protocol ΔSP	Mfr Protocol ΔSC	CEC Protocol ΔSC	CEC Protocol ΔTS	Total Power W	Total Capacity Btuh	Total EER*	Sensible Capacity Btuh	Sensible EER*s
Undercharge Closed											
L-95-3-40-C	60	-16.0	-17.5	-7.4	-10.4	-8.4	3,024	14,150	4.7	13,729	4.5
L-95-3-30-C	70	-10.4	-8.3	-7.0	-10.0	-5.2	3,071	18,772	6.1	18,221	5.9
L-95-3-20-C	80	-6.2	-4.1	-6.4	-9.4	-3.4	3,100	21,494	6.9	20,517	6.6
L-95-3-10-C	90	-4.2	-3.6	-5.0	-8.0	-2.8	3,120	22,856	7.3	21,350	6.8
Factory Charge Closed											
L-75629575-E3	100	-3.0	-4.0	-0.9	-4.0	-2.2	3,124	24,278	7.8	21,875	7.0
Overcharge Closed											
L-95-3+10-C	110	-1.3	-4.1	2.0	-1.0	-2.4	3,149	24,112	7.7	22,013	7.0
L-95-3+20-C	120	0.0	-4.2	3.9	0.9	-2.4	3,165	24,196	7.6	22,130	7.0
L-95-3+30-C	130	1.5	-4.3	5.3	2.3	-2.4	3,180	24,384	7.7	22,299	7.0
L-95-3+40-C	140	3.0	-4.3	6.2	3.2	-2.6	3,202	24,276	7.6	22,337	7.0
Undercharge 1-Finger											
L-95-3-40-1	60	-15.9	-16.9	-7.4	-10.4	-10.6	3,032	8,467	2.8	11,215	3.7
L-95-3-30-1	70	-9.5	-7.2	-7.0	-10.0	-7.2	3,079	13,514	4.4	15,909	5.2
L-95-3-20-1	80	-5.8	-3.0	-6.3	-9.3	-5.2	3,109	16,580	5.3	18,727	6.0
L-95-3-10-1	90	-3.9	-2.8	-4.8	-7.8	-4.6	3,128	17,999	5.8	19,438	6.2
Factory Chg 1-Finger											
L-75629575-1E3	100	-3.0	-3.0	-0.4	-3.4	-4.2	3,128	19,016	6.1	19,581	6.3
Overcharge 1-Finger											
L-95-3+10-1	110	-1.2	-3.5	2.3	-0.7	-4.2	3,157	19,418	6.2	20,207	6.4
L-95-3+20-1	120	0.3	-3.6	4.1	1.1	-4.4	3,172	18,816	5.9	19,939	6.3
L-95-3+30-1	130	1.6	-3.7	5.4	2.4	-4.4	3,190	19,208	6.0	20,278	6.4
L-95-3+40-1	140	3.3	-3.7	6.4	3.4	-4.4	3,207	19,201	6.0	20,275	6.3

Figure 47 shows the application energy efficiency impacts versus refrigerant charge per factory charge with closed dampers and 95F OAT. Total EER* and EER*s are maximized at 100 to 110% factory charge. Undercharging reduced efficiency by 35 to 54% and overcharging reduced efficiency by 0 to 2%. **Figure 47** provides the polynomial regression equation curve-fit of application sensible energy efficiency (EER*s) versus refrigerant charge per factory charge for the 3-ton TXV RTU4 at 95F OAT. **Equation 11** is the regression equation shown in **Figure 47**. **Equation 11** and **Equation 23** can be used to calculate EER*s impacts associated with refrigerant charge adjustments based on recovery and weigh-out of refrigerant charge and reported charge adjustment.

Equation 23 $$y = 10.42^6 - 59.93^5 + 136.81x^4 - 154.39x^3 + 85.05x^2 - 16.96x$$

Where,

y = EER*s impact at refrigerant charge per factory charge ratio (dimensionless)

x = refrigerant charge per factory charge ratio (dimensionless)

Figure 47: Application Efficiency Impacts versus Refrigerant Charge per Factory Faults for 3-ton TXV Unit with Economizer #6 Damper Closed and 376 to 407 scfm/ton at 95F OAT

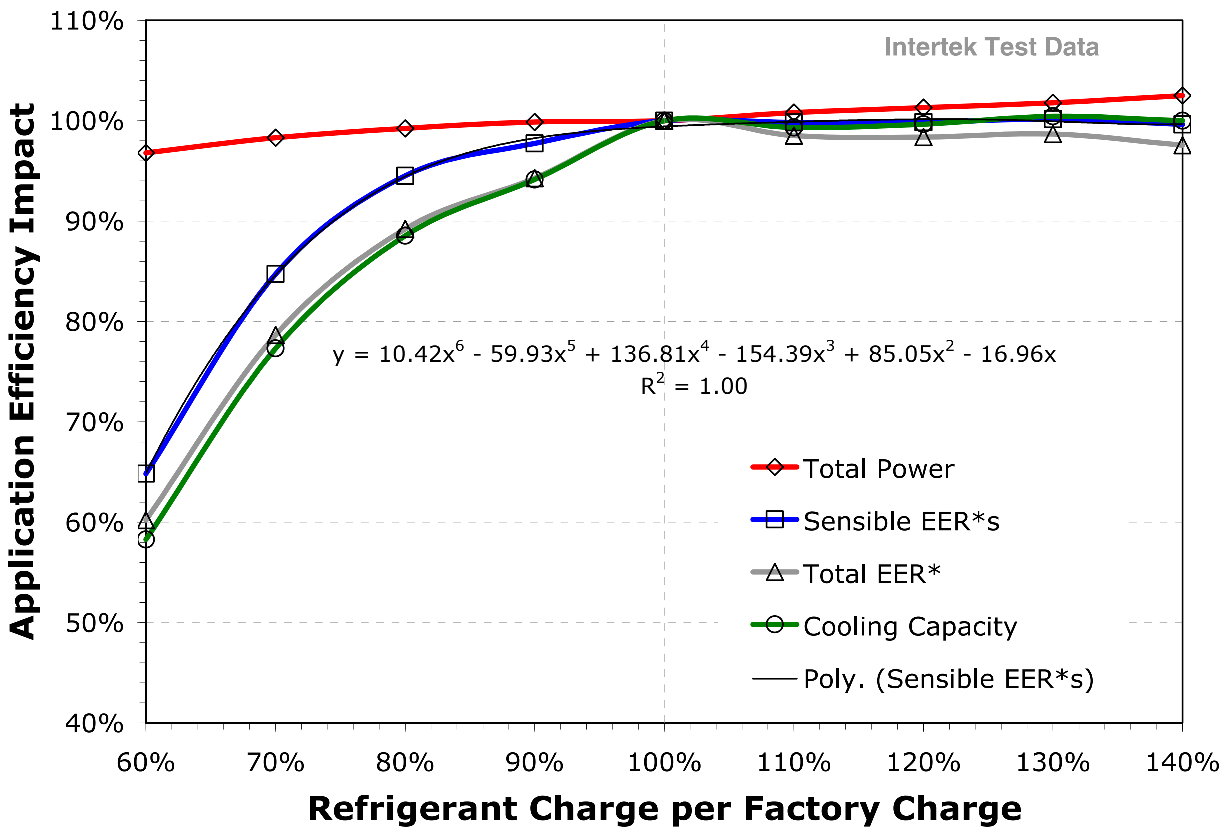
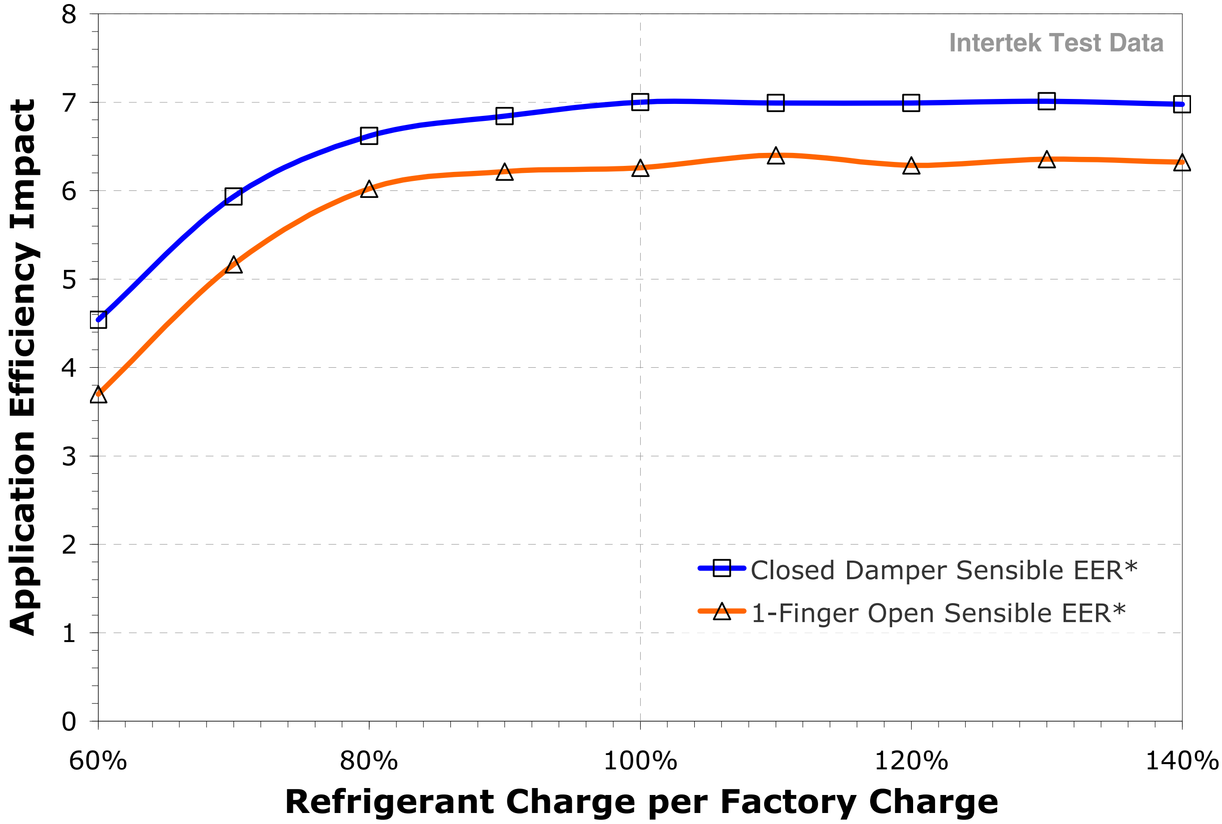


Figure 48 shows the application sensible energy efficiency impacts versus refrigerant charge per factory charge at 95F OAT with dampers closed and 1-finger open. Outdoor ventilation loads introduced by 1-finger damper positions reduced EER*s by 8 to 19% compared to closed damper. As noted previously, most of the ventilation load could be avoided with optimal minimum damper position and reducing unintended outdoor air leakage by sealing the economizer perimeter (under the hood) with UL-181 metal tape.

Figure 48: Application Sensible Efficiency Impact versus Refrigerant Charge per Factory Charge for 3-ton TXV Unit with Economizer #6 and Dampers Closed and 1-Finger Open and 376 to 407 scfm/ton at 95F OAT



Procedures for troubleshooting and servicing air conditioning systems also provided in technician training text books.[147] The most common problems are high or low discharge or suction pressure or continuous compressor operation. These problems are caused by a number of faults including: 1) dirty air filter, 2) blocked evaporator/condenser, 3) undersized unit (low cooling capacity or excessive outdoor air), 4) insufficient evaporator airflow, 5) refrigerant restriction, 6) non-condensables, 7) thermostat defective/set too low, 8) low line voltage (faulty contactor/transformer), 9) defective compressor/overload, or 10) refrigerant over/undercharge. Prior to adjusting refrigerant charge, technicians need to check and correct all other faults on the list. If none of the other faults are present and problem still exists, then refrigerant charge adjustments might be necessary.

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147 Tomczyk, J. 1995. Troubleshooting and Servicing Modern Air Conditioning and Refrigeration Systems. ESCO Press. Mt. Prospect, Ill.: Educational Standards Corporation.

4.5 Laboratory Tests of Field Measurement Instrument Accuracy

Laboratory tests were also performed on field measurement instruments from the largest manufacturers representing 80% of instruments used by technicians performing services in the HVAC Maintenance programs based on observations of technicians.

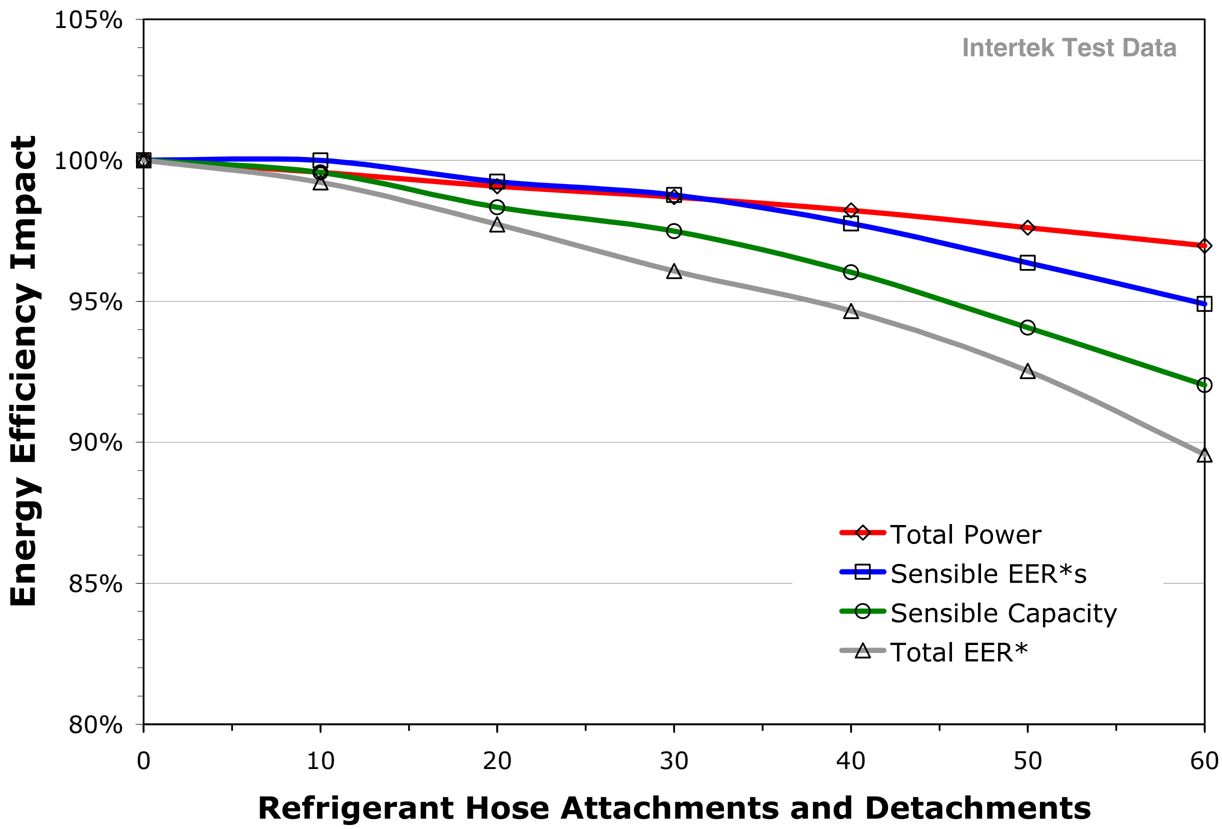
4.5.1 Refrigerant Hose Measurement Tests

When technicians attach and detach refrigerant hoses to discharge and suction pressure valves non-condensables can be accidentally added and refrigerant is released to the atmosphere. Refrigerant hose tests measured 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and efficiency per attach/detachment as shown in **Table 100** and **Figure 49**. **Table 100** shows the manufacturer refrigerant charge protocol passed DP and SP diagnostics with correct charge, but failed for SH (false alarm undercharge). Only after 60 attach/detachments do all of the manufacturer refrigerant charge diagnostics fail (DP, SP, and SH) with 25% undercharge in C1, 31% undercharge in C2, cooling capacity reduced by 13.1%, and efficiency reduced by 10.2%. Tests of liquid pressure attach/detachment might find more severe impacts due to liquid refrigerant having 20- to 40-times greater density.

Table 100: Refrigerant Hose Attachments and Detachments from Discharge and Suction Pressure Valves for 7.5-ton TXV RTU2

Test	Attach Detach Qty	C1/C2 Charge %	Mfr Protocol C1/C2 ΔDP	Mfr Protocol C1/C2 ΔSP	Mfr Protocol C1/C2 ΔSH	Cooling Cap Btuh	Total EER*
T2-MFB3-100C1-100C2-90A-95-CE-4	0	100/100	-7.5/-6.1	-4.5/-3.3	6.4/5.2	66,969	8.1
T2-RH1-CK1-CK2-100C1-100C2-90A-CE	10	96/95	-7.8/-8.5	-4.7/-3.2	7.7/5.3	66,162	8.1
T2-RH2-CK1-CK2-100C1-100C2-90A-CE	20	92/90	-9.2/-10.2	-4.8/-2.9	11.2/5.6	64,844	7.9
T2-RH3-CK1-CK2-100C1-100C2-90A-CE	30	88/85	-10.6/-11.2	-5.4/-2.9	17.7/7.1	63,509	7.8
T2-RH4-CK1-CK2-100C1-100C2-90A-CE	40	83/79	-12.3/-13.1	-6.5/-2.9	21.8/9.9	62,267	7.7
T2-RH5-CK1-CK2-100C1-100C2-90A-CE	50	79/74	-14.2/-15.3	-8.2/-3.7	24.6/16.2	60,485	7.5
T2-RH6-CK1-CK2-100C1-100C2-90A-CE	60	75/69	-16.3/-18	-10.5/-5.9	27.2/23.6	58,163	7.3

Figure 49: Performance Impacts of Refrigerant Hose Attachments and Detachments from Discharge and Suction Valves for 7.5-ton TXV RTU2



4.5.2 Refrigerant Tube Measurement Instrument Tests

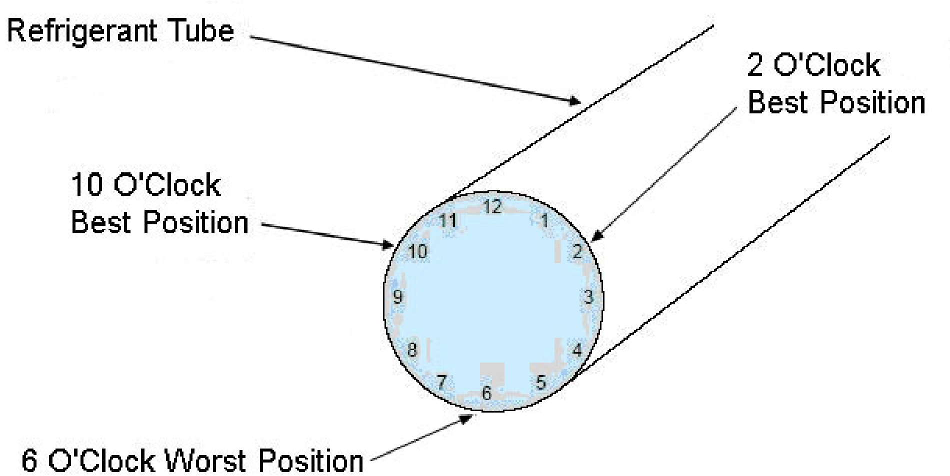
Refrigerant tube measurement instruments were tested at 55F, 95F, and 115F condenser entering air temperatures. Approximately 100 instruments from 9 manufacturers were tested. The refrigerant tube measurement instruments are described in **Table 101**. As shown in

Figure 50 the best positions to measure suction line temperatures are with sensors mounted at 10:00 or 2:00 o'clock (cross-section) near the Intertek sensors located at the service valves. The worst positions to measure suction line temperatures are with sensors mounted at the bottom 6:00 o'clock position with no insulation for bead or linear probes. These locations are only relevant on suction lines where liquid refrigerant or oil might be flowing at the bottom 6 o'clock position which can cause incorrect suction line temperature measurements. Other worst cases include improperly calibrated sensors or sensors mounted more than 12 inches from the service valve.

Table 101: Refrigerant Tube Measurement Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Type K Clamp Linear 0.25-1.375" dia.	8	A1-A8	10 or 2 o'clock	6 o'clock
2	Type K Clamp Sm. Scissors 0.375-1.375" dia.	8	B1-B8	10 or 2 o'clock	6 o'clock
3	Type K Clamp Lg. Scissors 0.375-2.25" dia.	8	C1-C8	10 or 2 o'clock	6 o'clock
4	10K Thermistor Clamp 0.25-1.4" dia	8	D1-D8	10 or 2 o'clock	6 o'clock
5	10K Thermistor velcro strap 3" dia.	7	E1-E7	10 or 2 o'clock	6 o'clock
6	Type K Clamp Sm. Scissors 0.25-1.375" dia.	8	F1-F8	10 or 2 o'clock	6 o'clock
7	10K Thermistor Clamp 0.375-2.125" dia.	5	G1-G5	10 or 2 o'clock	6 o'clock
9	Type K Clamp Sm. Scissors 0.25-2.125" dia.	1	I1	10 or 2 o'clock	6 o'clock
10	Type K Clamp Sm. Scissors 0.25-1.375" dia.	8	J1-J8	10 or 2 o'clock	6 o'clock
11	10K Thermistors 0.1875" x 1.125"	4	K1-K4	10 or 2 o'clock	6 o'clock and no insul.
12	10K Thermistors 0.25" dia. x 1.25"	8	L1-L8	10 or 2 o'clock	6 o'clock and no insul.
13	Type K bead AWG #24	8	N1-N8	10 or 2 o'clock	6 o'clock and no insul.

Figure 50: Best and Worst Positions to Attach Refrigerant Tube Sensors



Test results for refrigerant tube temperature measurement instruments are shown in **Figure 51** through **Figure 61**. The figures show that it can take 5 to 10 minutes or longer for sensors to correctly measure refrigerant suction (S) or liquid (L) line temperatures with the system already at steady-state conditions. Tests were conducted with eight sensors on liquid and suction lines. **Table 102** through **Table 112** provide average steady-state liquid and suction line temperatures for 15 to 20 minutes after sensors are attached. The greatest accuracy and smallest differences were found with specific Type-K scissors clamps (F1-F8) with differences of -0.1 +/- 0.06F on liquid lines and 0.08 +/- 0.04F on suction lines at 95F (**Table 106**). Some Type-K clamp probes have suction line accuracy ranging from 6.8 +/- 1.0F when tested at 115F outdoor conditions. Differences in accuracy are attributable to design and manufacturing. The largest differences were found with Type-K insulated bead probes and thermistors. Insulated bead probes had differences of 10.4 +/- 0.33F on suction lines at 95F, insulated cylindrical thermistors had differences of 9.7 +/- 0.7F, and clamp thermistors had differences of 5.4 +/- 0.22F. The largest differences are with the suction line measurements where tube temperatures are 25 to 40F less than ambient. The liquid line temperature is typically 8F to 12F above ambient so there are smaller variations from measured temperatures to actual tube temperatures. The average clamp thermistor liquid temperature differences were 5.1 +/- 0.09F at 95F indicating issues for liquid temperature (**Table 105**). It took 15 to 20 minutes to install bead and cylindrical probes and 15 to 20 minutes to bring test chambers to conditions so these probes were on tubes three times longer than clamp probes.

Tests of the best (10 o'clock) versus worst (6 o'clock) position for clamp thermistors (D1-D8) and insulated 10K thermistors are shown in **Figure 58** through **Figure 61** and **Table 109**, and **Table 110**. Liquid lines generally don't contain vapor so differences between best and worst positions are irrelevant. For clamp thermistors the best position suction line difference was 5.4 +/- 0.22F and worst position difference was 6.8 +/- 0.44F. **Table 110** shows the largest suction line difference was 19.4 +/- 0.03F for worst position (sensor D4), while **Table 109** shows the largest suction line difference was 10.2 +/- 0.5F for best position (sensor D2). For insulated 10K thermistors the best position suction line difference was 9.7 +/- 0.07F and worst position difference was 13.4 +/- 0.32F. These results indicate liquid at the bottom of the suction line can influence accuracy if the sensor is located at the bottom 6 o'clock position.

Figure 51: Type-K Pipe Linear Clamp A1-A8 Temperature Difference at 95F Ambient

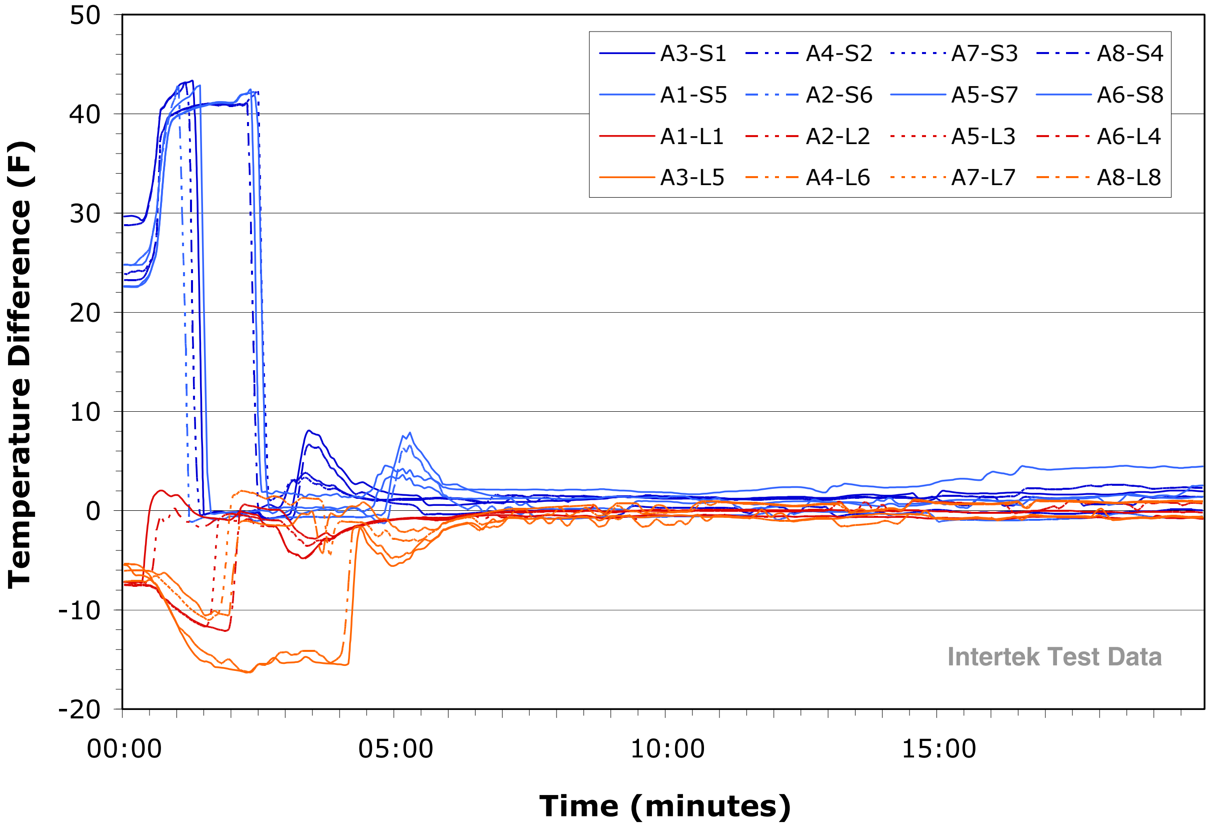


Table 102: Type-K Pipe Linear Clamp A1-A8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
A1-A8 Liquid	102.5±0.01	102.5±0.0	102.5±0.03	102.5±0.02	104.3±0.07	103.6±0.0	104.5±0.02	104.5±0.02	103.4±0.06
L1-L8 Liquid	103.2±0.01	103.2±0.01	103.2±0.02	103.2±0.02	103.7±0.01	103.7±0.01	103.7±0.02	103.7±0.02	103.4±0.02
Diff. Liquid	-0.7±0.01	-0.7±0.01	-0.7±0.04	-0.7±0.02	0.6±0.06	-0.1±0.01	0.8±0.03	0.9±0.03	**-0.1±0.05**
A1-A8 Suction	70.4±0.05	68.5±0.0	70.5±0.01	68.6±0.0	78±0.15	75.2±0.14	76.2±0.09	75.1±0.09	72.8±0.24
L1-L8 Suction	69.2±0.04	69.2±0.04	68.7±0.03	68.7±0.03	73.9±0.06	73.9±0.06	74.1±0.03	74.1±0.03	71.5±0.24
Diff. Suction	1.2±0.04	-0.8±0.04	1.8±0.03	-0.1±0.03	4.1±0.09	1.3±0.09	2.1±0.09	1±0.07	**1.3±0.09**

Figure 52: Type-K Scissors Clamp B1-B8 Temperature Difference at 95F Ambient

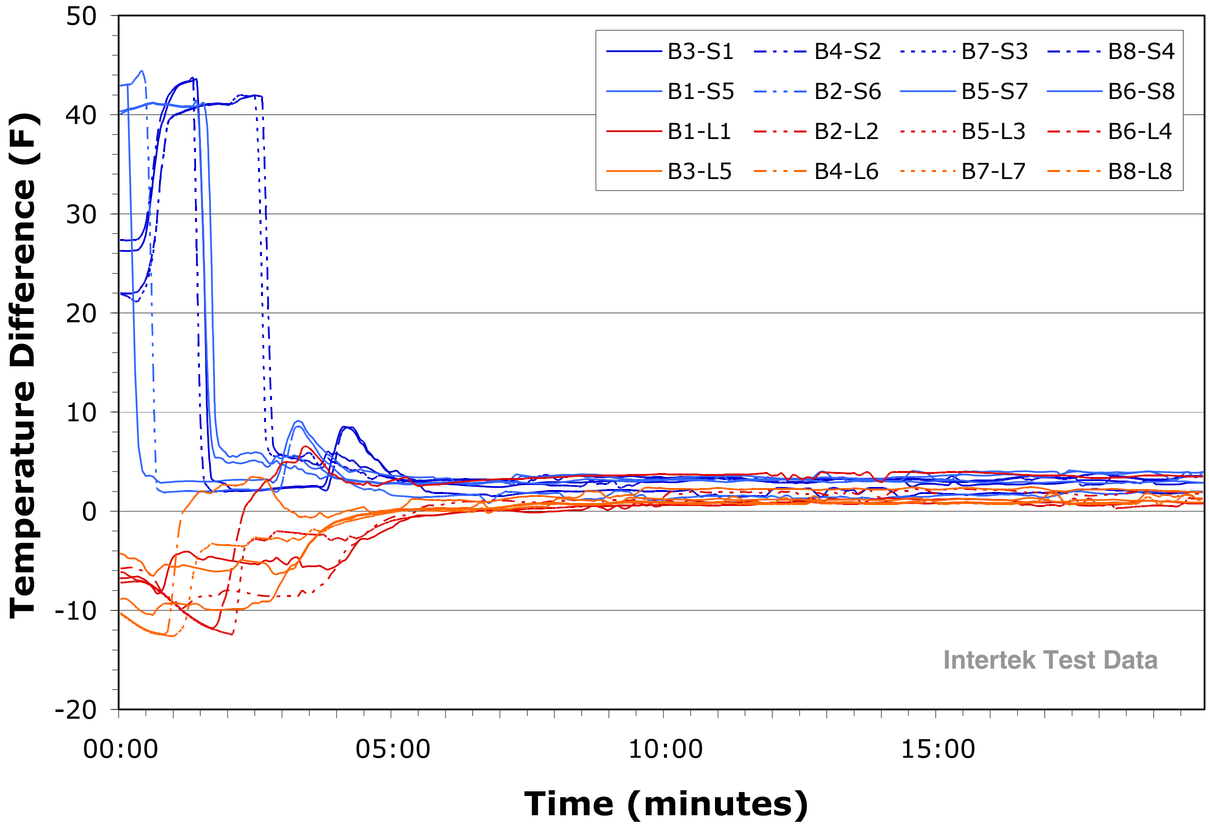


Table 103: Type-K Scissors Clamp B1-B8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
B1-B8 Liquid	104.5±0.0	105.5±0.01	104.5±0.0	105.6±0.0	104.7±0.06	107.6±0.0	104.7±0.06	104.8±0.07	105.2±0.07
L1-L8 Liquid	103.5±0.04	103.5±0.04	103.4±0.02	103.4±0.02	103.9±0.04	103.9±0.04	103.7±0.01	103.7±0.01	103.6±0.02
Diff. Liquid	0.9±0.04	1.9±0.04	1.1±0.02	2.2±0.02	0.7±0.05	3.6±0.04	0.9±0.05	1.1±0.07	**1.6±0.06**
B1-B8 Suction	72.5±0.0	70.5±0.0	71.5±0.02	70.6±0.0	77.5±0.0	78.4±0.06	77.4±0.06	77.6±0.01	74.5±0.22
L1-L8 Suction	68.7±0.03	68.7±0.03	68.7±0.03	68.7±0.03	74.5±0.02	74.5±0.02	74.2±0.04	74.2±0.04	71.5±0.27
Diff. Suction	3.8±0.03	1.8±0.03	2.8±0.03	1.9±0.03	2.9±0.02	3.8±0.05	3.2±0.05	3.4±0.04	**2.9±0.05**

Figure 53: Type-K Large Scissors Clamp C1-C8 Temperature Difference at 95F Ambient

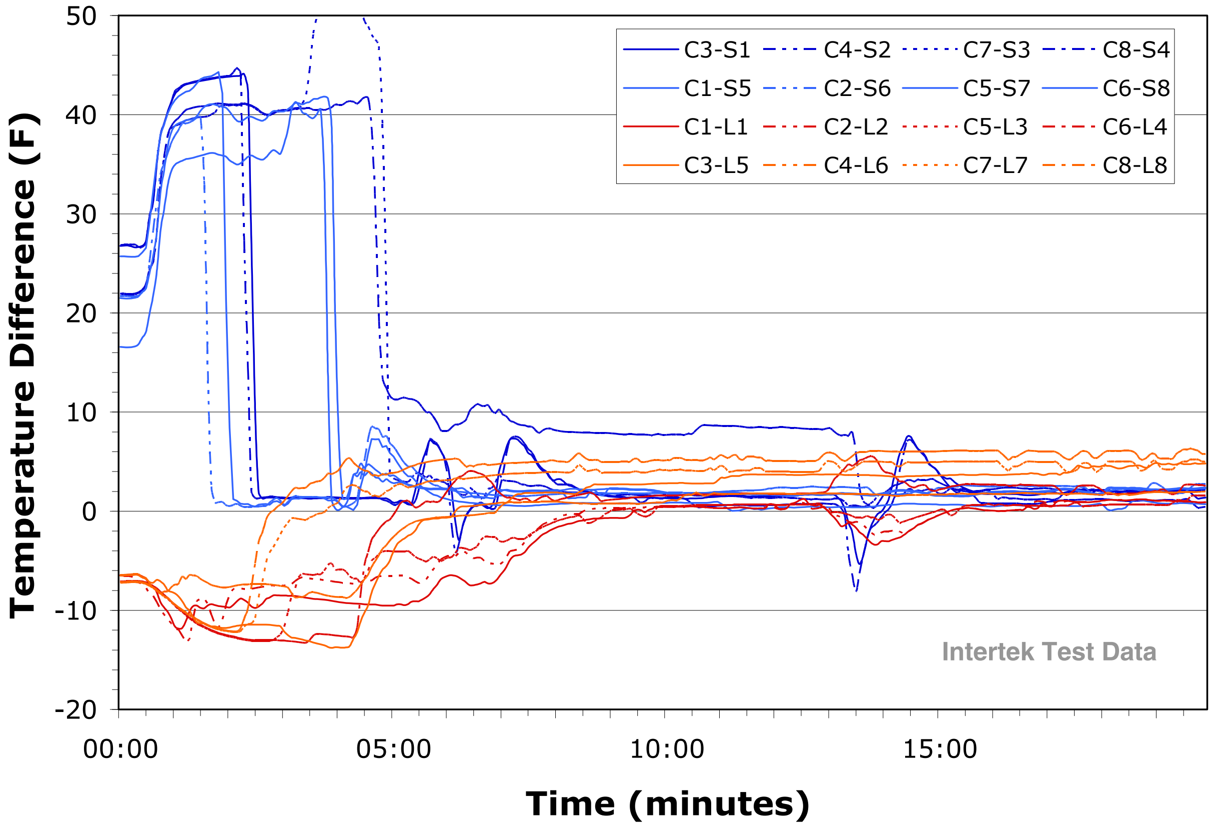


Table 104: Type-K Large Scissors Clamp C1-C8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
C1-C8 Liquid	104.1±0.09	104.1±0.09	107.9±0.08	105.6±0.01	104.6±0.06	106.2±0.08	109.2±0.07	110.3±0.07	106.5±0.15
L1-L8 Liquid	103.4±0.02	103.4±0.02	103.8±0.02	103.8±0.02	103.9±0.02	103.9±0.02	104.4±0.02	104.4±0.02	103.9±0.04
Diff. Liquid	0.7±0.09	0.7±0.08	4.2±0.09	1.8±0.02	0.7±0.07	2.3±0.07	4.8±0.07	5.9±0.06	**2.6±0.13**
C1-C8 Suction	69.4±0.05	71.1±0.08	71.4±0.05	70.5±0.05	75.9±0.09	75.9±0.09	76.2±0.08	76.6±0	73.4±0.19
L1-L8 Suction	68.8±0.04	68.8±0.04	69.4±0.1	69.4±0.1	73.9±0.06	73.9±0.06	74.4±0.03	74.4±0.03	71.6±0.25
Diff. Suction	0.6±0.04	2.4±0.05	2±0.08	1.2±0.11	2±0.05	2.1±0.05	1.8±0.07	2.2±0.03	**1.8±0.04**

Figure 54: Type-K Clamp Thermistor D1-D8 Temperature Difference at 95F Ambient

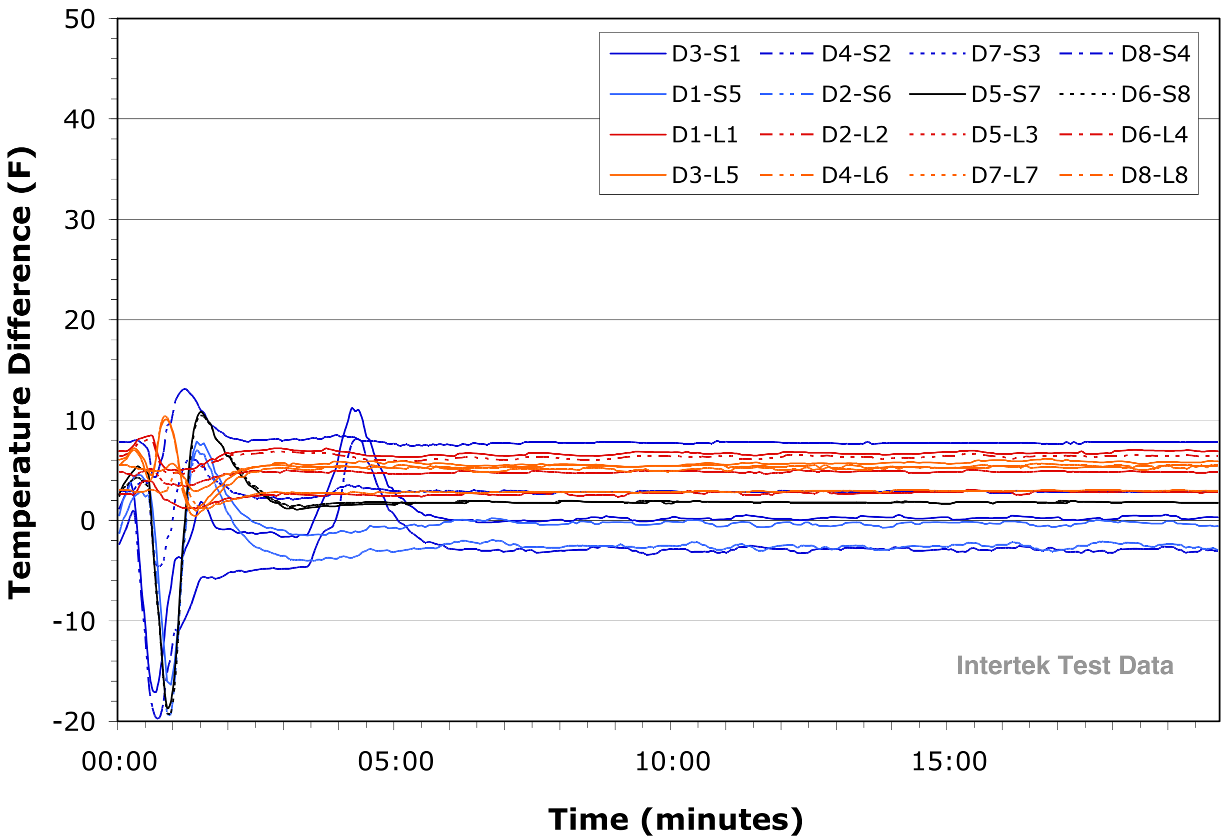


Table 105: Clamp Thermistor D1-DC8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
D1-D8 Liquid	105.5±0.02	105.1±0.01	104.4±0.01	103.9±0.02	108.9±0.01	106.9±0.01	107.1±0	109.5±0.02	106.4±0.13
L1-L8 Liquid	98.6±0.02	98.6±0.02	98.5±0.02	98.5±0.02	104±0.01	104±0.01	104.1±0.01	104.1±0.01	101.3±0.26
Diff. Liquid	6.8±0.03	6.4±0.02	5.9±0.02	5.4±0.03	4.9±0.01	2.8±0.02	3±0.01	5.3±0.02	**5.1±0.09**
D1-D8 Suction	68.5±0.04	66.3±0.05	69.3±0.03	66.2±0.05	81.2±0.02	81.2±0.02	80.1±0.01	85±0.02	74.7±0.5
L1-L8 Suction	68.8±0.04	68.8±0.04	69.1±0.04	69.1±0.04	79.4±0.02	79.4±0.02	77.2±0.01	77.2±0.01	73.6±0.46
Diff. Suction	-0.3±0.03	-2.5±0.05	0.3±0.03	-2.9±0.03	1.8±0.01	1.8±0.01	2.8±0.01	7.8±0.01	**1.1±0.21**

Figure 55: Type-K Scissors Clamp F1-F8 Temperature Difference at 95F Ambient

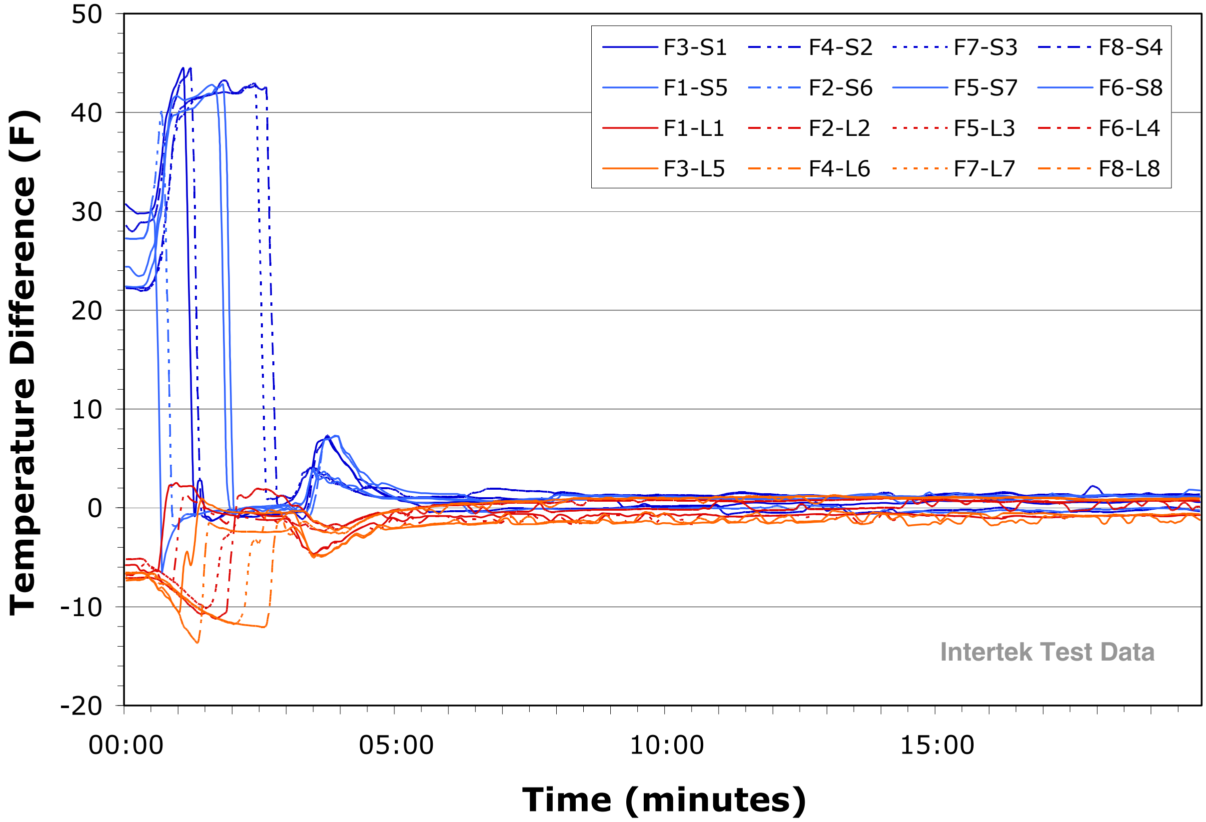


Table 106: Type-K Large Scissors Clamp F1-F8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
F1-F8 Liquid	102.5±0.0	102.5±0.01	102.1±0.07	102.5±0.02	104.5±0.0	104.2±0.07	104.5±0.0	104.6±0.0	103.4±0.07
L1-L8 Liquid	103.3±0.02	103.3±0.02	103.2±0.02	103.2±0.02	103.8±0.01	103.8±0.01	103.6±0.02	103.6±0.02	103.5±0.02
Diff. Liquid	-0.8±0.02	-0.9±0.02	-1.1±0.06	-0.7±0.03	0.7±0.01	0.4±0.07	0.9±0.02	1±0.02	**-0.1±0.06**
F1-F8 Suction	69.7±0.07	68.5±0.0	69.5±0.0	68.6±0.0	75.5±0.0	75.6±0.0	75.5±0.0	75.6±0.04	72.3±0.22
L1-L8 Suction	68.7±0.03	68.7±0.03	68.8±0.03	68.8±0.03	74.3±0.01	74.3±0.01	74.2±0.02	74.2±0.02	71.5±0.26
Diff. Suction	1±0.06	-0.2±0.03	0.7±0.03	-0.2±0.03	1.1±0.01	1.2±0.01	1.3±0.02	1.4±0.03	**0.8±0.04**

Figure 56: Type-K Insulated Bead Probes N1-N8 Temperature Difference at 95F Ambient

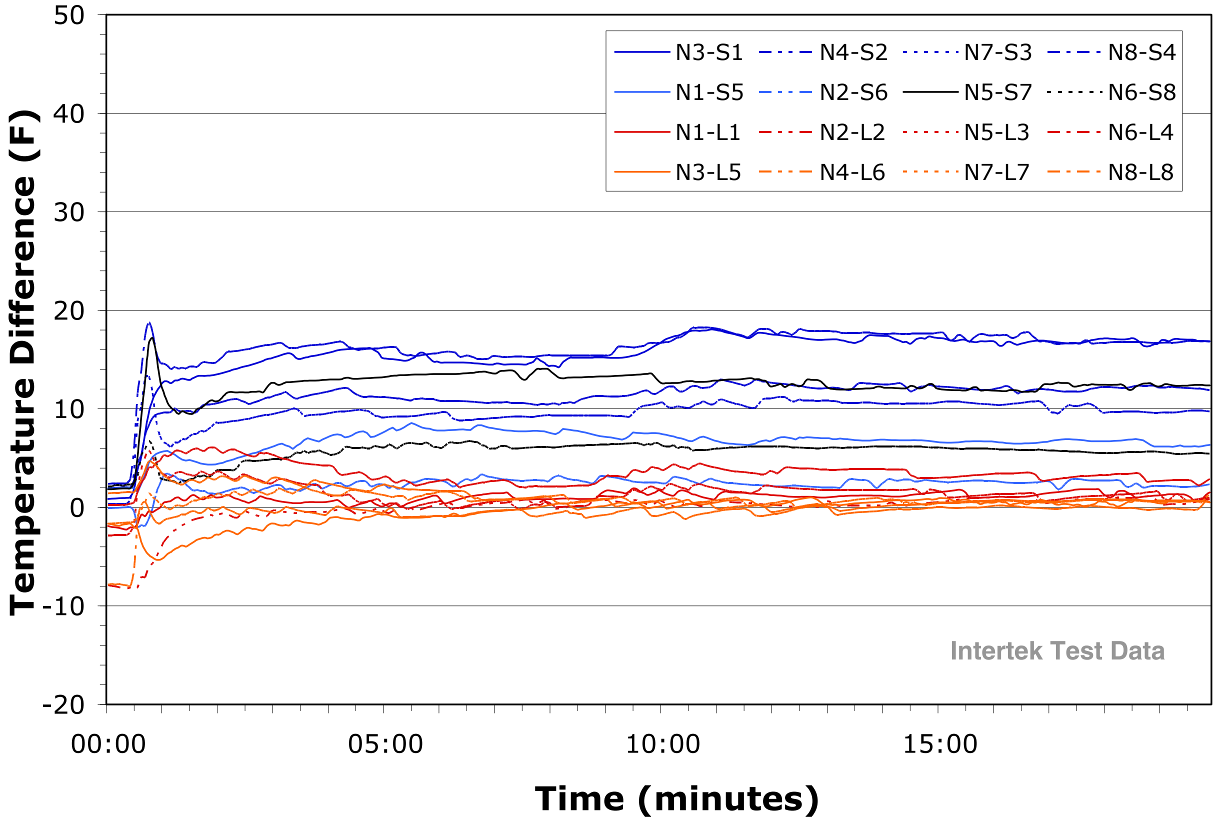


Table 107: Type-K Insulated Bead Probe N1-N8 Temperature Difference at 95F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
N1-N8 Liquid	106.2±0.14	105.6±0.14	103.1±0.05	102.6±0.03	106.7±0.13	108.6±0.16	103.5±0.02	103.6±0	105±0.14
L1-L8 Liquid	104.8±0.12	104.8±0.12	102.6±0.03	102.6±0.03	105.5±0.12	105.5±0.12	102.9±0.02	102.9±0.02	104±0.13
Diff. Liquid	1.3±0.06	0.8±0.05	0.5±0.04	0±0.03	1.2±0.05	3.1±0.06	0.6±0.02	0.7±0.02	**1±0.06**
N1-N8 Suction	75.3±0.08	71.1±0.09	85.6±0.09	80.8±0.09	89.1±0.08	82.6±0.0	85±0.09	91.8±0.07	82.7±0.44
L1-L8 Suction	68.7±0.04	68.7±0.04	68.8±0.09	68.8±0.09	76.8±0.04	76.8±0.04	74.8±0.02	74.8±0.02	72.3±0.35
Diff. Suction	6.6±0.05	2.5±0.05	16.9±0.04	12.1±0.03	12.2±0.06	5.7±0.04	10.1±0.08	17±0.07	**10.4±0.33**

Figure 57: Type-K Linear Clamp A1-A8 Temperature Difference at 115F Ambient

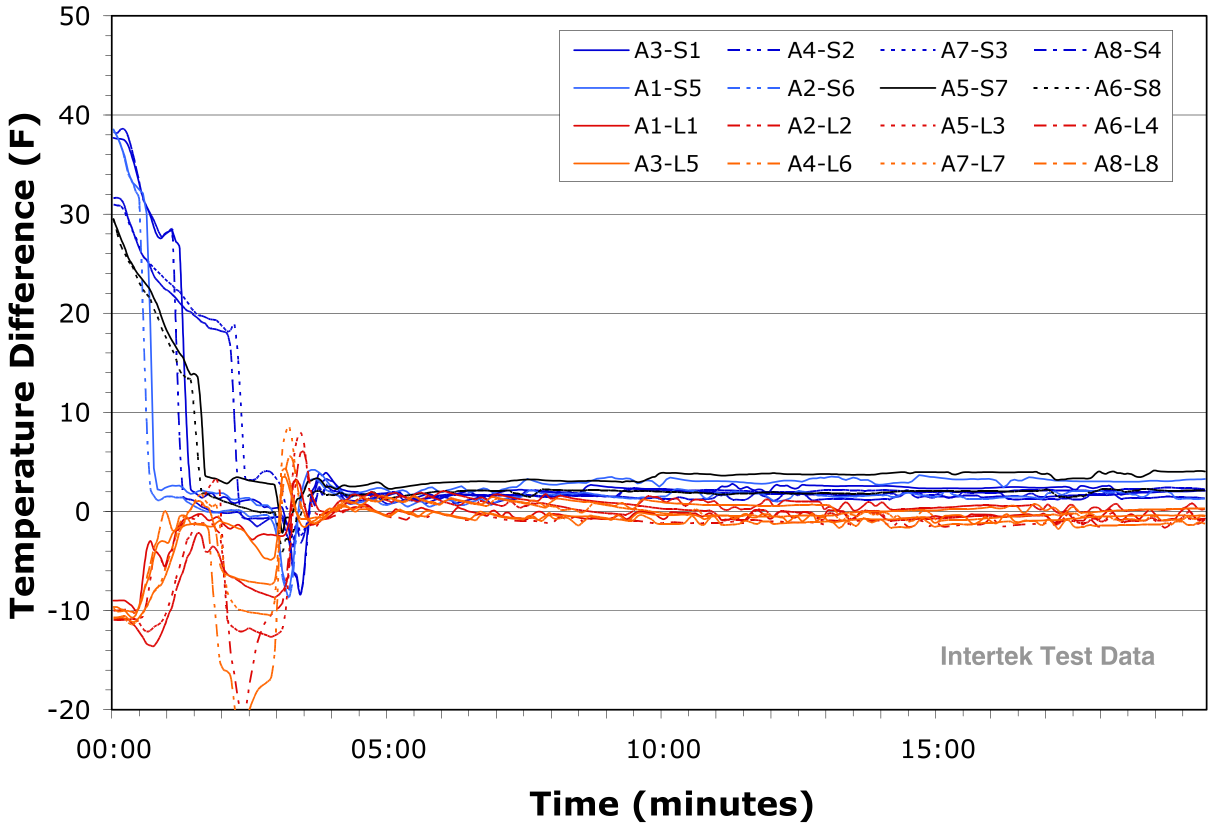


Table 108: Type-K Linear Clamp A1-A8 Temperature Difference at 115F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
A1-A8 Liquid	126.1±0.08	125.4±0.05	125.6±0.06	125±0.09	127±0.09	128.1±0.08	127.1±0.08	128±0.09	126.5±0.08
L1-L8 Liquid	126.5±0.06	126.5±0.06	126.3±0.07	126.3±0.07	127.7±0.05	127.7±0.05	127.7±0.07	127.7±0.07	127.1±0.07
Diff. Liquid	-0.5±0.06	-1.1±0.05	-0.6±0.05	-1.3±0.04	-0.7±0.05	0.4±0.04	-0.6±0.04	0.2±0.04	**-0.5±0.04**
A1-A8 Suction	80.4±0.03	78.9±0.08	80.1±0.09	79.6±0.01	90.2±0.07	88.4±0.05	88.5±0.0	87.8±0.07	84.3±0.31
L1-L8 Suction	77.3±0.03	77.3±0.03	77.9±0.05	77.9±0.05	86.4±0.02	86.4±0.02	86.3±0.03	86.3±0.03	82±0.42
Diff. Suction	3.1±0.03	1.7±0.06	2.3±0.05	1.7±0.05	3.8±0.05	2±0.04	2.1±0.03	1.5±0.05	**2.3±0.05**

Figure 58: Clamp Thermistor D1-D8 Temperature Difference at 115F Ambient

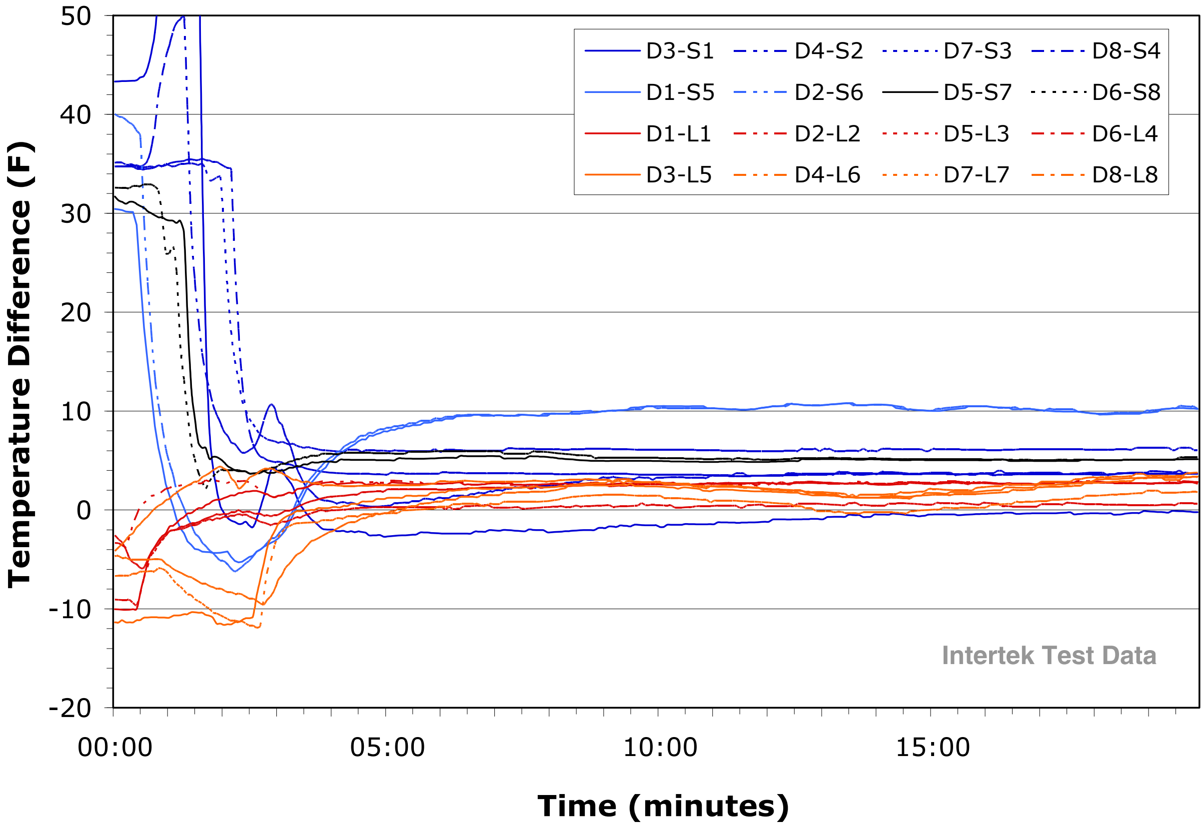


Table 109: Clamp Thermistor D1-D8 Temperature Difference at 115F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
D1-D8 Liquid	127.2±0.05	127.2±0.05	126.9±0.06	127.2±0.06	128.1±0.04	130.3±0.05	127.8±0.06	129.2±0.06	128±0.08
L1-L8 Liquid	124.4±0.04	124.4±0.04	124.3±0.05	124.3±0.05	127.6±0.04	127.6±0.04	126.8±0.06	126.8±0.06	125.8±0.14
Diff. Liquid	2.7±0.01	2.8±0.02	2.6±0.11	2.9±0.11	0.6±0.02	2.7±0.01	1.1±0.12	2.4±0.12	**2.2±0.06**
D1-D8 Suction	91.9±0.02	92±0.02	81.8±0.03	85.9±0.03	92.8±0.01	92.8±0.01	93.4±0.02	90.9±0.01	90.2±0.26
L1-L8 Suction	81.9±0.03	81.9±0.03	82.1±0.04	82.1±0.04	87.7±0.01	87.7±0.01	87.3±0	87.3±0	84.8±0.26
Diff. Suction	10.1±0.05	10.2±0.05	-0.3±0.02	3.7±0.02	5±0.01	5.1±0.01	6.1±0.01	3.6±0.01	**5.4±0.22**

Figure 59: Worst Position Clamp Thermistor D1-D8$_w$ Temperature Difference at 115F Ambient

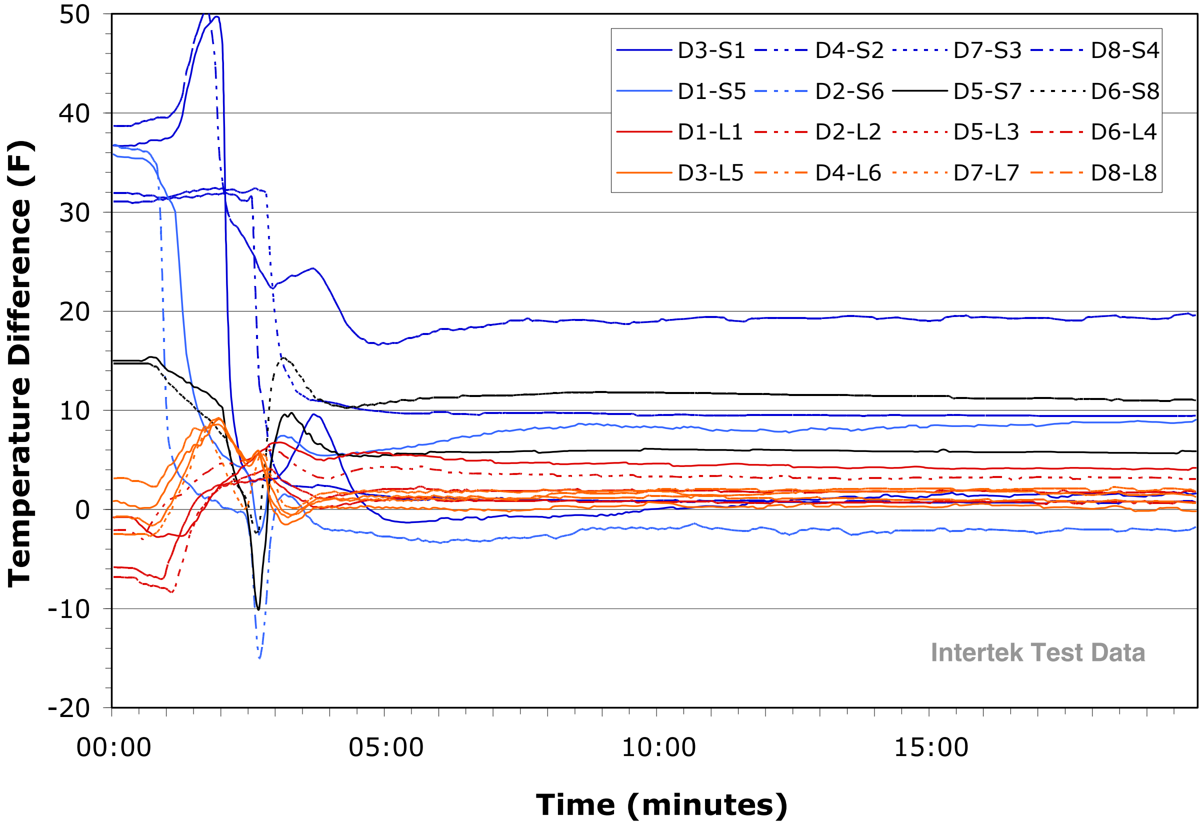


Table 110: Worst Position Clamp Thermistor D1-D8$_w$ Temperature Difference at 115F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
D1-D8$_w$ Liquid	128.1±0.01	127.2±0.01	104.3±0.04	105.7±0.04	128.2±0.02	127.2±0.02	110.7±0.03	109.5±0.04	117.6±0.69
L1-L8 Liquid	124±0.01	124±0.01	104.1±0.03	104.1±0.03	126.4±0	126.4±0	108.7±0.02	108.7±0.02	115.8±0.92
Diff. Liquid	4.1±0.01	3.2±0.01	0.2±0.04	1.7±0.03	1.8±0.02	0.8±0.01	2±0.02	0.9±0.02	**1.8±0.08**
D1-D8$_w$ Suction	77.3±0.03	66.6±0.02	83.1±0.03	101±0.02	90.1±0.01	95.5±0.01	97.3±0	88.6±0.02	87.4±0.72
L1-L8 Suction	68.7±0.02	68.7±0.02	81.6±0.02	81.6±0.02	84.3±0.01	84.3±0.01	87.8±0.01	87.8±0.01	80.6±0.69
Diff. Suction	8.7±0.03	-2.1±0.02	1.5±0.02	19.4±0.03	5.8±0.01	11.2±0.02	9.5±0.01	0.7±0.01	**6.8±0.44**

Figure 60: Insulated 10K Thermistor 3/16" Dia. K1-K8 Temperature Difference at 115F Ambient

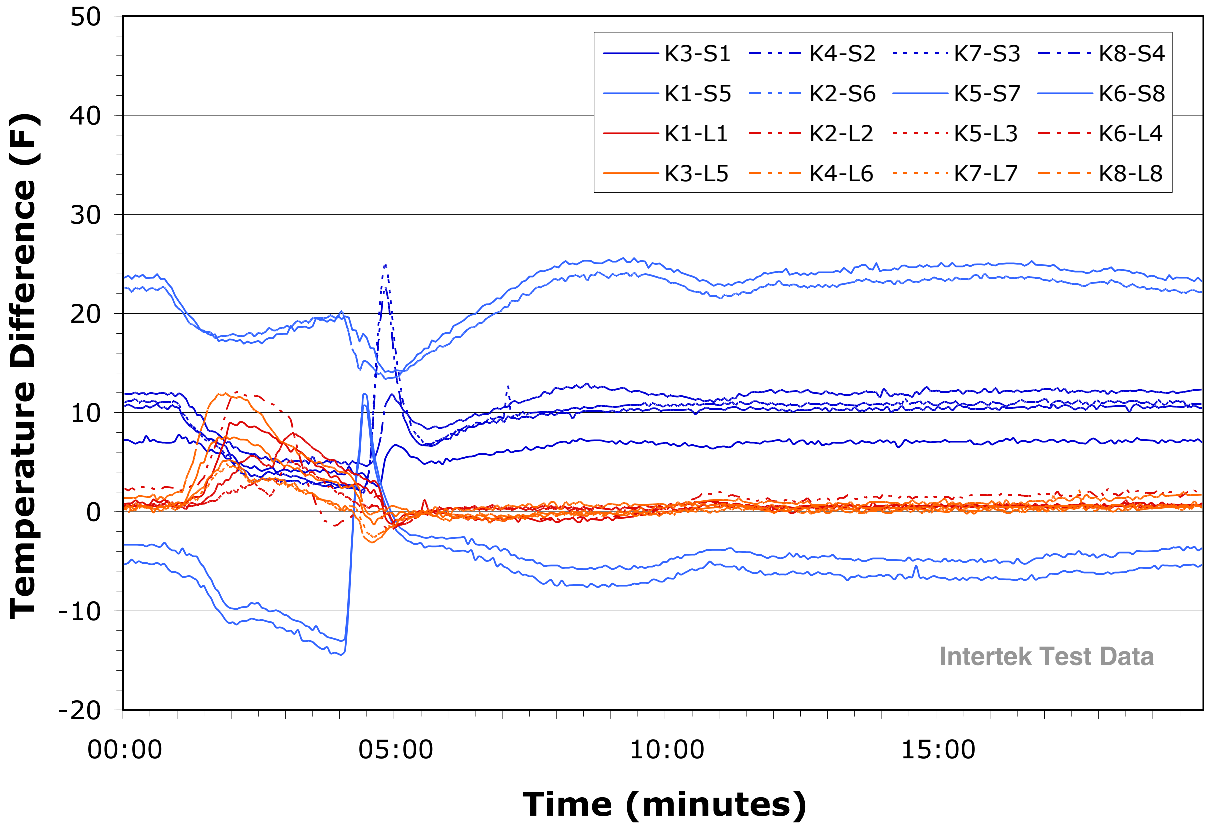


Table 111: Insulated 10K Cylindrical Thermistor 3/16" Dia. K1-K8 Temperature Difference at 115F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
K1-K8 Liquid	126.1±0.03	127.2±0.05	129.4±0.05	130.3±0.03	126.9±0.03	127±0.03	130.6±0.07	130.8±0.04	128.5±0.12
L1-L8 Liquid	125.4±0.01	125.4±0.01	129.1±0.06	129.1±0.06	126.3±0.01	126.3±0.01	130.3±0.03	130.3±0.03	127.8±0.19
Diff. Liquid	0.6±0.02	1.8±0.04	0.3±0.04	1.2±0.07	0.6±0.02	0.6±0.03	0.4±0.06	0.6±0.02	**0.8±0.03**
K1-K8 Suction	99.5±0.03	98.2±0.03	82±0.03	87±0.02	84.6±0.09	82.8±0.1	98.5±0.03	98.1±0.03	91.3±0.5
L1-L8 Suction	75.1±0.1	75.1±0.1	74.9±0.03	74.9±0.03	89±0.01	89±0.01	87.5±0.01	87.5±0.01	81.6±0.64
Diff. Suction	24.4±0.11	23.1±0.1	7.1±0.03	12.2±0.02	-4.4±0.09	-6.2±0.1	11±0.03	10.5±0.03	**9.7±0.7**

Figure 61: Worst Position Insulated 10K Cylindrical Thermistor 3/16" Dia. K1-K8$_w$ Temperature Difference at 115F Ambient

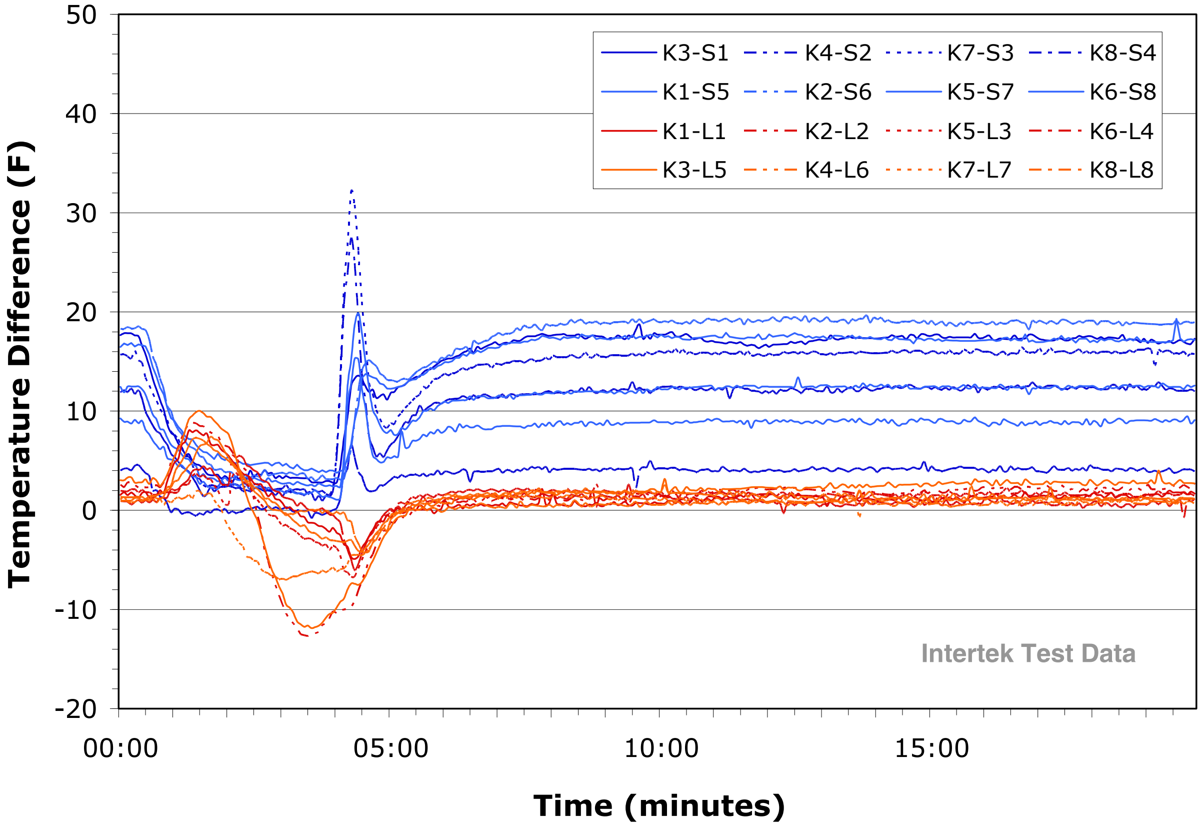


Table 112: Worst Position Insulated 10K Cylindrical Thermistor 3/16" Dia. K1-K8$_w$ Temperature Difference at 115F Ambient

Description	Sensor 1 F	Sensor 2 F	Sensor 3 F	Sensor 4 F	Sensor 5 F	Sensor 6 F	Sensor 7 F	Sensor 8 F	Average F
K1-K8$_w$ Liquid	127.9±0.04	128.6±0.06	128.4±0.05	126.8±0.04	129.8±0.06	128.8±0.08	127.7±0.05	128.2±0.08	128.3±0.06
L1-L8 Liquid	126.5±0.06	126.5±0.06	125.6±0.04	125.6±0.04	128.1±0.05	128.1±0.05	127±0.05	127±0.05	126.8±0.09
Diff. Liquid	1.4±0.05	2.2±0.05	2.8±0.04	1.1±0.03	1.6±0.04	0.7±0.05	0.7±0.04	1.2±0.05	**1.5±0.05**
K1-K8$_w$ Suction	95.5±0.06	97.3±0.04	96.3±0.04	83.1±0.05	101.2±0.04	97.7±0.04	103.7±0.05	100.1±0.04	96.8±0.39
L1-L8 Suction	78.4±0.03	78.4±0.03	79±0.05	79±0.05	88.7±0.03	88.7±0.03	87.7±0.02	87.7±0.02	83.5±0.46
Diff. Suction	17.2±0.06	18.9±0.03	17.3±0.05	4.1±0.03	12.4±0.03	8.9±0.04	15.9±0.04	12.3±0.04	**13.4±0.32**

4.5.3 Supply and Return Air Measurement Instrument Tests

Table 113 provides a summary of the supply and return air measurement instruments. There are approximately 93 instruments or sensors to test from 5 manufacturers. Tests were performed with proper factory charge at the following conditions 80DB/67WB/95F and 85DB/80WB/115F.

Table 113: Supply and Return Air Measurement Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Digital Capacitance RH (WB) NTC (DB)	10	P1-P10	3	Bottom of duct, insul, return
2	Digital Capacitance RH (WB) NTC (DB)	5	Q1-Q10	3	Bottom of duct, insul, return
3	Digital Capacitance RH (WB) NTC (DB)	10	R1-R10	3	Bottom of duct, insul, return
4	Digital Capacitance RH (WB) NTC (DB)	5	S1-S5	3	Bottom of duct, insul, return
5	Digital Capacitance RH (WB) NTC (DB)	5	U1-U5	3	Bottom of duct, insul, return
6	Digital Capacitance RH (WB) NTC (DB)	5	V1-V5	3	Bottom of duct, insul, return
7	Digital Capacitance RH (WB) NTC (DB)	10	W1-W10	3	Bottom of duct, insul, return
8	Digital Capacitance RH (WB) NTC (DB)	5	V1-V5	3	Bottom of duct, insul, return
9	K-Type with wet wick	20	N1-N20	3	Dirty wick and muddy waters
10	K-Type with wet wick	10	O1-O10	3	Dirty wick and muddy waters
11	NTC with wet wick	3	Z1-Z3	3	Dirty wick and muddy waters

4.5.4 Pressure Measurement Instrument Tests

Table 114 provides a summary of the pressure measurement instruments used by technicians in the field. There were approximately 63 instruments or sensors from 8 manufacturers. Tests were performed at five discharge and suction pressures (DP/SP) in pounds per square inch gauge (psig): 1) R22 low pressure (190DP/35SP), 2) R22 average pressure (270DP/70SP), 3) R22 high and R410 low pressure (320DP/105SP), 4) R410A average pressure (390DP/120SP), and 5) R410A high pressure (470DP/125SP). Two digital pressure manifolds were found to be leaking refrigerant when taken out-of-the-box. These were removed from the sample. Laboratory tests of 15 digital and 7 analog field pressure measurement instruments have been completed. Digital pressure measurement out-of-box test results are provided in **Table 115**. Analog pressure measurement out-of-box test results are provided in **Table 116**. The average difference between laboratory and digital pressure measurement instruments is 0.57 +/- 0.24% based on measurements at ten different pressures with 15 instruments from 6 manufacturers. The average difference between laboratory and analog pressure measurement instruments is 1.76 +/- 0.57% based on measurements at ten different pressures with 7 instruments from 2 manufacturers.

Table 114: Digital and Analog Pressure Measurement Instruments

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Digital w/2 K-type clamps	5	AO1-AO5	Out of box	EPA hoses w/Refrig. hot chamber
2	Digital w/vacuum + 2 K-clamps	2	AP1-AP2	Out of box	EPA hoses w/Refrig. hot chamber
3	Digital w/vacuum + 2 K-clamps	2	AQ1-AQ2	Out of box	EPA hoses w/Refrig. hot chamber
4	Digital w/2 NTC clamps	3	AR1-AR3	Out of box	EPA hoses w/Refrig. hot chamber
5	Digital w/2 NTC clamps + logger	3	AS1-AS3	Out of box	EPA hoses w/Refrig. hot chamber
6	Digital w/2 NTC clamps	5	AT1-AT5	Out of box	EPA hoses w/Refrig. hot chamber
7	Digital	15	AU1-AU15	Out of box	EPA hoses w/Refrig. hot chamber
8	Digital w/1 K-type clamps	1	AV1	Out of box	EPA hoses w/Refrig. hot chamber
9	Digital w/2 K-type clamps	2	BKl-BK2	Out of box	EPA hoses w/Refrig. hot chamber
10	Digital w/2 K-type clamps	2	BL1-BL2	Out of box	EPA hoses w/Refrig. hot chamber
11	Analog	5	AW1-AW5	Out of box	EPA hoses w/Refrig. hot chamber
12	Analog	5	AX1-AX5	Out of box	EPA hoses w/Refrig. hot chamber
13	Analog	5	AY1-AY5	Out of box	EPA hoses w/Refrig. hot chamber
14	Analog	5	AZ1-AZ5	Out of box	EPA hoses w/Refrig. hot chamber
15	Analog	2	BA1-BA2	Out of box	EPA hoses w/Refrig. hot chamber

Table 115: Digital Pressure Measurement Out-of-Box Test Results

Sensor	R22 Low SP psig	R22 Base SP psig	R22 High SP psig	R410A Base SP psig	R410A High SP psig	R22 Low DP psig	R22 Base DP Psig	R410A Low DP psig	R410A Base DP psig	R410A High DP psig
AO2	35.8	70.8	106.6	121.8	146.5	191.5	270.7			
Intertek	35.2	69.9	105.6	120.9	145.6	190.7	269.9			
Difference	**-1.8%**	**-1.2%**	**-0.9%**	**-0.7%**	**-0.6%**	**-0.4%**	**-0.3%**			
AP1	35.3	71.0	105.6	120.8	145.8	190.9	270.5	320.6	390.6	469.8
Intertek	35.2	70.8	105.2	120.4	145.5	191.0	270.6	320.6	390.7	470.1
Difference	**-0.4%**	**-0.3%**	**-0.4%**	**-0.3%**	**-0.2%**	**0.0%**	**0.0%**	**0.0%**	**0.0%**	**0.1%**
AQ1	35.4	71.1	105.6	120.4	145.4	191.0	270.4	320.5	390.5	469.6
Intertek	35.2	70.8	105.2	120.4	145.4	191.0	270.6	320.6	390.8	470.1
Difference	**-0.7%**	**-0.5%**	**-0.4%**	**0.0%**	**0.0%**	**0.0%**	**0.1%**	**0.0%**	**0.1%**	**0.1%**
AR1	35.4	71.1	105.3	120.7	144.9	190.0	268.8	317.9	387.8	466.9
Intertek	35.2	70.8	105.1	120.4	145.2	190.9	270.5	320.1	390.5	470.0
Difference	**-0.7%**	**-0.5%**	**-0.2%**	**-0.2%**	**0.2%**	**0.5%**	**0.6%**	**0.7%**	**0.7%**	**0.7%**
AR2	35.4	70.5	104.9	119.8	145.8	190.5	270.9	320.0	392.5	472.4
Intertek	35.4	70.5	105.2	120.0	145.5	190.3	270.2	319.9	391.0	471.1
Difference	**0.0%**	**0.0%**	**0.3%**	**0.2%**	**-0.2%**	**-0.1%**	**-0.3%**	**0.0%**	**-0.4%**	**-0.3%**
AS1	38.7	70.9	105.2	120.5	146.1	191.7	271.8	322.0	392.0	471.2
Intertek	35.2	70.8	104.9	120.3	145.5	190.5	270.5	320.6	390.8	470.5
Difference	**-10.0%**	**-0.2%**	**-0.3%**	**-0.2%**	**-0.4%**	**-0.7%**	**-0.5%**	**-0.4%**	**-0.3%**	**-0.1%**
AS2	36.0	71.2	105.6	120.0	146.1	191.3	271.2	320.8	392.0	471.8
Intertek	35.4	70.5	105.0	120.0	145.5	190.4	270.2	319.8	391.2	471.2
Difference	**-1.7%**	**-1.0%**	**-0.6%**	**0.0%**	**-0.4%**	**-0.5%**	**-0.4%**	**-0.3%**	**-0.2%**	**-0.1%**
AT1	35.3	71.1	105.2	120.4	145.6	191.0	271.0	320.0	391.0	470.0
Intertek	35.2	70.7	104.9	120.3	145.3	191.1	270.5	320.6	390.8	470.8
Difference	**-0.3%**	**-0.5%**	**-0.2%**	**-0.1%**	**-0.2%**	**0.1%**	**-0.2%**	**0.2%**	**-0.1%**	**0.2%**
AT2	35.8	70.9	104.9	120.0	145.6	197.0	278.0	327.0	397.0	476.0
Intertek	35.4	70.5	104.8	120.0	145.5	190.3	270.2	319.8	391.2	471.2
Difference	**-1.1%**	**-0.6%**	**-0.1%**	**0.0%**	**0.0%**	**-3.5%**	**-2.9%**	**-2.3%**	**-1.5%**	**-1.0%**
AU1	34.0	71.0	104.0	120.0	145.0	193.0	270.0	320.0	391.0	472.0
Intertek	35.2	70.7	104.9	120.2	145.5	191.1	270.5	320.6	390.8	470.4
Difference	**3.4%**	**-0.4%**	**0.9%**	**0.2%**	**0.3%**	**-1.0%**	**0.2%**	**0.2%**	**-0.1%**	**-0.3%**
AV1	37.5	72.5	107.0	122.0	146.0	191.0	273.0	322.5	394.0	453.0
Intertek	35.4	70.5	104.7	120.0	145.7	190.3	270.1	319.9	391.3	470.4
Difference	**-5.9%**	**-2.8%**	**-2.2%**	**-1.7%**	**-0.2%**	**-0.4%**	**-1.1%**	**-0.8%**	**-0.7%**	**3.7%**
BK1	35.0	71.0	105.0	120.0	151.0	194.0	270.0	320.0	390.0	456.0
Intertek	35.2	70.7	104.9	120.2	145.5	190.6	270.9	320.9	391.0	470.4
Difference	**0.6%**	**-0.5%**	**-0.1%**	**0.1%**	**-3.8%**	**-1.8%**	**0.3%**	**0.3%**	**0.3%**	**3.1%**
BK2	35.0	70.0	105.0	120.0	146.0	191.0	271.0	321.0	391.0	471.0
Intertek	35.4	70.5	104.5	120.0	145.8	190.3	270.1	319.8	390.5	470.4
Difference	**1.1%**	**0.7%**	**-0.5%**	**0.0%**	**-0.2%**	**-0.4%**	**-0.3%**	**-0.4%**	**-0.1%**	**-0.1%**
BL1	39.0	77.0	105.0	120.0	159.0	190.0	270.0	320.0	389.0	472.0
Intertek	35.2	70.6	104.8	120.2	145.5	190.7	270.8	321.0	390.9	470.3
Difference	**-10.8%**	**-9.0%**	**-0.2%**	**0.2%**	**-9.3%**	**0.3%**	**0.3%**	**0.3%**	**0.5%**	**-0.4%**
BL2	36.0	71.0	105.0	121.0	146.0	606.0	607.0	320.0	391.0	494.0
Intertek	35.4	70.5	104.5	120.0	145.7	190.3	261.0	320.4	390.4	470.4
Difference	**-1.7%**	**-0.7%**	**-0.5%**	**-0.8%**	**-0.2%**	**-218.4%**	**-132.6%**	**0.1%**	**-0.2%**	**-5.0%**
Average	36.0	71.4	105.3	120.5	147.0	219.3	293.6	320.9	391.4	470.4
Intertek	35.3	70.6	105.0	120.2	145.5	190.6	269.8	320.3	390.8	470.5
Difference	**-2.0%**	**-1.2%**	**-0.4%**	**-0.2%**	**-1.0%**	**-15.1%**	**-8.8%**	**-0.2%**	**-0.1%**	**0.02%**

Table 116: Analog Pressure Measurement Out-of-Box Test Results

Sensor	R22 Low SP psig	R22 Base SP psig	R22 High SP Psig	R410A Base SP psig	R410A High SP psig	R22 Low DP psig	R22 Base DP Psig	R410A Low DP psig	R410A Base DP psig	R410A High DP psig
AW1	35.0	70.0	105.0	124.0	150.0	190.0	270.0	329.0	410.0	490.0
Intertek	35.2	70.6	104.8	120.1	145.3	191.1	270.6	320.6	390.9	469.8
Difference	**0.6%**	**0.9%**	**-0.2%**	**-3.2%**	**-3.2%**	**0.6%**	**0.2%**	**-2.6%**	**-4.9%**	**-4.3%**
AW2	40.0	76.0	110.0	121.0	145.0	190.0	270.0	321.0	380.0	480.0
Intertek	35.4	70.5	104.4	120.0	145.7	190.3	269.9	320.2	391.4	470.4
Difference	**-13.0%**	**-7.8%**	**-5.4%**	**-0.8%**	**0.5%**	**0.2%**	**0.0%**	**-0.3%**	**2.9%**	**-2.0%**
AY1	35.0	70.0	105.0	125.0	145.0	190.0	270.0	315.0	385.0	465.0
Intertek	35.2	70.6	104.7	120.1	145.5	191.1	270.6	320.7	390.9	470.1
Difference	**0.6%**	**0.9%**	**-0.3%**	**-4.1%**	**0.3%**	**0.6%**	**0.2%**	**1.8%**	**1.5%**	**1.1%**
AY2	35.0	70.0	110.0	120.0	145.0	190.0	270.0	320.0	395.0	470.0
Intertek	35.5	70.5	104.2	120.0	145.7	190.4	269.9	320.2	400.0	470.4
Difference	**1.4%**	**0.7%**	**-5.6%**	**0.0%**	**0.5%**	**0.2%**	**-0.1%**	**0.1%**	**1.3%**	**0.1%**
AZ1	35.0	70.0	112.0	126.0	150.0	190.0	270.0	320.0	400.0	470.0
Intertek	35.3	70.6	104.6	120.0	145.5	191.1	270.5	320.7	390.9	470.1
Difference	**0.7%**	**0.8%**	**-7.1%**	**-5.0%**	**-3.1%**	**0.6%**	**0.2%**	**0.2%**	**-2.3%**	**0.0%**
AZ2	39.0	75.0	110.0	121.0	150.0	200.0	275.0	320.0	395.0	470.0
Intertek	35.5	70.5	104.1	120.0	145.8	190.4	269.9	320.2	390.4	470.4
Difference	**-9.9%**	**-6.4%**	**-5.7%**	**-0.8%**	**-2.9%**	**-5.1%**	**-1.9%**	**0.1%**	**-1.2%**	**0.1%**
BA1	35.0	75.0	104.0	125.0	150.0	190.0	270.0	325.0	390.0	470.0
Intertek	35.3	70.6	104.5	120.0	145.5	191.1	270.5	320.8	390.9	470.2
Difference	**0.7%**	**-6.2%**	**0.5%**	**-4.2%**	**-3.1%**	**0.6%**	**0.2%**	**-1.3%**	**0.2%**	**0.0%**
Average	36.5	72.7	107.3	122.7	147.5	191.7	270.8	321.7	392.5	474.2
Intertek	35.3	70.6	104.4	120.0	145.6	190.8	270.2	320.4	392.4	470.2
Difference	**-3.3%**	**-3.0%**	**-2.8%**	**-2.2%**	**-1.3%**	**-0.5%**	**-0.2%**	**-0.4%**	**-0.02%**	**-0.84%**

4.5.5 Airflow Measurement Instrument Tests

Table 117 provides a summary of the airflow measurement instruments. There are approximately 25 instruments or sensors available from 8 manufacturers. Tests can be performed with airflow of 2,000, 2,500, and 3000 cfm (3 tests) at the following conditions 80DB/67WB/95F. The worst case measurements can be performed with airflow measurements taken at non standard locations. Laboratory tests of pitot-tube array airflow measurements from 1 manufacturer (AC1) are shown in **Table 118**. The difference between laboratory and field measurement instruments is 10.3 +/- 1.1% based on three measurements at 2,000, 2,500, and 3,000 cfm.

Table 117: Airflow Measurement Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Digital Vane Anemometer Data Logger	1	AA1	8	Non standard locations
2	Digital Vane Anemometer	1	AB1	8	Non standard locations
3	Pitot-Tube Array Airflow Grid	1	AC1	8	Non standard locations
4	Fan Powered Flow Hood	1	AD1	8	Non standard locations
5	Balometer hot-wire anemometer flow hood	1	AE1-AE3	8	Non standard locations
6	Mini-vane anemometer	2	AF1-AF2	8	Non standard locations
7	Vane anemometer	2	AG-1AG2	8	Non standard locations
8	Hot Wire Anemometer	1	AH1	8	Non standard locations
9	Integral Differential Pressure.	1	AI1	8	Non standard locations
10	Hot Wire Anemometer	2	AJ1-AJ2	8	Non standard locations
11	Mini-vane anemometer	2	AK1-AK2	8	Non standard locations
12	Hot-wire anemometer	2	AL1-AL2	8	Non standard locations
13	Micro-manometer	2	AM1-AM2	8	Non standard locations
14	Hot Wire Anemometer	1	AN1	8	Non standard locations
15	Digital Diff. Pressure Sensor	1	Veris1	8	Non standard locations
16	Digital Thermal Airflow Sensor	1	Veris2	8	Non standard locations

Table 118: Airflow Measurement Instrument Tests of Pitot-Tube Array Flow Grid (AC1)

Description	Intertek Airflow cfm	Intertek Airflow cfm/ton	Pitot-Tube Array Airflow cfm	Pitot-Tube Array Airflow cfm/ton	Difference
AC1	2,000	267	2,222	296	11.1%
AC1	2,500	333	2,770	369	10.8%
AC1	3,000	400	3,270	436	9.0%

4.5.6 Vacuum Pump Measurement Instrument Tests

Table 119 provides a summary of vacuum pump measurement instruments. There are 4 vacuum pumps and 3 micron gauges available from 7 manufacturers. The worst case measurements can be performed with no vacuum/liquid drier, 30 minute vacuum with drier, and 60 minute vacuum with drier. Tests can be performed with airflow at approximately 3000 cfm at the following conditions 80DB/67WB/95F to evaluate the efficiency impact associated with each evacuation method.

Table 119: Vacuum Pump Measurement Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Vacuum Pump 10 cfm, 2-stage	1	VP1	1	30 min and 60 min vacuum
2	Vacuum Pump 8 cfm, 2-stage	1	VP2	1	30 min and 60 min vacuum
3	Vacuum Pump 8 cfm 2-stage	1	VP3	1	30 min and 60 min vacuum
4	Vacuum Pump 5 cfm	1	VP4	1	30 min and 60 min vacuum
5	Digital Micron Gauge	1	VP5	1	30 min and 60 min vacuum
6	Digital Micron Gauge	1	VP6	1	30 min and 60 min vacuum
7	Digital Micron Gauge	1	VP7	1	30 min and 60 min vacuum

4.5.7 Fan Belt Tension and Alignment Measurement Instrument Tests

Table 120 provides a summary of the fan belt tension and alignment measurement instruments. There are 14 belt tension and alignment instruments available from 5 manufacturers. Fan belt tension and alignment measurement instruments tests can be performed with airflow at approximately 3000 cfm. Belts can be tested with proper tension and alignment, as well as loose and tight tension and misalignment of 0.25 and 0.375 inches at the following conditions 80DB/67WB/95F. The worst case measurements can be performed with fan belt tension either loose or tight and the belt misaligned by either ¼ or 3/8 inches. Out-of-box fan belt tests indicated tension was looser than manufacturer recommendations. Out-of-box fan belt alignment tests indicated the belt was properly aligned.

Table 120: Fan Belt Tension and Alignment Measurement Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Digital Force Gauge	2	BD1-BD2	1	6 tests (loose, tight, misaligned)
2	Belt Tension Checker 1302546	3	BE1-BE3	1	6 tests (loose, tight, misaligned)
3	Belt Tensiometer 102761	3	BF1-BF3	1	6 tests (loose, tight, misaligned)
4	Belt Tension Finder, 108039-A	3	BG1-BG3	1	6 tests (loose, tight, misaligned)
5	Digital Laser Alignment Tool	1	BH1	1	6 tests (loose, tight, misaligned)
6	Digital Sonic Tension Tool	1	BI1	1	6 tests (loose, tight, misaligned)
7	Digital Force Gauge	2	BD1-BD2	1	6 tests (loose, tight, misaligned)

4.5.8 Cold Weather Charging Hood and Digital Refrigerant Scale Measurement Instrument Tests

Table 121 provides a summary of the cold weather charging hood and wireless digital refrigerant scale. There are 4 instruments available from 1 manufacturer. Two tests can be performed at the following conditions 80DB/67WB/95F. The worst case measurements can be performed with cold weather conditions down to 55F and inaccurate measurements and check scale with known weights or 1, 5, 10, 15, 25, 50, and 100 pounds to +/- 0.25 ounces.

Table 121: Cold Weather Charging Hood and Wireless Scale Instrument Tests

	Description	Qty	Labels	Best Tests	Worst Case Tests
1	Wireless Digital Refrigerant Scale	2	BB1-BB2	1	Inaccurate
2	Low Temperature Charging Hood	2	BC1-BC2	1	Doesn't fit condenser

5 Using Laboratory Data for Load Impact Evaluations

Laboratory tests were performed to support load impact evaluations of ratepayer-funded HVAC Maintenance and Installation programs. Tests were performed to evaluate the application energy efficiency impacts of HVAC maintenance faults, fault detection diagnostic (FDD) and instrumentation accuracy, and improve the Database for Energy Efficiency Resources (DEER). One of the primary purposes of laboratory tests is to provide field measurement protocols and analytical methods to evaluate the application energy efficiency impacts of individual and multiple HVAC maintenance measures. This section provides information regarding how to use laboratory data for load impact evaluations of the following HVAC maintenance measures: refrigerant charge adjustments, condenser blockage, evaporator blockage, and economizer perimeter sealing.

5.1 Evaluating Refrigerant Charge Impacts Using Laboratory Tests

One important purpose of laboratory testing is to develop accurate regression equations of sensible energy efficiency (EER^*_s) versus refrigerant charge per factory charge in order to evaluate energy efficiency improvements of refrigerant charge adjustment measures implemented in HVAC maintenance programs. The refrigerant charge fault test regression equations can be used to calculate EER*s impacts based on recovery and weigh-out of refrigerant charge and the reported charge adjustment per circuit (see **Equations 10**, **16**, **20**, and **24**). Field observations of 35 units found an average difference between recovered and pre-existing refrigerant charge of 15.1 +/- 3.2% corresponding to an average EER*s impact of 8.9 +/- 3.9%. This is 1.9 times greater than the average EER*s impact of 4.6% that could be calculated based on the average difference between recovered (94.4%) and pre-existing charge (79.1%).

5.2 Evaluating Evaporator Coil Cleaning Impacts Using Laboratory Tests

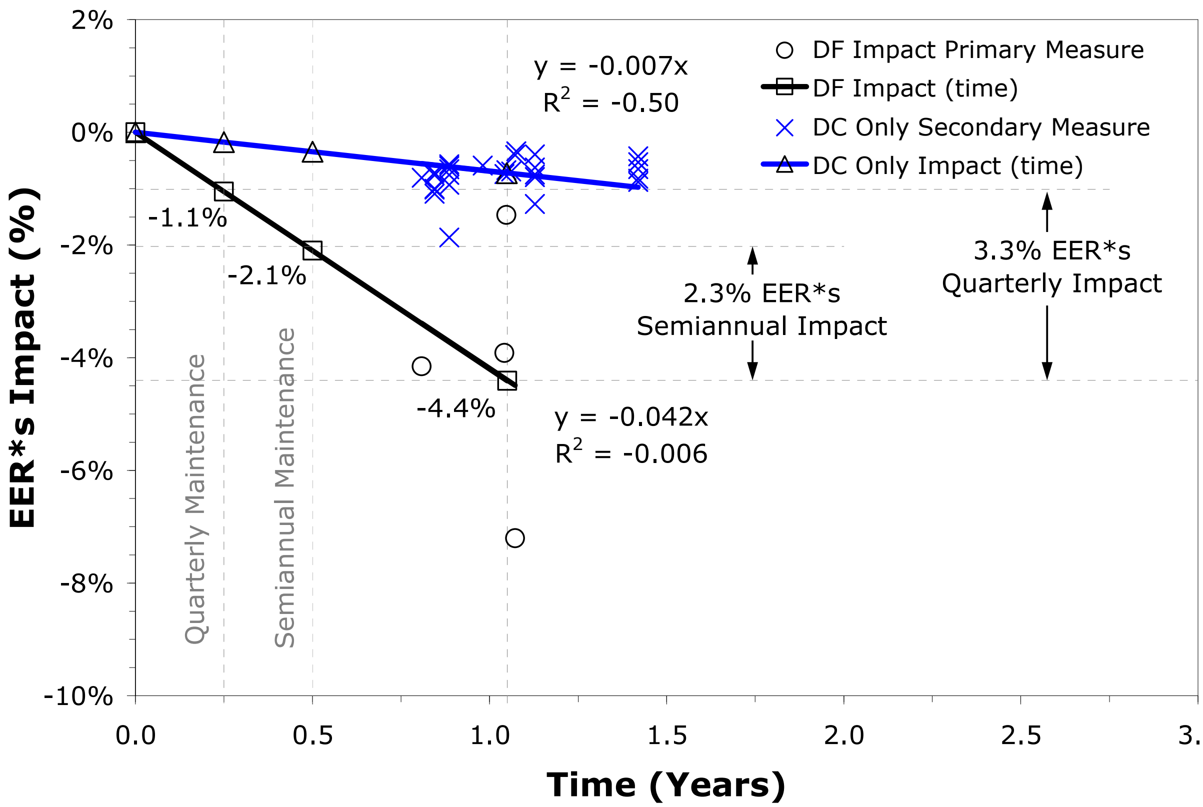
Evaporator blockage is primarily caused by dirty filters (DF) and secondarily caused by dirty coils (DC). The dirt deposition rate is dependent on the four variables: 1) outdoor airflow, 2) atmospheric particulates, 3) operational time, and 4) filter media effectiveness. Filters are designed to remove dirt from air to maintain indoor air quality. Most commercial buildings have MERV 8 pleated filters to remove particles such as pollen, mold, and dust (3 to 10 microns).[148] ACCA 180 requires quarterly filter replacement to avoid bypass caused by excessive filter loading and maintain adequate airflow and energy efficiency. Dirty filters reduce airflow and sensible cooling capacity causing the air conditioner to take more time to satisfy the drybulb thermostat. This causes the unit to operate longer or continuously. Low airflow causes reduced evaporator temperatures where the filter temperature drops below the dew point causing moisture to collect on the filter which mixes with dirt further reducing airflow and increasing condensation on the filter. Water on the coil and filter eventually freeze and the entire filter and coil can form a solid block of ice which reduces airflow, sensible capacity, and efficiency even more. Filter blockage causes longer operational time which is not included in the laboratory measurements. Short-term laboratory tests were performed to evaluate evaporator blockage without icing of the filter or coil. In between evaporator blockage tests the compressors were disabled and the evaporator fan was operated by itself to melt ice off the coil and filer. The ACCA 180 standard addresses this problem by requiring quarterly air filter replacement. The California HVAC maintenance program work papers did not fully understand the energy efficiency impacts of dirty filters. Consequently, no energy savings were claimed for quarterly filter replacement per ACCA 180 which is required in the programs.

Figure 62 provides field measurements performed at schools on 28 small packaged units. The units were located in regions with relatively clean air, typical air filter media, and average operational times. Airflow was measured before and after cleaning dirty evaporator coils previously cleaned 12.6 +/- 0.8 months earlier (1.05 +/- 0.07 years). The average measured airflow improvement for the 28 units was 0.9 +/- 0.1% equivalent to 5% evaporator blockage. This corresponds to an average EER*s impact of 0.7 +/- 0.1% per year based on **Equation 22** as shown by the sold blue line in **Figure 62**. DF blockage reduced airflow by 5.2% +/- 2.4% corresponding to an average EER*s impact of 4.4% as shown by the solid black line in **Figure 62**. This is equivalent to 27% evaporator blockage (see **Table 86**).[149] For both sets of data the R-squared for the linear regressions are low indicating a larger sample is necessary to more accurately evaluate this measure. DF blockage caused approximately 6 times more impact than DC only blockage due to the filter which collects most of the dirt.

[148] ASHRAE Standard 52.5. 2006. Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size.

[149] Reduced airflow of 5.2% due to dirty coils plus filters corresponds to 27% evaporator blockage for the 3-ton non-TXV unit based on 4% reduced airflow at 20% blockage and 7.9% reduced airflow at 35% blockage.

Figure 62: EER*s Impact versus Time for Evaporator DF and DC Blockage



Clean air filters and coils do not impact the baseline EER*s. Therefore, the linear regression zero intercepts are correct. While 22 measurements were performed on units with clean filters, the analysis must include dirty air filters since these are the primary cause of reduced airflow, inlet static pressure, and EER*s. Furthermore, all HVAC maintenance programs in California require filter replacement for incentive payments. While the programs did not explicitly define energy savings based on air filter replacement, the ex ante estimates of 5 to 7% savings for this measure support the inclusion of air filter replacement. The regression equation based on DF blockage data for 4 units is provided in **Equation 24**.

Equation 24 $y_{df} = -0.043t$

Where,

y_{df} = EER*s impact due to DF blockage with respect to time (dimensionless)

t = time since the previous DF replacement (years)

Based on field measurements of 4 units with DF blockage the average estimated EER* impact is 4.4%. ACCA 180 recommends quarterly or semi-annual air filter replacement.[150] Quarterly maintenance impacts EER*s by 1.1% over 3 months and semiannual maintenance impacts EER*s by 2.1% EER* over 6 months. The difference between one-year and 3 to 6 months of dirt deposition corresponds to an EER*s impact of 2.3 to 3.3% assuming the pre-existing base case was annual maintenance. For regular maintenance per ACCA 180, the 2.3 to 3.3% savings would continue for as long as consistent air filter and coil cleaning services are performed with an effective useful life (EUL) of 3 years. For one-time coil cleaning and air filter replacement, the savings would be approximately one-half of the 1-year impact or 2.2% with a 1-year EUL. Dirty filters cause even larger EER*s impacts in the field due to longer operational times which were not measured in the laboratory.

5.3 Evaluating Condenser Coil Cleaning Impacts Using Laboratory Tests

Evaluating condenser blockage removal based on field measurements and laboratory tests requires measuring discharge pressure ratios before and after removing the blockage. Condenser coil blockage is caused by the deposition of dirt and debris on coil. Dirt blocks the condenser which reduces airflow and heat transfer which cause increased discharge pressure (DP) which reduces cooling efficiency. The dirt deposition rate is dependent on three variables: 1) condenser airflow, 2) atmospheric particulates, and 3) operational time.

Figure 63 provides field measurements performed at schools on 28 small packaged units. Compressor discharge pressure was measured before and after cleaning dirty condenser coils previously cleaned 12.6 +/- 0.8 months earlier (1.05 +/- 0.07 years).[151] The average measured discharge pressure reduction was 4.8 +/- 0.7%. The R-squared for the linear regression is low indicating a larger sample is necessary to more accurately evaluate this measure. Clean coils do not impact the baseline EER*s. Therefore, the linear regression equation must intercept zero.

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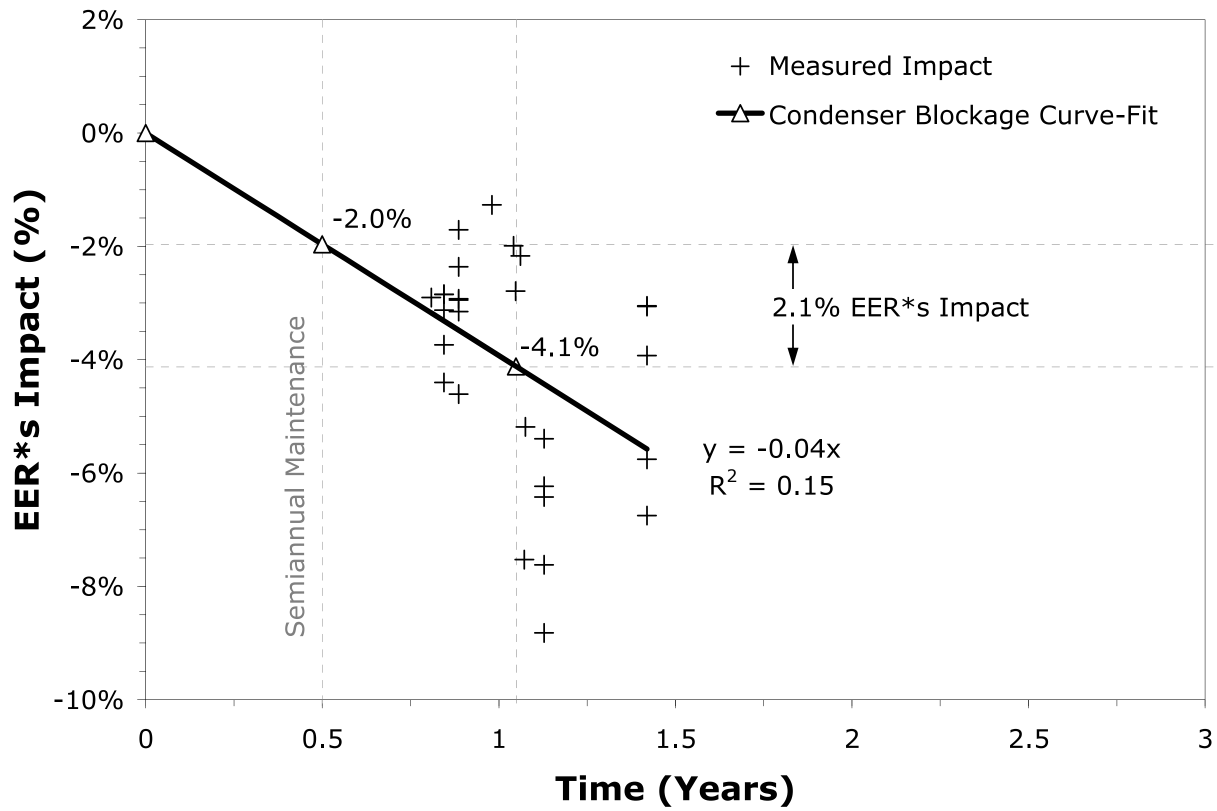
[150] Table 5-22. Rooftop Units. Item A: Check for particulate accumulation on filters. Clean or replace if accumulation results in pressure drop or airflow outside of established operating limits as necessary to ensure proper operation. Frequency: Quarterly. ANSI/ASHRAE/ACCA 2008. Standard 180: Standard Practice for Inspection and Maintenance of Commercial Building HVAC Systems.

[151] Discharge pressure measurements were made at the same pre/post OAT by waiting until the next day to take the post measurement with clean and dry condenser coils and similar operating conditions.

Figure 63: EER*s Impact versus Time for Condenser Coil Blockage



The DP difference corresponds to an average EER*s impact of 4.1 +/- 0.6% based on **Equation 21**. This is equivalent to laboratory tests with 17% condenser coil blockage.[152] Based on field measurements, the estimated EER*s impacts due to 0.5 to 1.05 years of dirt accumulation on 28 condensers is shown in **Figure 63**. The linear regression equation is provided in **Equation 25.**

Equation 25 $y_c = -0.04t$

Where,

y_c = EER*s impact of condenser blockage with respect to time (dimensionless)

t = time since previous removal of condenser blockage (years)

Based on field measurements of 28 units with condenser blockage the average estimated EER*s impact is 4.1%. ACCA 180 recommends semi-annual condenser coil cleaning.[153] Based on

[152] Increased DP of 4.8% due to dirty coils corresponds to 17% condenser blockage for the 3-ton non-TXV unit based on 3.7% increased DP at 10% blockage and 7.2% increased DP at 20% blockage.

[153] Table 5-22. Rooftop Units. Item P: Check for evidence of build-up on or fouling on heat exchange surfaces. Clean restore as needed to ensure proper operation. Frequency: Semi-annually. ANSI/ASHRAE/ACCA 2008. Standard 180: Standard Practice for Inspection and Maintenance of Commercial Building HVAC Systems.

these field measurements, semi-annual maintenance impacts EER*s by 2%. The difference between one year and six months of dirt deposition corresponds to an EER*s impact of 2.1 +/- 0.3% assuming the pre-existing base case was annual maintenance. For regular maintenance per ACCA 180, the 2.1% savings would continue for the length of the service agreement with a 3-year effective useful life (EUL). For one-time coil cleaning, the savings would be approximately one-half of the 1-year impact or 2% with a 1-year EUL.

5.4 Evaluating Economizer Perimeter Sealing Impacts Using Laboratory Tests

Economizer perimeter sealing (EPS) impacts can be evaluated based on field measurements of the OAF before and after removing the hood and sealing the perimeter where it attaches to the cabinet using UL-181 waterproof tape. Laboratory tests of four packaged units found an average OAF difference of 6 +/- 2% resulting in an EER*s improvement of 5 +/- 2%. EPS improves cooling and heating efficiency. **Equation 1** can be used to calculate the pre- and post-OAF with compressors operating if the outdoor and return air temperature difference of at least 20F. The OAF will be more accurate if the outdoor and return temperature difference is greater than 20F. **Equation 26** can be used to evaluate EER*s impacts based on the sealed and unsealed perimeter OAF reduction and minimum damper position.

Equation 26 $$y_{eps} = 0.915 \times [\Delta \mathrm{OAF}] \times [1 + (0.1 * \mathrm{MDP})^{0.1}]$$

Where,

y_{eps} = EER*s impact due to economizer perimeter sealing (dimensionless)

ΔOAF = difference between unsealed and sealed OAF (%)

MDP = minimum damper position (0=closed, 1 to 3-fingers open)

Table 122 provides the measured versus predicted EER*s impact from EPS based on sealed and unsealed OAF difference using **Equation 25**.

Table 122: Measured versus Predicted EER*s Impact from EPS based on Sealed and Unsealed OAF Reduction Using Equation 25

Economizer Damper Position	Minimum Damper Position (MDP)	Average ΔOAF	Average Measured EER*s Impact	Average Predicted EER*s Impact	Difference
Closed (2.0V)	0	5.9%	5.4%	5.4%	0.0%
1-Finger (5.1V)	1	4.3%	4.9%	4.7%	0.2%
2-Fingers (6V)	2	3.1%	4.4%	3.7%	0.7%
3-Fingers (6.9V)	3	2.5%	2.6%	3.3%	-0.7%
Average		**3.9%**	**4.3%**	**4.3%**	**0.0%**

6 CONCLUSIONS

The following conclusions are based on three years of laboratory test results and four years of field measurements to inform the design, implementation, and analysis of test results.

6.1 Laboratory Test Results

Laboratory tests were conducted on one- and two-compressor packaged HVAC units from the largest manufacturers representing 75% of systems that received IOU program services. The specific non-TXV models tested in the laboratory represent 14% of total units that received incentives in one of the largest commercial HVAC maintenance programs. Including heat pumps with similar evaporator, compressor, expansion devices, economizer and manufacturer RCA protocols, the models tested in the laboratory represent 25% of total non-TXV units. The TXV models tested represent 23% of total units (i.e., tons of cooling) that participated in the same program. Laboratory tests were conducted on economizers from the largest manufacturers representing 90% of all economizers receiving HVAC maintenance program services. Laboratory tests were also performed on field measurement instruments representing 80% of instruments used by technicians performing services in the HVAC maintenance programs based on observations of technicians. The laboratory test results are applicable to non-tested systems. The test results can be used to develop improved estimates of cooling system performance by modifying simulation algorithms in eQuest. Laboratory test data can also be used to develop more accurate ex ante estimates of DEER energy savings for HVAC maintenance measures with cooling system impacts. The test data will also assist in the development of non-DEER work papers.

The following laboratory tests have been completed.

- Out-of-box efficiency,
- AHRI verification,
- Unit-specific or circuit-specific manufacturer refrigerant charge diagnostics,
- Economizer outdoor airflow,
- Economizer efficiency at 95F OAT and various damper positions,
- Economizer efficiency compared to non-functional economizer,
- Airflow faults,
- Refrigerant charge faults,
- Evaporator blockage faults,
- Condenser blockage faults,
- Restriction faults,
- Non-condensable faults, and

- Multiple faults.

In addition, laboratory tests have evaluated economizer perimeter frame leakage, economizer functionality and modes of failure, measurement instrument accuracy, refrigerant charge FDD, and issues associated with attaching and detaching refrigerant hoses to suction and discharge pressure valves. These tests were conducted to better understand FDD and impact issues related to HVAC maintenance measures as well as suggest new measures that are currently not part of the HVAC maintenance or installation programs that significantly impact space cooling and heating efficiency.

Field data collection protocols have been tested in the field and the laboratory to evaluate energy efficiency impacts of condenser and evaporator coil cleaning, refrigerant charge adjustment, and economizer repair. Efficiency impacts have been correlated to laboratory test results of EER*s versus compressor discharge pressure, evaporator airflow, refrigerant charge, and outdoor air fractions. The protocols can be used by technicians to reduce unintended outdoor airflow, establish minimum outdoor airflow per AHSRAE 62.1, and improve cooling and heating efficiency. Test results are summarized by similar tests that have been completed across all units.

6.1.1 Out-of-Box Tests

Out-of-box efficiency tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU1 and RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4. Initial out-of-box tests were performed on each unit without an economizer installed to determine whether or not any adjustments were required to perform the ANSI/AHRI Standard 210/240 or ANSI/AHRI Standard 340/360 certification tests to verify published manufacturer ratings. Each unit was shipped from the factory with fan speed settings established for typical field conditions where total ESP ranges from 0.5 to 1.2 inches of water (IWC). Typical field ESP values are 3 to 10 times greater than ESP values specified in the ANSI/AHRI test procedure so fan speed will generally be reduced to perform ANSI/AHRI verification tests. The minimum ESP is 0.25 IWC for the 7.5-ton units and 0.15 IWC for the 3-ton units. While there is no ESP upper limit, airflow cannot exceed 450 cfm/ton of rated cooling capacity.

The 7.5-ton non-TXV RTU3 out-of-box airflow was 400 scfm with ESP ranging from 0.802 to 0.897 IWC and 7-inch diameter pulley. The out-of-box 2-compressor application EER* tests were 24 to 27% less than the published AHRI efficiency ratings and the one-compressor EER* tests were 45 to 50% less than published AHRI ratings. With 2-compressors operating, the average SEER* was 9.4 and the cyclic degradation coefficient (C_d) was 0.084. With 1-compressor operating, the average SEER* was 6.7 and average C_d was 0.089. With the 10-inch diameter pulley, the EER* was 9.47 to 9.61 or 13 to 14% less than the AHRI rating.

The 7.5-ton TXV RTU1 and RTU2 out-of-box EER* tests were 6 to 24% less than the rated 11 EER with factory fan speed from 915 to 1047 rpm providing 544 and 678 scfm/ton airflow. Airflow was 21 to 51% greater than the 450 scfm/ton maximum airflow specified in ANSI/AHRI 340/360. A 2-hp fan motor was installed in RTU1 to reduce fan power and airflow and increase EER* from 8.39 to 9.53 or 13% less than the AHRI rating. For RTU2, the out-of-box application EER* was 10.3 or 6% less than the AHRI rating. For RTU2, the average SEER was 11.77 and average C_d was 0.215. The C_d was 2 to 3 times greater than previously assumed in DEER models.

The 3-ton non-TXV RTU5 out-of-box airflow was 600 scfm/ton at 945 rpm and motor sheave at 3 turns. This was 33% greater than the 450 scfm/ton maximum allowable airflow specified in ANSI/AHRI 210/240. The out-of-box application EER* was 10.2 to 10.5 or 6 to 9% less than the AHRI rating. The fan speed was reduced to 756 rpm at 5.5 turns to increase EER* to 10.7 or 4% less than the AHRI rating. Intertek recovered refrigerant, evacuated to 500 microns and weighed in the factory charge. With factory charge the EER* was 10.9 or 3% less than the AHRI rating.

The 3-ton TXV RTU4 out-of-box airflow was 662 scfm/ton at 808 rpm and motor sheave at 2.5 turns. This was 47% greater than the 450 scfm/ton maximum allowable airflow in ANSI/AHRI 340/360. The out-of-box application EER* was 4% less than the AHRI rating. The fan speed was reduced to 663 rpm at 5 turns to increase EER* to 11.4 or 2% greater than the AHRI rating. Intertek recovered refrigerant, evacuated to 500 microns and weighed in the factory charge. With factory charge the EER* was 11.5 or 3% greater than the AHRI rating.

Overall conclusions and observations from the out-of-box tests are listed below.

- Systems taken directly “out of the box” and setup for testing without any modifications from factory settings generally operated at higher supply airflow rates than allowed by the applicable AHRI standard (ANSI/AHRI 210/240 or ANSI/AHRI 340/360).
- The combination of high supply airflow rates and ESP increased fan power and reduced overall application efficiency below the rated values for both of the dual compressor systems.
- Single-compressor 3-ton non-TXV RTU5 and 3-ton TXV RTU4 tested within the published AHRI efficiency tolerances with higher than rated supply airflow rates and fan power.
- The out-of-box delivered refrigerant charge in several circuits deviated from factory charge, indicating issues with the manufacturing process or refrigerant loss during shipping or setup.

6.1.2 AHRI Verification Tests

AHRI verification tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4. After the out-of-box tests were completed Intertek technicians recovered and weighed the refrigerant charge from each unit to evaluate discrepancies from the manufacturer factory charge. Intertek technicians then evacuated each refrigerant circuit to 500 microns Hg and waited for at least 60 minutes with the evacuated system at or below 1000 microns prior to weighing in the factory charge. In addition, the evaporator fan speed was adjusted on each unit to achieve the required airflow and ESP specifications required under the ANSI/AHRI testing standards.

The 7.5-ton non-TXV RTU3 refrigerant charge was recovered from each circuit and weighed at 7.6 lbs in circuit 1 and 8.1 lbs in circuit 2 which is the manufacturer factory charge. Each circuit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the factory charge into each circuit. The motor speed was adjusted to 6 turns to achieve 339 scfm/ton, and 6.4 ounces of refrigerant was added to each circuit to achieve 105% of factory charge and increase cooling capacity. With these modifications the AHRI-test efficiency was 10.47 EER at 95F or within 4.8% of the published rating and the cooling capacity was 86,269 Btuh or within 4.1% of the published AHRI ratings. The measured IPLV was 11.03 or within 4.1% of the published IPLV rating.

The 7.5-ton TXV RTU2 refrigerant charge was recovered from each circuit and weighed at 6.44 lbs in circuit 1 which is within 1% of the manufacturer factory charge and 4.68 lbs in circuit 2 which is 25% less than the factory charge. Each circuit unit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the factory charge into each circuit. The motor speed was adjusted to 6 turns to achieve 381 scfm/ton for RTU2 and 416 scfm/ton for RTU1. With these modifications the AHRI-test efficiency for RTU2 was 10.9 EER or within -0.7% of the AHRI rating. The 25% undercharge in C2 had less than a 1% impact on the steady-state efficiency of RTU2. The measured efficiency of RTU1 was 10.6 EER at 95F or within 4.9% of the published rating. The measured IPLV was 11.92 for RTU2 and 11.56 for RTU1. Measured IPLV values were 0.5 to 4% greater than rated values.

The 3-ton non-TXV RTU5 refrigerant was recovered and weighed at 5.3 lbs or 4% less than the manufacturer factory charge of 5.5 lbs. The unit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the factory charge. The motor speed was adjusted to 5 turns to achieve 430 scfm/ton. At this airflow, the AHRI-test efficiency was 10.97 EER at 95F and 12.96 EER at 82F. These test results were 2 to 7% less than published AHRI ratings. The SEER test was 12.3 and within 95% of the published rating. All measured values for the 3-ton non-TXV unit were within ANSI/AHRI tolerances.

The 3-ton TXV RTU4 refrigerant was recovered and weighed at 6.4 lbs or 9% less than the manufacturer factory charge of 7 lbs. The unit was evacuated to 500 microns Hg and held at or below 1000 microns for 60 minutes prior to weighing in the factory charge. The ESP was increased to 0.36 IWC to achieve the 400 scfm/ton AHRI-rated airflow. The AHRI-test efficiency was 11.5 EER at 95F and 14.0 EER at 82F OAT which were 2 to 3% higher than the published ratings. The SEER test was 13.1 and the average C_d was 0.137. The EER was within 2%, cooling capacity was within 2.9%, and SEER was within 0.5% of the published ratings per ANSI/AHRI tolerances.

Overall conclusions and observations from the AHRI verification tests are listed below.

- Modifications were required to each unit to establish required AHRI test conditions and achieve the published efficiency and capacity ratings within the AHRI allowable tolerances for EER, cooling capacity and/or SEER if applicable.
- Modifications included operating fans at lower speeds through a combination of adjusting the motor sheave or installing a larger diameter pulley to reduce airflow rates to within the AHRI test limits, and reducing static pressure by operating the code tester fan at higher speeds to establish the AHRI-specified ESP.
- For each unit, the refrigerant charge was recovered from each circuit and carefully weighed to determine deviations from the manufacturer nameplate factory charge. Each circuit was evacuated to 500 microns and held at or below 1000 microns for 30 minutes, and the factory charge was carefully weighed into each circuit. The 7.5-ton non-TXV unit needed to be

slightly overcharged by 5% in each circuit to meet the published AHRI EER and cooling capacity ratings.

- For all units cabinet panel joints were sealed with tape to eliminate outdoor air leakage and tested in the horizontal airflow configuration with no economizer to achieve the published AHRI ratings.

6.1.3 Manufacturer Refrigerant Charge Diagnostic Tests

Unit-specific manufacturer refrigerant charge diagnostic tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4.

The 7.5-ton non-TXV RTU3 circuit-specific manufacturer refrigerant charge diagnostics are based on suction temperature (ST) as a function of outdoor drybulb (DB) temperature (i.e., condenser entering air) and suction pressure (SP). The manufacturer refrigerant charge ST tolerances and the CEC superheat (SH) tolerances are +/-5F. For steady-state conditions with factory charge and no economizer, both circuits indicated “false alarm” undercharge with respect to the manufacturer ST and CEC SH protocols. With 105% factory charge in both circuits C1 passed manufacturer ST and CEC SH protocols but C2 failed both protocols with a “false alarm” undercharge even though EER and cooling capacity were within published AHRI ratings. When tested at field conditions with economizer installed and dampers closed the manufacturer ST and CEC SH protocols provided correct results more often for C1 than C2. C2 was generally diagnosed as undercharged except when C2 was overcharged by 56 to 68%. For overcharge tests the protocols did not generally detect the faults. For undercharge, the protocols did detect the faults which reduced EER*s by 25 to 53%. For the 7.5-ton non-TXV unit, the manufacturer ST protocols correctly identified 20% undercharge tests, but the overall accuracy was only 25 +/-4% due to misdiagnosing overcharge or correct charge as undercharge. The CEC SH protocols provided similar accuracy of 23 +/- 4%. The CEC TS protocol provided accuracy of 85 +/- 3% for evaluating low airflow or low sensible cooling capacity due to overventilation, refrigerant over/undercharge, refrigerant restrictions, and non-condensables.

The 7.5-ton TXV RTU1 and RTU2 the circuit-specific manufacturer refrigerant charge diagnostics are based on discharge pressure (DP), suction pressure (SP), and superheat (SH) as a function of outdoor and return (i.e., evaporator coil entering) DB/WB temperatures. The manufacturer tolerances are +/-10 psig for DP, +/-5 psig for SP and +/-5F for SH. The manufacturer does not provide liquid pressure ports so subcooling cannot be evaluated. For steady-state conditions RTU2 and RTU1 passed all manufacturer charge diagnostics with correct factory charge in both circuits. With RTU2 circuit 2 undercharged by 10%, the manufacturer DP/SP/SH protocols did not detect any faults but there was very little impact on application sensible efficiency or capacity. Undercharging refrigerant 20 to 40% reduced EER*s by 3 to 20% and the manufacturer DP/SP/SH protocols correctly diagnosed the faults. Overcharging by 5 to 40% reduced EER*s by 0 to 1% and the manufacturer protocols only partially diagnosed the overcharge. For the 7.5-ton TXV unit, the manufacturer DP/SP/SH protocols generally provided correct diagnosis at factory charge and 20 to 40% undercharge with overall accuracy of 54 +/- 4%. The CEC TS protocol provided accuracy of 90 +/- 4% for evaluating low airflow or low sensible capacity due to overventilation, refrigerant over/undercharge, refrigerant restrictions, non-condensables, and multiple-faults.

The 3-ton non-TXV RTU5 circuit-specific manufacturer refrigerant charge diagnostics are based on ST as a function of outdoor DB temperature (i.e., condenser entering air) and SP. The manufacturer refrigerant charge ST and CEC SH tolerances are +/-5F. For steady-state conditions with factory charge and no economizer, both protocols indicated "false alarm" undercharge with respect to manufacturer ST and CEC SH protocols. With the economizer installed and dampers closed or 1-finger open, the manufacturer ST protocol correctly diagnosed 10 to 40% undercharge and 10 to 40% overcharge for 89% accuracy. With dampers closed, the CEC SH protocols correctly diagnosed 10 to 40% undercharge. With dampers 1-finger open, the CEC protocols correctly diagnosed 10 to 40% undercharge and 20 to 40% overcharge for 61% accuracy. Undercharging refrigerant by 10% to 40% reduced EER*s by 0 to 28%. Overcharging by 10 to 40% reduced EER*s by 0 to 3%. The manufacturer ST protocol overall accuracy was 45 +/- 8%, and the CEC SH protocol overall accuracy was 51 +/- 8%. The CEC TS protocol overall accuracy was 99 +/- 1% for evaluating low sensible cooling capacity due to overventilation and refrigerant over/undercharge.

The 3-ton TXV RTU4 unit-specific manufacturer refrigerant charge diagnostics are based on DP and SP as a function of outdoor DB temperature (i.e., condenser entering air) and SC. The manufacturer tolerances are +/-10 psig for DP, +/-5 psig for SP, and +/-1F for SC. The CEC subcooling tolerances are +/-3F. For steady-state conditions with factory charge and no economizer, the manufacturer DP/SP/SC protocols indicated correct charge. When tested at field conditions with economizer installed and dampers closed or 1-finger open, the manufacturer subcooling protocol correctly diagnosed all charge conditions. The manufacturer DP/SP protocols misdetected 10 to 40% overcharge and 10 to 20% undercharge and correctly diagnosed 30 to 40% undercharge. The CEC subcooling protocol correctly diagnosed 10 to 40% undercharge and 40% overcharge, but misdiagnosed factory charge and 10 to 30% overcharge. Undercharging refrigerant by 10 to 40% reduced EER*s by 1 to 41% and overcharging increased EER*s by 0 to 2%. For this unit misdiagnosing overcharge as correct does not cause any issues. The manufacturer subcooling protocol provided 97% accuracy, but the combined accuracy of the manufacturer DP/SP/SC protocols was 68 +/- 7%. The CEC subcooling protocol accuracy was 51 +/- 14%. The CEC TS protocol accuracy was 88 +/- 6% for evaluating low airflow or low sensible cooling capacity due to overventilation and refrigerant over/undercharge.

Table 123 compares manufacturer and CEC refrigerant charge and airflow diagnostic accuracy. 7.5-ton non-TXV RTU3 the manufacturer ST protocol accuracy was 25 +/- 4% for all tests, and the CEC SH protocol accuracy was 23 +/- 4% for all tests. For the 7.5-ton TXV RTU1 and RTU2 the manufacturer discharge, suction, and superheat protocols accuracy ranged from 46 to 60% with overall accuracy of 54 +/- 4%. For the 3-ton non-TXV RTU5, the manufacturer ST protocol accuracy was 45 +/- 8% for all tests, and the CEC SH protocol accuracy was 51 +/- 8% for all tests. For the 3-ton TXV RTU4, the manufacturer subcooling protocol accuracy was 97% for all tests while the discharge and suction pressure protocols accuracy was 57 to 51%, and the combined accuracy was 68 +/- 7%. For the 3-ton TXV RTU4, the CEC SC protocol accuracy was 51 +/- 14% for all tests. Based on all test results, the manufacturer RC protocols were correct 48 +/- 3% of the time, while the CEC RC protocols were correct 31 +/- 4% of the time. The CEC TS protocols were correct 90 +/- 2% of the time when diagnosing low airflow or low sensible cooling capacity due to overventilation or other maintenance faults. For comparison, studies of medical diagnostic accuracy indicate general accuracy of 31% with 55% accuracy for easier cases and 5.8% for more difficult cases.[154]

Table 123: Manufacturer and CEC Refrigerant Charge and Airflow Diagnostic Accuracy

Description	Mfr ST RC Accuracy %	Mfr DP RC Accuracy %	Mfr SP RC Accuracy %	Mfr SH RC Accuracy %	Mfr SC RC Accuracy %	CEC SH RC Accuracy %	CEC SC RC Accuracy %	CEC TS Airflow Accuracy %	CEC TS Capacity Accuracy %
7.5-ton non-TXV	25					23		83	87
7.5-ton TXV		55	60	46				91	89
3-ton non-TXV	45					51		100	99
3-ton TXV		57	51		97		51	89	86
Average	**48 +/- 2**					**31 +/- 4**		**90 +/- 2**	

Overall conclusions and observations from the charge diagnostic tests are listed below.

- Due to being tested under conditions and with faults they were not intended to diagnose, the accuracy of manufacturer refrigerant charge (RC) protocol was 48 +/- 3% based on a sample of 992 measurements over the range of faults tested. The accuracy of CEC SH and SC protocols was 31 +/- 4% based on 445 measurements. The unit-specific manufacturer RC protocols generally diagnose more parameters on both the high- and low-side providing 47% greater accuracy than the generic CEC SH and SC protocols.
- The CEC TS protocol average accuracy was 90 +/- 2% based on 736 tests evaluating low airflow or low sensible cooling capacity due to overventilation, evaporator/condenser blockage, refrigerant over/undercharge, refrigerant restrictions, and non-condensables.
- The 7.5-ton non-TXV RTU3 manufacturer ST protocol correctly identified 20% undercharge tests, but general diagnostic accuracy was only 25% due to misdiagnosing overcharge or correct charge as undercharge. The CEC SH protocols accuracy was 23%. The CEC TS protocol accuracy was 85 +/- 3% for evaluating low airflow or low sensible cooling capacity

[154] JAMA Internal Medicine. Physicians' Diagnostic Accuracy, Confidence, and Resource Requests. A Vignette Study. 2013. Journal American Medical Association (JMMA). Ashley N. D. Meyer, PhD; Velma L. Payne, PhD, MBA; Derek W. Meeks, MD; Radha Rao, MD; Hardeep Singh, MD, MPH. http://archinte.jamanetwork.com/article.aspx?articleid=1731967. A total of 118 physicians with broad geographical representation within the United States correctly diagnosed 55.3% of easier and 5.8% of more difficult cases (P<.001). Despite a large difference in diagnostic accuracy between easier and more difficult cases, the difference in confidence was relatively small (7.2 vs 6.4 out of 10, for easier and more difficult cases, respectively) (P<.001) and likely clinically insignificant. Overall, diagnostic calibration was worse for more difficult cases (P<.001) and characterized by overconfidence in accuracy. Higher confidence was related to decreased requests for additional diagnostic tests (P=.01); higher case difficulty was related to more requests for additional reference materials (P=.01). "Expert Opinion" Software for Medical Diagnosis and Treatment. Robert M. Ross, MD. JAMA Intern Med. 2014;174(4):638-639. doi:10.1001/jamainternmed.2013.13794.

due to overventilation, evaporator/condenser blockage, refrigerant over/undercharge, refrigerant restrictions, non-condensables, and multiple-faults.

- The 7.5-ton TXV RTU1 and RTU2 manufacturer protocols generally provided correct diagnosis at factory charge and 20 to 40% undercharge with overall accuracy of 54%. The CEC TS protocol average accuracy was of 90 +/- 4% for evaluating low airflow or low sensible capacity due to overventilation, evaporator/condenser blockage, refrigerant over/undercharge, refrigerant restrictions, non-condensables, and multiple-faults.
- The 3-ton non-TXV RTU5 manufacturer ST protocols were accurate for 10 to 40% under and overcharge and the CEC SH protocols were accurate for 10 to 40% undercharge. The manufacturer ST protocol accuracy was 45% and the CEC SH protocol accuracy was 51%. The CEC TS protocol accuracy was 99 +/- 1% for evaluating low sensible cooling capacity due to overventilation, evaporator/condenser blockage, refrigerant over/undercharge, and multiple-faults.
- The 3-ton TXV RTU4 manufacturer SC protocol accuracy was 97%, but the combined diagnostic accuracy of the DP/SP/SC protocols was 68%. The CEC SC protocols provided correct diagnosis for 10 to 40% undercharge with overall accuracy of 51%. The CEC TS protocol accuracy was 88 +/- 6% for evaluating low sensible cooling capacity due to overventilation and refrigerant over/undercharge.
- Charge protocols were tested using laboratory grade instruments for establishing evaporator coil entering conditions when required by the protocol. It will be difficult to measure coil entering conditions in the field without temporarily closing and sealing outdoor air dampers. This is especially important on two compressor systems where the upper evaporator circuit is aligned with the economizer dampers in the vertical configuration. Higher coil entering temperatures due to uneven mixing of return and outdoor air causes more refrigerant to evaporate and higher suction and superheat temperatures. Using the manufacturer and CEC protocols, one circuit was diagnosed as undercharged until overcharged by 56 to 68% due to evaporator coil inlet temperature stratification and other issues.
- Manufacturer troubleshooting procedures and refrigerant charge protocols generally provide circuit-specific FDD information on additional faults besides refrigerant charge, making them more versatile, useful, and accurate for performing general HVAC maintenance services.

6.1.4 Field Data Collection Protocols for Evaluating HVAC Maintenance Measures

Field data collection protocols were tested in the field and the laboratory to evaluate energy efficiency impacts of the following HVAC maintenance measures.

- Refrigerant charge measures were evaluated in the field and the laboratory based on the ratio of refrigerant charge per factory charge (i.e., recovery and weigh-out of refrigerant) and regression equations correlated to laboratory tests of sensible efficiency versus percent factory charge.

- Condenser coil cleaning measures were evaluated in the field and the laboratory based on the ratio of as-found dirty-to-clean condenser coil discharge pressure and OAT and regression equations based on laboratory tests of sensible efficiency versus discharge pressure ratio at constant OAT due to coil blockage emulated by installing plastic-corregated cardboard on the upstream side of the condenser.
- Evaporator coil cleaning measures were evaluated in the field and the laboratory based on the as-found dirty-to-clean evaporator coil airflow ratio and regression equations based on laboratory tests of sensible efficiency versus evaporator airflow ratio due to coil blockage emulated by installing plastic-corregated cardboard on the upstream side of the evaporator.
- Economizer outdoor airflow repair or adjustment measures were evaluated in the field and the laboratory based on the as-found outdoor air fraction and regression equations based on laboratory tests of sensible efficiency versus outdoor air fraction. Outdoor airflow fractions were measured using average outdoor, return, and mixed air temperatures when the outdoor drybulb temperature is at least 20 degrees Fahrenheit greater than the return air temperature. These protocols can also be used by technicians to reduce unintended outdoor airflow, establish minimum outdoor airflow per AHSRAE 62.1, and improve cooling and heating efficiency.

6.1.5 Economizer Outdoor Airflow Tests

Economizer outdoor airflow tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4. Based on tests of five economizers installed on four units, the average closed economizer damper OAF was 18 +/- 3% of total system airflow which meets or exceeds ASHRAE 62.1 minimum ventilation requirements for most buildings. Opening economizer dampers from 1-to-3-fingers provided 27 to 39% OAF which far exceeds ASHRAE 62.1 minimum requirements and reduced sensible efficiency by 20 +/- 3% compared to closed dampers. Tests with tape sealing the gap under the hood between the economizer perimeter and cabinet found an OAF reduction of 6 +/- 2% and improved sensible efficiency of 5 +/- 2%. For all units tested, the average fully-open damper OAF was 68 +/- 5% which limits economizer free cooling.

The 7.5-ton non-TXV RTU3 economizer #4 outdoor airflow tests were performed with the evaporator fan on, compressors on, outdoor conditions of 95 and 115F, and indoor conditions of 75F DB and 62F WB. Tests were also performed with the evaporator fan blower motor on and compressors off at 55F OAT with dampers closed to fully open. For the first tests performed at 95 and 115F OAT with compressors operating and no economizer installed the OAF due to cabinet leakage was 6 to 7.5%. For the first tests with economizer perimeter unsealed, and dampers from closed to fully open, the OAF ranged from 16.7 to 66.9%. A second set of tests were performed at 55F OAT with no compressors operating with unsealed and sealed economizer perimeter. For the second tests with no economizer the OAF due to cabinet leakage was 2.3%. For the second tests with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 16 to 55.4%. For the second tests with perimeter sealed and dampers from closed to fully open, the OAF ranged from 13.4% to 58.1%. Sealing the economizer #4 perimeter (under the hood) reduced the OAF by 2 to 3.3%.

The 7.5-ton TXV RTU2 economizer tests were performed with three economizers. Economizer #0 was purchased with an OEM analog controller/actuator and was only able to open dampers slightly beyond 2-fingers. Economizer #0 was retrofitted with a digital controller/actuator (referred to as economizer #1) to test from closed to fully open positions. Economizer #2 was purchased from another manufacturer to test another unit with digital controller/actuator. Economizer #0, #1, and #2 were only tested with the evaporator fan blower motor on, compressors off, outdoor conditions of 55F and indoor conditions of 75F DB and 62F WB. Economizer #1 was also tested with compressors on at 95F OAT with dampers closed to fully open. With no economizers installed the outdoor air leakage varied from 2.5 to 5.7%. For economizer #0 with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 17.9 to 30.3%. Sealing the economizer #0 perimeter reduced outdoor airflow by 2.1 to 2.7%. For economizer #1 and 400 scfm/ton airflow with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 12.1 to 72.7% at 55F OAT. For economizer #1, the difference between measuring OAF at 55F OAT with no compressors operating and 95F OAT with both compressors operating was 1.7 +/- 1.3%. Sealing the economizer #1 perimeter reduced unintended outdoor airflow by 2.7 to 3.9%. For economizer #1 and airflow from 267 to 450 scfm/ton with perimeter unsealed and dampers from closed to fully open, the average OAF ranged from 12.2 to 74.7%. For economizer #2, and 360 scfm/ton airflow with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 29.7 to 78.1%. at 55F OAT. Sealing the economizer #2 perimeter reduced outdoor airflow by 12.6 to 14%.

The 3-ton non-TXV RTU5 economizer #5 tests were performed with the evaporator fan on, compressors off, outdoor conditions of 55F, and indoor conditions of 75F DB and 62F WB. Tests were also performed with compressors on at 95F OAT with dampers closed to fully open. With no economizer installed the OAF due to cabinet leakage was 7.1%. With no economizer installed the OAF varied from 2 to 8.4% depending on OAT. For economizer #5 with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 23.5 to 68% at 55F OAT. Sealing the economizer #5 perimeter reduced unintended outdoor air leakage by 1.8 to 9.5%.

The 3-ton TXV RTU4 economizer #6 tests were performed with the evaporator fan blower motor on, compressors off, outdoor conditions of 55F, and indoor conditions of 75F DB and 62F WB. Tests were also performed with compressors on at 95F OAT with dampers closed to fully open. With no economizer installed the OAF was 3.9% at 95F OAT. For economizer #5 with perimeter unsealed and dampers from closed to fully open, the OAF ranged from 19.9 to 66.4% at 55F OAT. Sealing the economizer #5 perimeter (under the hood) reduced unintended outdoor air leakage by 2.5 to 7.6%.

Overall conclusions and observations from the economizer outdoor airflow tests are listed below.

- The OAF provided by each economizer varied as a function of damper position. Manufacturers typically do not provide OAF as a function of ISP and damper position. Technicians currently have no reliable method to establish optimal damper position.

- Average closed damper OAF was 18 +/- 3%. This meets or exceeds ASHRAE 62.1 minimum outdoor air ventilation requirements for office and retail buildings.
- Sealing the gap between economizer frame and cabinet reduced outdoor airflow by an average of 6 +/-2% and improved EER*s by 5 +/- 2%.
- Average 100% fully-open damper OAF was 68+/- 5%, thus limiting the amount of free cooling supplied by the economizer when fully open.

6.1.6 Economizer Efficiency Tests at 95F OAT

Economizer efficiency tests at 95F OAT were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4. All tests were performed with indoor conditions of 72F DB and 62F WB and outdoor conditions of 95F. Tests were performed with economizer dampers closed to fully open.

The 7.5-ton non-TXV RTU3 efficiency reduction with economizer #4 installed and closed dampers was 23.5% for total EER* and 15.4% for EER*s compared to no economizer. The total EER* with closed dampers was 48% less than the AHRI rating. Opening dampers from 1- to 3-fingers reduced total EER* by 15 to 40% and EER*s by 8 to 20% compared to closed dampers. Sealing the perimeter frame improved the application efficiency by 5 +/- 2%.

The 7.5-ton TXV RTU2 efficiency reduction with economizer #1 installed and closed dampers was 13.6% for total EER* and 9.5% for EER*s compared to no economizer. The total efficiency with closed dampers was 31% less than the AHRI rating. Opening dampers from 1- to 3-fingers reduced total EER* by 20 to 30% and EER*s by 9 to 21% compared to closed dampers. Sealing the perimeter frame improved the application efficiency by 2 to 12%.

The 3-ton non-TXV RTU5 efficiency reduction with economizer #5 installed and closed dampers was 30% for total EER* and 17% for EER*s compared to no economizer. The total EER* with closed dampers was 42% less than the AHRI rating. Opening dampers from 1- to 3-fingers reduced total EER* by 22 to 69% and EER*s by 12 to 32% compared to closed dampers. Sealing the perimeter frame improved the application efficiency by 2 to 23%.

The 3-ton TXV RTU4 efficiency reduction with economizer #6 installed and closed dampers was 15% for total EER* and 4% for EER*s compared to no economizer. The total EER* with closed dampers was 31% less than the AHRI rating. Opening dampers from 1- to 3-fingers reduced total EER* by 21 to 47% and EER*s by 10 to 21% compared to closed dampers. Sealing the perimeter frame improved the application efficiency by 3 to 14%.

Overall conclusions and observations from the economizer efficiency tests are listed below.

- The average application efficiency for all units tested at typical field conditions with no economizer was 8.8 or 20% less than the AHRI rating. The average application efficiency for all units with an economizer and closed dampers was 7.0 or 37% less than the AHRI rating, and with 1-finger open dampers the average application efficiency was 5.6 or 49% less than the AHRI rating.

- Opening dampers by 1 to 3-fingers reduced total application efficiency by 14 to 69% and application sensible efficiency by 9 to 32% compared to no economizer.
- Sealing the economizer perimeter frame (under the hood) to reduced unintended outdoor airflow by 6 +/- 2% and improved EER*s by 5.4 +/- 2%.
- The CEC TS protocol correctly diagnosed low cooling capacity due to excess outdoor airflow for all tests.
- Outdoor airflow ventilation loads have a significant impact on cooling and heating efficiency especially when the minimum damper position is more open than necessary to meet the ASHRAE 62.1 minimum ventilation requirements.

6.1.7 Economizer Efficiency Tests at 70, 65, 60, and 55F

Economizer efficiency tests were performed on the 7.5 ton non-TXV RTU3. The 7.5-ton non-TXV RTU3 economizer efficiency tests were performed at 70, 65, 60, and 55F OAT to evaluate the performance of economizer #4 with outdoor air dampers 100% open to emulate a functional economizer and dampers closed with 1st stage compressor (C1) cooling to emulate a non-functional economizer. Indoor return air conditions were constant for all tests at 75F drybulb and 62F wetbulb. Outdoor conditions vary from 70F DB and 60F WB (70/60) to 65/57, 60/54, and 55/51. The economizer fan-only operated for at least 2 minutes plus time required for conditioned zone temperature to increase 2F above thermostat setpoint where C1 is energized. At 70F OAT, the economizer plus 1st stage cooling was approximately 6% more efficient than closed dampers plus 1st stage cooling. At 65F OAT the economizer plus 1st-stage cooling is 25% more efficient, at 60F OAT it is 59% more efficient, and at 55F OAT the economizer fan only meets the load and is 120% more efficient than closed dampers plus 1st-stage cooling.

Laboratory tests indicated economizer outdoor temperature changeover set points should be set at 70F or lower to achieve energy savings. If the economizer operates properly it can increase cooling capacity and save energy when outdoor temperatures are less than 70F. The test results demonstrate how economizer savings are related to minimum damper position. As minimum damper position increases from closed to 100% open, economizer savings approach zero.

6.1.8 Airflow Fault Tests

Airflow fault tests were performed on the 7.5 ton non-TXV RTU3 and 7.5-ton TXV RTU2.

The 7.5-ton non-TXV RTU3 airflow faults tests were performed with the 1.5-HP blower motor and economizer #4 installed, dampers closed, and economizer perimeter unsealed. The faulted tests were performed by increasing ESP to reduce airflow by 17% (333 scfm/ton) and 33% (267 scfm/ton). Tests were performed at 82, 95, and 115F OAT and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. The baseline airflow was 400 scfm/ton. For tests at 95F OAT and airflow decrease of 17 to 33%, the sensible cooling capacity was reduced by 8 to 25% and EER*s was reduced by 5 to 21%. For tests at 82F the sensible cooling capacity was reduced by 7 to 17% and the application efficiency was reduced by 5 to 13%. For tests at 115F the application sensible cooling capacity was reduced by 9 to 30% and EER*s was reduced by 7 to 29%. The EER*s was reduced by 3 to 18% with 1-finger open dampers compared to closed dampers. As expected, lower airflow, higher OAT, and overventilation (dampers open 1-finger) had the largest negative impacts on cooling capacity and efficiency. At 82F the CEC ΔTS protocol passed all tests and did not detect low airflow of 267 scfm/ton. At 115F the CEC ΔTS protocol indicated low cooling capacity for all tests caused by 15 to 34% outdoor airflow and 115F OAT. At 95F the CEC ΔTS protocol only provided correct diagnosis for the 333 scfm/ton tests with closed or 1-finger open dampers. The ΔTS indicated low sensible capacity for the 400 scfm/ton tests and did not detect low airflow for the 267 scfm/ton test.

The 7.5-ton TXV RTU2 standard and high ESP airflow faults tests were performed with economizer #1 installed, dampers closed, and economizer perimeter unsealed. Standard ESP tests were performed with the 2-HP blower motor and high ESP tests were performed with the 3-HP blower motor. Different motor sheave turns and pulley combinations were used to adjust fan speed and airflow from 250 to 435 scfm/ton. Tests were performed at 95F OAT and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. The baseline for standard ESP tests was 349 scfm/ton, and the baseline for high ESP tests was 318 scfm/ton. These were the airflow rates with maximum application sensible efficiency. For standard ESP tests and 28% decrease in airflow, EER*s decreased by 5%. For standard ESP and 25% increase in airflow, the EER*s decreased by 3%. For high ESP tests and 21% decrease in airflow, sensible cooling efficiency decreased by 2%. For high ESP and 34% increase in airflow, EER*s decreased by 15%. These tests demonstrate that adjusting fan speed and airflow at the unit causes less impact on efficiency than increasing ESP with the code tester to emulate faults in the HVAC duct system. For standard static pressure, the CEC protocol ΔTS was 3.7F at 252 scfm/ton airflow indicating low airflow and the sensible cooling capacity was 10% less than non-faulted 100% airflow tests. The CEC ΔTS values for all other standard static tests were between -3F and +3F. For high static pressure, the CEC ΔTS was 3.5F at 250 scfm/ton airflow indicating low airflow and the application sensible cooling capacity was 10% less than non-faulted 100% airflow test. The CEC ΔTS was -3.9F at 427 scfm/ton airflow indicating low sensible cooling capacity, but capacity was acceptable so this was a “false alarm.” All other ΔTS tests were between -3F and +3F with airflow and sensible cooling capacity.

Overall conclusions and observations from the airflow fault tests are listed below.

- Increasing or decreasing airflow to balance the system and meet AHSRAE 62.1 standards did not significantly impact sensible efficiency.
- The CEC temperature split (ΔTS) protocol reliably diagnosed faults associated with low airflow, low cooling capacity, and excess outdoor airflow causing lower cooling capacity by 7% or more. The CEC ΔTS protocol was less reliable with combined faults of low airflow and evaporator coil blockage indicating the importance of checking and correcting obvious maintenance faults such as dirty filters or coil blockage before checking RCA.
- For tests where ESP was controlled by the code tester, application sensible cooling capacity and efficiency were reduced by 7 to 29% with 17 to 33% lower airflow. Sensible efficiency was reduced by 3 to 18% with 1-finger open dampers compared to closed dampers.

- For tests where ESP was controlled by supply and return dampers, the optimal airflow and efficiency were achieved with lower than rated airflow (i.e., 318 and 349 scfm/ton). Adjusting fan speed and airflow at the unit caused less impact on efficiency than increasing ESP with the code tester.
- Lower airflow, higher OAT, and overventilation (dampers open 1-finger) had the largest negative impacts on cooling capacity and efficiency.
- Adjusting ESP using the code tester to emulate low airflow faults produced incorrect false alarm findings and is not recommended.

6.1.9 Refrigerant Charge Fault Tests

Refrigerant charge fault tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, 3-ton non-TXV RTU5, and 3-ton TXV RTU4. Laboratory tests were performed with the economizer installed, dampers closed (or 1-finger open), economizer perimeter unsealed, outdoor conditions of 82, 95, or 115F, and indoor conditions of 75F DB and 62F WB.

The first refrigerant charge fault tests for the 7.5-ton non-TXV RTU3 were performed with charge varying from 80 to 156% of factory charge and unequal refrigerant charge percentages per circuit. The first faulted tests were performed in the vertical position. The non-faulted test (i.e., 100% factory charge) was performed in the horizontal position. Total and sensible efficiency were maximized at 100% factory charge. For the first set of tests with closed dampers at 95F OAT, undercharging refrigerant by 20% reduced EER*s by 33% and overcharging by 20 to 56% reduced EER*s by 2 to 4%. Opening dampers from closed to 1-finger increased outdoor airflow from 17 to 24% and reduced EER*s by 3 to 36% depending on outdoor air temperature. The manufacturer suction temperature refrigerant charge protocols correctly identified 20% undercharge tests, but were only 38% accurate due to misdiagnosing overcharge or correct charge as undercharge. The CEC superheat protocols provided similar accuracy with 35% correct diagnoses. The CEC temperature split protocol provided 100% accuracy by correctly diagnosing low capacity for all undercharge tests and all tests at 115F OAT. It also correctly diagnosed proper airflow for all tests at 82F and 95F OAT.

The second refrigerant charge fault tests were performed for the 7.5-ton non-TXV RTU3 with charge varying from 60 to 140% of factory charge and equal refrigerant charge percentages per circuit. All of the second tests were performed in the horizontal position. Total and sensible efficiency were maximized at 120 to 130% factory charge. For the second set of tests at 95F OAT, undercharging by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% increased EER*s by 0 to 2%. The manufacturer suction temperature protocols provided 55% accuracy by correctly diagnosing undercharge for both circuits and overcharge for circuit 2 at 130 to 140% factory charge. The superheat CEC superheat protocols provided 46% accuracy by correctly diagnosing undercharge for both circuits. The CEC temperature split protocol provided 100% accuracy by correctly diagnosing low cooling capacity for all tests less than 110% of factory charge and correct airflow and capacity for 110 to 140% of factory charge.

The 7.5-ton TXV RTU2 refrigerant charge fault tests were performed with charge varying from 60 to 140% of factory charge. The application sensible efficiency was maximized from 100 to 110% of factory charge. Undercharging refrigerant by 10 to 40% decreased EER*s by 1 to 20% and overcharging by 10 to 40% decreased sensible efficiency by 0 to 1%. The manufacturer protocols provided 58% accuracy by correctly diagnosing factory charge and 20 to 40% undercharge. The CEC temperature split protocols provided 100% accuracy by correctly diagnosing proper airflow and cooling capacity for all tests from 30% undercharge to 40% overcharge and low capacity for 40% undercharge.

The 3-ton non-TXV RTU5, refrigerant charge fault tests were performed with charge varying from 60 to 140% of factory charge. Total and sensible efficiency are maximized from 100 to 110% factory charge. Undercharging refrigerant by 10 to 40% reduced EER*s by 0 to 28%, and overcharging by 10 to 40% reduced sensible efficiency by 0 to 3%. Opening dampers from closed to 1-finger increased outdoor airflow from 23.5 to 32.6% and reduced efficiency by 8 to 17% depending on OAT.

The 3-ton TXV RTU4 refrigerant charge fault tests were performed with charge varying from 60 to 140% of factory charge. Total and sensible efficiency are maximized at 100% factory charge. Undercharging refrigerant by 10 to 40% reduced EER*s by 1 to 41% and overcharging by 10 to 40% increased sensible efficiency by 0 to 2%. Opening dampers from closed to 1-finger increased outdoor airflow from 20 to 28% and reduced efficiency by 8 to 19%. The manufacturer subcooling protocol provided 100% accuracy but the overall accuracy was 56% due to discharge/suction pressure protocols correctly diagnosing factory charge and 30 to 40% undercharge but not detecting 10 to 20% undercharge and 10 to 40 overcharge. The CEC subcooling protocols provided 56% accuracy by providing correct diagnosis for -10 to -40% undercharge and +40% overcharge, “false alarm” undercharge at factory charge, and missed detection at +10 to +30% overcharge. The CEC temperature split protocols provided 78% accuracy.

Overall conclusions and observations from the refrigerant charge fault tests are listed below.

- For the first set of tests on the 7.5 ton non-TXV RTU3 in the vertical position at 95F OAT, undercharging refrigerant by 20% reduced EER*s by 34% and overcharging by 20 to 56% reduced EER*s by 2 to 4%. For the second set of tests on the same unit in the horizontal position at 95F OAT, undercharging by 5 to 40% reduced EER*s by 4 to 47% and overcharging by 5 to 40% increased EER*s by 0 to 2%. For tests on other units undercharging by 15 to 30% reduced EER*s by 10 to 14% and overcharging by 5 to 40% reduced EER*s by 0 to 3%.
- The manufacturer refrigerant charge protocols were 53% accurate and the CEC refrigerant charge protocols were 44% accurate with correctly diagnosing refrigerant charge faults. The protocols were most accurate in diagnosing undercharge, but tended to misdiagnose factory charge as undercharge and overcharge as correctly charged. The CEC temperature split protocols were 93% accurate diagnosing low capacity due to charge faults.
- The “optimal” refrigerant charge in terms of EER*s and capacity will generally exceed factory charge at high OAT with excess outdoor airflow.

- For all units tested overcharging produced negligible or slight improvements in efficiency and capacity. However, overcharging will increase discharge pressure and power use and could cause premature compressor failure.

6.1.10 Evaporator Blockage Fault Tests

Evaporator blockage tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, and 3-ton non-TXV RTU5. Evaporator blockage occurs over time as dirt and debris in the return air and outdoor air are deposited on the air filter and coil. Evaporator coil blockage was emulated by blocking the air filter inlet by 5 to 50% with plastic corrugated cardboard to reduce evaporator airflow. Laboratory tests were performed with the economizer installed, dampers closed, economizer perimeter unsealed, outdoor conditions of 95F, and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge.

The 7.5-ton non-TXV RTU3 evaporator blockage faults reduced evaporator airflow by 1 to 12%. Evaporator coil blockage reduced application sensible capacity by 1 to 7%, sensible efficiency by 1 to 4%, and total application efficiency by 1 to 3%. The manufacturer ΔST and CEC ΔSH refrigerant charge protocols incorrectly provided false alarm undercharge diagnostics for each test. The CEC ΔTS protocol identified evaporator blockage as low capacity for all tests except 35 and 50% blockage which are incorrectly diagnosed as proper airflow.

The 7.5-ton TXV RTU2 evaporator blockage faults reduced evaporator airflow by 8 to 18%, application sensible capacity by 13.5%, EER*s by 11.9%, and total EER* by 3.9%. The manufacturer ΔST refrigerant charge protocol correctly diagnosed proper charge for 0 to 30% blockage and provided false alarm overcharge diagnostics for 50% blockage. The CEC ΔSH refrigerant charge protocol correctly diagnosed proper charge for all tests. The CEC ΔTS protocol did not detect 50% evaporator coil blockage even though sensible cooling capacity and airflow were 12 to 14% less than the unblocked baseline test. The misdetection was caused by low airflow increasing temperature split, and coil blockage reducing evaporator heat transfer surface area.

The 3-ton non-TXV RTU5 evaporator blockage faults reduced evaporator airflow by 1 to 13%, EER*s by 1 to 9%, and sensible capacity by 1 to 11%. The manufacturer ΔST refrigerant charge protocol correctly identified proper charge for each test. The CEC ΔSH protocol correctly identified proper charge for all tests up to 20% blockage and indicates a slight undercharge for 35 to 50% blockage tests. The CEC ΔTS protocol identified evaporator blockage as low capacity.

Overall conclusions and observations from the evaporator blockage fault tests are listed below.

- Evaporator coil blockage reduced airflow by 1 to 18%, application sensible capacity by 6 to 14%, sensible efficiency by 5 to 11%, and total efficiency by 1 to 4%.
- Evaporator blockage reduced EER*s more than total efficiency, while low airflow only lowered total application efficiency with virtually no change in sensible efficiency. Evaporator blockage reduced airflow by 1 to 17.5%, sensible efficiency by 1 to 11.2%, and total efficiency by -2.5% to 3.5%.

- Cooling capacity and efficiency impacts due to evaporator coil blockage have been correlated to changes in evaporator airflow.
- Evaporator coil blockage reduces airflow and evaporator heat transfer surface area which make it difficult for temperature split to diagnose the fault.
- These tests indicate the importance of technicians following systematic procedures of checking and correcting obvious maintenance faults such as cleaning the evaporator coil and installing clean air filters before performing FDD services.

6.1.11 Condenser Blockage Fault Tests

Condenser blockage tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, and 3-ton non-TXV RTU5. Condenser coil blockage was emulated by blocking the condenser inlet by 5 to 80% with plastic corrugated cardboard to reduce airflow. Laboratory tests were performed with the economizer installed, dampers closed, economizer perimeter unsealed, outdoor conditions of 95F, and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge.

The 7.5-ton non-TXV RTU3 condenser blockage faults increased discharge pressure by 2 to 33% across both refrigerant circuits. Tests were performed with economizer #4 dampers closed and 330 scfm/ton airflow. Total and sensible application efficiency were maximized with no blockage. Condenser coil blocking decreased EER*s by 2 to 24% and total application efficiency by 7 to 28%. Total power increased by 24%. Both circuits failed manufacturer ΔST refrigerant charge protocols except for 80% blockage where C2 passes. Both circuits failed the CEC ΔSH for all tests. The manufacturer and CEC refrigerant charge protocols misdiagnosed condenser coil blockage as a “false alarm” undercharge based on high suction and superheat temperatures with respect to recommended target values. The CEC ΔTS protocol correctly identified low cooling capacity faults for all condenser coil blockage tests.

The 7.5-ton TXV RTU2 condenser blockage faults increased discharge pressure by 2 to 33% across both refrigerant circuits. Tests were performed with economizer #1 dampers closed and 360 scfm/ton airflow. Total and sensible application efficiency were maximized with no blockage. Blocking the condenser coil by 5 to 80% reduced EER*s by 2 to 26 and increased discharge pressure by 2 to 33% and total power by 1 to 24%. Condenser fan power decreased by 8% up to the third condenser blockage increment (representing a 9% increase in discharge pressure), but as coil blockage increased condenser fan power increased to 10% above the baseline. Both circuits passed manufacturer refrigerant charge protocols in the baseline test. At discharge pressures greater than 4% above the baseline, the manufacturer refrigerant charge protocol misdiagnoses condenser coil blockage as a false alarm overcharge. Discharge pressure and superheat failed the manufacturer protocol when pressure was greater than or equal to 9% above the baseline.

The 3-ton non-TXV RTU5 condenser blockage faults produced a discharge pressure increase of 2 to 30%. Tests were performed with economizer #5 dampers closed and 360 scfm/ton airflow. Total and sensible application efficiencies were maximized with no blockage. Condenser coil blockage decreased sensible efficiency by 1 to 25% and total efficiency by 2 to 36%. Total power increased by 23%. The refrigerant system passed manufacturer ΔST for 0 to 5% blockage and CEC ΔSH for 0 to 10% blockage. The manufacturer and CEC protocols indicated a false alarm undercharge for all other tests based on high suction and superheat temperatures with respect to recommended target values. The manufacturer and CEC refrigerant charge protocols misdiagnosed condenser coil blockage as a "false alarm" undercharge based on high suction and superheat temperatures with respect to recommended target values. The CEC ΔTS protocol correctly identified low cooling capacity faults for all condenser coil blockage tests.

Overall conclusions and observations from the coil blockage fault tests are listed below.

- Condenser coil blockage decreased discharge pressure by 2 to 49% at constant OAT and lowered sensible and total application efficiency by about the same amount as the coil blockage discharge pressure increase. For example, a 10% increase in discharge pressure due to condenser coil blockage caused sensible efficiency to decrease by 9% and total efficiency to decrease by 11%.
- The manufacturer refrigerant charge protocols misdiagnosed condenser coil blockage as a false alarm overcharge based on high discharge pressure and undercharge based on high superheat with respect to recommended target values.

6.1.12 Restriction Fault Tests

Restriction tests were performed on the 7.5 ton non-TXV RTU3 and 7.5-ton TXV RTU2. Restriction fault tests were performed by partially closing a "service" valve installed upstream of the liquid line driers on each circuit to cause reductions in suction pressure and liquid line temperature to emulate a restriction at the liquid line drier or expansion device. Tests were performed with the economizer installed, economizer perimeter unsealed, dampers closed, outdoor conditions of 82, 95, and 115F, and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge. In general, restrictions lower evaporator saturation temperature and increase superheat causing restricted circuits to be misdiagnosed as undercharged.

The 7.5-ton non-TXV RTU3 circuit 1 restriction caused sensible efficiency to decrease by 12 to 36% and cooling capacity to decrease by 15 to 39% depending on OAT. Tests were performed with economizer #4 installed and airflow from 267 to 400 scfm/ton. The circuit 1 restriction tests imposed a 34 to 44 psig suction pressure drop and 15 to 20F liquid temperature drop. The suction pressure decreased by 45 to 66% and evaporator saturation temperature decreased by 59 to 95% depending on OAT conditions. The manufacturer provides information for "troubleshooting" and diagnosing refrigerant restrictions from five other faults including refrigerant undercharge. If the liquid line temperature 12 to 24 inches upstream of the TXV entrance is 2 to 3F colder than ambient air, then there is a restriction upstream. If the temperature drop across the filter drier is greater than 3F, then there is a filter drier restriction. Tests for no restriction found a 1 to 2F filter-drier inlet minus outlet temperature increased while the C1 restriction caused a 17 to 18F temperature decrease.

The 7.5-ton TXV RTU2 circuit 1 restriction caused sensible efficiency to decrease by 8 to 14% and cooling capacity to decrease by 11 to 17% depending on OAT. Tests were performed with economizer #1 installed and airflow at 400 scfm/ton. Circuit 1 restriction tests imposed a 15 to 28 psig suction pressure drop and 15 to 20F liquid temperature drop. The suction pressure decreased by 19 to 39% and evaporator saturation temperature was decreased by 22 to 51% depending on OAT conditions.

Overall conclusions and observations from the restriction fault tests are listed below.

- Restrictions lowered suction pressure and evaporator saturation temperature (typically below 32F causing coil icing) and increased superheat causing restricted circuits to be misdiagnosed as undercharged.
- Circuit 1 restriction on the 7.5 ton non-TXV caused a 17 to 20F temperature drop across the filter drier and impacted efficiency by two or three times more than the circuit 1 restriction on the 7.5 ton TXV system which caused a 16 to 21F temperature drop across the filter drier. Unrestricted filter driers exhibited either no temperature drop or a 1F temperature increase.
- The manufacturer provides information for "troubleshooting" and diagnosing refrigerant restrictions from five other faults including refrigerant undercharge. If the liquid line temperature 12 to 24 inches upstream of the TXV entrance is 2 to 3F colder than ambient air, then there is a restriction upstream. If the temperature drop across the filter drier is greater than 3F, then there is a filter drier restriction.

6.1.13 Non-Condensable Fault Tests

Non-condensable tests were performed on the 7.5 ton non-TXV RTU3 and 7.5-ton TXV RTU2. Tests were performed with economizer installed, economizer perimeter unsealed, dampers closed, outdoor conditions of 82, 95, and 115F, and indoor conditions of 75F DB and 62F WB. All tests were performed with factory charge.

The 7.5-ton non-TXV RTU3 non-condensable fault tests were performed by adding 0.4 ounces of nitrogen to circuit 1 to emulate non-condensable air and water vapor faults. The weight of nitrogen was normalized with respect to the factory charge (oz/oz) so 0.4 ounces of nitrogen represents 0.33% of the factory charge. Tests were performed with economizer #4 installed and airflow at 360 scfm/ton. The circuit 1 NC caused application efficiency to decrease by 13 to 19%, cooling capacity to decrease by 10 to 14%, and total power to increase by 5 to 6%. Circuit 1 NC increased DP by 18 to 27% depending on OAT conditions. The manufacturer provides information for "troubleshooting" and diagnosing non-condensables from nine other faults including refrigerant undercharge.

The 7.5-ton TXV RTU2 non-condensable fault tests were performed by adding 0.5 ounces of nitrogen was alternately added to circuit 1 and 2, for a total of 1 ounce added per circuit or 2 ounces total for the two-circuit system. The weight of nitrogen is normalized with respect to the factory charge (oz/oz) so 0.5 to 2 ounces of nitrogen represents 0.25% to 1% of the factory charge. Tests were performed with economizer #1 installed and airflow at 360 scfm/ton. The NC reduced application efficiency by 9 to 22% and cooling capacity by 4 to 7% and increased total power by 10 to 20%. NC increased DP by 18 to 29% depending on how much NC was added to each circuit.

Laboratory tests performed in 2010 of a 3-ton R22 split-system with only the condenser fan operating indicated 45 minutes were required for the saturation temperature to increase by 3F or more above the condenser entering OAT to identify 1% non-condensable nitrogen in the refrigerant system. The test took 45 minutes for discharge and liquid line temperatures to reach equilibrium with the condenser entering OAT and for non-condensable nitrogen to coalesce in the condenser from being more dispersed throughout the system after the compressor and evaporator fan are turned off.

Overall conclusions and observations from the non-condensable fault tests are listed below.

- The 7.5 ton non-TXV application efficiency impact was 16% from 0.33% non-condensable nitrogen per factory charge added to circuit 1 compared to the 7.5-ton TXV application efficiency impact of 9% from adding 0.25% nitrogen per factory charge to circuit 1.
- For the 7.5-ton non-TXV system, EER*s decreased by 13 to 17%, sensible capacity decreased by 10 to 14%, and compressor power increased by 6 to 7% from adding 0.33% nitrogen per factory charge to circuit 1.
- For the 7.5-ton TXV system, the efficiency decreased by 9 to 22%, application sensible cooling capacity decreased by 4 to 7%, discharge pressure increased by 18 to 29%, and compressor power increased by 8 to 26% from adding 1% non-condensable nitrogen per factory charge to both circuits.
- For the 7.5-ton non-TXV system, the manufacturer and CEC refrigerant charge protocols misdiagnosed non-condensables as an undercharge because they do not include discharge pressure or refrigerant liquid line temperature measurements which are required to evaluate condenser heat transfer faults (i.e., non-condensables or condenser blockage).
- The manufacturer provides “troubleshooting” procedures for diagnosing non-condensables from nine other faults including refrigerant undercharge. Testing for non-condensables with only the condenser fan operating takes about 45 minutes for discharge and liquid line temperatures to reach equilibrium with OAT and for non-condensables to coalesce in the condenser after the compressor and evaporator fan are turned off.

6.1.14 Multiple Fault Tests

Multiple fault tests were performed on the 7.5 ton non-TXV RTU3, 7.5-ton TXV RTU2, and 3-ton non-TXV RTU5. Multiple faults include 20% evaporator coil blockage, 30% condenser coil blockage, 80-120% factory charge, 65-110% airflow, C1 refrigerant restriction, and 2 -fingers open dampers.

The 7.5-ton non-TXV RTU3 multiple faults tests included 80-120% factory charge, 30% condenser blockage, and 20% evaporator coil blockage. For 80% charge, 20% evaporator blockage and 30% condenser blockage the measured sensible efficiency decrease was 23% and predicted decrease was 25%. For 100% charge, 20% evaporator blockage and 30% condenser blockage the measured and predicted sensible efficiency decrease was 10%. For 120% charge, 20% evaporator blockage and 30% condenser blockage the measured and predicted sensible efficiency decrease was 8%. The manufacturer ΔST and CEC ΔSH protocol correctly identified charge faults for 35% of tests with multiple faults. The CEC ΔTS protocol correctly identified proper airflow and low capacity faults for 100% of tests with multiple faults.

The 7.5-ton TXV RTU2 multiple fault tests were performed with 85-115% factory charge, 30% condenser coil blockage, 65-110% airflow, C1 restriction, and 2-fingers open outdoor air. For 85% charge and 30% condenser blockage the measured sensible efficiency decrease was 12% and the predicted decrease was 11%. For 85% charge, 30% condenser blockage, and C1 restriction the measured sensible efficiency decrease was 29% and the predicted decrease was 35%. The condenser blockage increased DP forcing more refrigerant through C1 restriction to mitigate its impact. For 115% charge and 30% condenser blockage the measured sensible efficiency decrease was 11% and the predicted decrease was 9%. For 115% charge, 30% condenser blockage, and C1 restriction the measured decrease was 23% and predicted decrease was 33%. For 100% charge, 30% condenser blockage, and C1 restriction the measured sensible efficiency decrease was 25% and predicted decrease was 34%. The manufacturer ΔDP and ΔSP protocols correctly identified charge faults for 56% of tests with multiple faults. The CEC ΔSH protocol correctly identified charge faults for 38% of tests with multiple faults. The CEC ΔTS protocol correctly identified proper airflow, low airflow, and low capacity faults for 84% of tests with multiple faults.

For the 3-ton non-TXV RTU5 tests were performed with multiple faults including 80-120% factory charge, 30% condenser blockage, and 20% evaporator coil blockage. For 80% charge, 30% condenser blockage, and 20% evaporator blockage the measured sensible efficiency decreased by 17% and predicted decrease was 19%. For 100% charge, 20% evaporator blockage and 30% condenser blockage the measured sensible efficiency decreased by 14% and predicted decrease was 13%. For 120% charge, 20% evaporator blockage and 30% condenser blockage the measured sensible efficiency decreased by 15% and predicted decrease was 14%. The manufacturer ΔST correctly identified charge faults for 80% of tests. The CEC ΔSH protocol correctly identified charge faults for 60% of tests with multiple faults. The CEC ΔTS protocol correctly identified proper airflow and low capacity faults for 100% of tests with multiple faults.

Overall conclusions and observations from the multiple fault tests are listed below.

- Tests of multiple faults for the 7.5-ton non-TXV, 7.5-ton TXV, and 3-ton non-TXV units found an average difference of 0.2% between measured impacts and predicted impacts based on the sum of individually tested faults.
- Multiple faults do not appear to mitigate the impacts of individual faults except for condenser coil blockage and refrigerant restrictions where condenser blockage increased discharge pressure forcing more refrigerant through an imposed restriction, thus mitigating its impact. However, if the condenser was cleaned the full impact of the restriction would reoccur.

- Troubleshooting multiple faults through a logical progression will reduce or eliminate “false alarms,” misdetection, and misdiagnosis.

6.1.15 Refrigerant Hose Tests

Attachment and detachment tests of refrigerant hoses without EPA low-loss fittings to suction and discharge line Schrader valves found 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and efficiency per attachment/detachment.

6.1.16 Temperature Measurement Tests

Temperature measurement instruments tests indicate certain sensors provide readings that will lead to inaccurate fault diagnostics. Tests were conducted with eight sensors on liquid and suction lines. Largest differences were found for suction line measurements where tube temperatures are 25 to 40F less than outdoor ambient temperature. The liquid line temperature is typically 8 to 12F above ambient temperature so there are smaller variations from measured temperatures to actual tube temperatures. Smallest differences were found for specific Type-K thermocouple clamp probes with accuracy ranging from 1.1 +/- 0.6F on suction lines at 115F. Some Type-K clamp probes have suction line accuracy ranging from 6.8 +/- 1.0F when tested at 115F outdoor conditions. Differences in accuracy are attributable to design (i.e., sensor, clamps, thermal contact, insulation, etc.) and manufacturing (quality of materials, fit, finish, operability. durability, etc). Largest differences were found for Type-K thermocouple bead probes and thermistors: insulated bead probes had differences of 10.7 +/- 3.3F, insulated cylindrical thermistors had differences of 9.7 +/- 7.1F, and clamp thermistors had differences of 5.4 +/- 2.1F. Tests of measurement instruments indicate that it can take 5 to 10 minutes or longer for sensors to reach steady-state and correctly measure correct air and refrigerant temperatures. Not allowing sensors to reach steady-state can cause inaccurate measurements and fault diagnostics and we know that field technicians are under pressure to complete their work as quickly as possible.

6.1.17 Air Measurement Instrument Tests

Supply and return air measurement tests could be performed with proper factory charge at outdoor conditions of 82F, 95F, and 115F and indoor conditions of 75DB and 62F WB. There are approximately 93 instruments or sensors to test from 5 manufacturers.

6.1.18 Pressure Measurement Instrument Tests

Pressure measurement instrument tests were performed at five liquid and suction pressures. There were approximately 63 instruments or sensors to test from 8 manufacturers. Worst case measurements were performed with refrigerant in hoses outfitted with EPA low-loss fittings left to soak in hot 130F chamber to strain sensors. Two digital pressure manifolds were found to be leaking refrigerant when taken out-of-the-box. These were removed from the sample. Preliminary laboratory tests of 15 digital and 7 analog field pressure measurement instruments have been completed. Best case average difference between laboratory and digital instruments is 0.57 +/- 0.24% based on measurements at ten different pressures with 15 instruments from 6 manufacturers. Best case average difference in accuracy between laboratory and analog instruments is 1.76 +/- 0.57% based on measurements at ten different pressures with 7 instruments from 2 manufacturers.

6.1.19 Airflow Measurement Instrument Tests

Airflow measurement instrument tests will be performed with airflow ranging from 1,200 to 3,000 scfm at 95F outdoor conditions and indoor conditions of 75F DB and 62F WB. There are approximately 25 instruments or sensors to test from 8 manufacturers. The worst case measurements were performed with airflow measurements taken at non standard locations. Preliminary laboratory tests of the Pitot-tube array airflow grid from 1 manufacturer have been completed. The average difference between laboratory and field measurement instruments is 10.2 +/- 0.64% based on three measurements at 2,000, 2,500, and 3,000 cfm (i.e., field measurement instruments were 10.2% lower than Intertek measurement instruments).

6.1.20 Vacuum Pump Measurement Instrument Tests

Vacuum pump measurement instruments will be performed with no vacuum/liquid drier, 30 minute vacuum with drier, 60 minute vacuum with drier. There are 4 vacuum pumps and 3 micron gauges to test from 7 manufacturers. In order to evaluate the efficiency impact associated with each evacuation method, tests will be performed with airflow at approximately 350 to 400 scfm/ton and outdoor conditions of 95F and indoor conditions of 75F DB and 62F WB.

6.1.21 Fan Belt Measurement Instrument Tests

Fan belt tension and alignment measurement instruments tests could be performed on the 3-ton units. There are 14 belt tension and alignment instruments to test from 5 manufacturers. Belts will be tested with proper tension and alignment, as well as loose and tight tension and misalignment of 0.25 and 0.375 inches at the following conditions 80DB/67WB/95F. The worst case measurements will be performed with fan belt tension either loose or tight and the belt misaligned by either ¼ or 3/8 inches. Out-of-box fan belt tests indicated tension was looser than manufacturer recommendations, but belts were properly aligned.

6.1.22 Cold Weather Charging Hood and Digital Refrigerant Scale Tests

The cold weather charging jacket is designed to work at outdoor drybulb temperatures ranging from 37 to 70F.[155] Laboratory tests of the cold weather charging jacket will be performed with outdoor conditions of 37F, 55F, and 70F and indoor conditions of 75F DB and 62F WB. Digital refrigerant scale tests will be performed at outdoor conditions of 95F. There are 4 instruments to test from 1 manufacturer. The charging scale will be tested using known weights or 1, 5, 10, 15, 25, 50, and 100 pounds to +/- 0.25 ounces.

The charging hood is designed to block the condenser outlet and increase discharge pressure to emulate warmer OAT conditions for testing TXV-equipped systems using the subcooling method. The charging Jacket is designed to be attached to the top of the condenser with hooks, so the air inlet is not restricted. The system is turned and the jacket fills up with air from the condenser air outlet. The top of the charging hood is adjusted with a drawstring until the proper pressure differential is reached between the high and low side (between 160psi and 220psi for R410A; and between 100psi and 145psi for R22). The manufacturer indicates that after the air conditioning system must stabilize for 15 minutes at the recommended high-to-low-side pressure differential before checking refrigerant charge with the subcooling method only.

According to the manufacturer technicians use cardboard or other materials to block the condenser air outlet to perform refrigerant charge diagnostic testing at cold outdoor conditions. The manufacturer indicates that the charging hood can be used to emulate recommended high-to-low-side-pressure differences in order to test refrigerant charge for a TXV-equipped air conditioner.

6.1.23 Using Laboratory Data for Load Impact Evaluations

Laboratory tests were performed to provide information regarding how to use laboratory data for load impact evaluations of the following HVAC maintenance measures: refrigerant charge adjustments, condenser blockage, evaporator blockage, and economizer perimeter sealing.

Refrigerant charge regression equations based on laboratory test data can be used to calculate EER*s impacts based on recovery and weigh-out of refrigerant charge and the reported charge adjustment per circuit. Field observations of 35 units found an average difference of 15.1 +/- 3.2% between recovered and pre-existing refrigerant charge. This corresponds to an average

[155] http://www.fieldpiece.com/products/detail/s365-charging-jacket-for-txv-systems/cold-weather-charging/

sensible efficiency impact of 8.9 +/- 3.9% using regression equations based on laboratory tests of EER*s versus refrigerant charge faults per factory charge.

Evaporator blockage regression equations based on laboratory tests can be used to calculate EER*s impacts based on airflow measurements before and after removing dirty filter (DF) or dirty coil (DC) only blockage. Field observations of 4 units with dirty coils and filters cleaned 12.6 +/- 0.8 months earlier found an airflow improvement of 5.2 +/- 2.4%. This is equivalent to 27% blockage and 4.4% EER*s impact based on laboratory tests. Quarterly maintenance would cause a 1.1% EER*s impact and semiannual maintenance would cause a 2.1% EER*s impact. The difference between these impacts corresponds to an EER*s improvement of 2.3 to 3.3% for one year. The EUL depends on frequency of coil cleaning and filter replacement. Dirty filters cause even larger EER*s impacts in the field due to longer operational times which were not measured in the laboratory.

Condenser blockage regression equations based on laboratory tests can be used to calculate EER*s impacts based on discharge pressure ratios before and after removing the blockage. Field measurements of 28 units found an average DP impact of 4.8 +/- 0.7% based on cleaning dirty coils previously cleaned 12.6 +/- 0.8 months earlier. This corresponds to an average EER*s impact of 4.1 +/- 0.6% equivalent to 17% blockage. Semi-annual maintenance would cause a 2% EER*s impact. The difference corresponds to an EER*s improvement of 2.1 +/- 0.3% due to enhanced HVAC maintenance services. The EUL depends on frequency of coil cleaning.

Economizer perimeter sealing (EPS) can be evaluated based on field measurements of the OAF before and after removing the hood and sealing the perimeter where it attaches to the cabinet. Laboratory tests found an average OAF difference of 6 +/- 2% resulting in an EER*s improvement of 5.4 +/- 2%. **Equation 1** can be used to calculate the pre- and post-OAF with compressors operating if the outdoor minus return air temperature is at least 20F. **Equation 26** can be used to evaluate EER*s impacts based on the sealed and unsealed perimeter OAF difference and minimum damper position.

6.2 Recommendations

The following recommendations are provided to improve HVAC maintenance programs based on three years of laboratory test results.

6.2.1 Include Outdoor Ventilation within HVAC Maintenance

Reducing overventilation or unintended outdoor air leakage through economizer dampers and the perimeter frame can increase cooling and heating capacity and improve energy efficiency.. Optimizing minimum damper position to meet ASHRAE 62.1 outdoor air ventilation requirements or sealing unintended leakage around the economizer perimeter can improve cooling efficiency by 5 to 24%. Sealing the economizer perimeter requires removal of the economizer hood, sealing the gap between economizer perimeter frame and cabinet with UL-181 metal tape and reinstalling the hood. Heating energy efficiency impacts due to overventilation and cabinet leakage at various outdoor air temperatures have not been tested.[156]

6.2.2 Develop AHRI Tests to Emulate Field Conditions

AHRI test procedures need to be developed to provide consistent and accurate rating information to better emulate field conditions including more realistic external static pressure, fan speeds, and the impact economizer outdoor air leakage and ventilation outdoor airflow have on cooling and heating capacity and energy efficiency. The tested out-of-box efficiency without an economizer is 18 to 26% less than the rated efficiency due to factory fan speeds providing much higher airflow and static pressure than allowed under the ANSI/AHRI test procedures. The tested out-of-box efficiency with an economizer with closed dampers is 38 to 53% less than the rated efficiency.

6.2.3 Consider Manufacturer Protocols and Troubleshooting Procedures for Fault Detection and Diagnosis

Unit- or circuit-specific troubleshooting and fault detection diagnostic procedures are currently available from manufacturers to improve HVAC installation and maintenance. Manufacturer training and certification needs to be made available through widespread workforce education and training efforts in order to overcome market barriers and improve technician competence. Laboratory tests indicate manufacturer troubleshooting procedures can be effective if used in a systematic manner to diagnose faults such as: excessive outdoor air, low cooling/heating capacity, blocked condenser/evaporator, refrigerant restrictions, non-condensables, and refrigerant overcharge or undercharge. Laboratory tests indicate current manufacturer troubleshooting procedures and refrigerant charge protocols will likely be less effective at diagnosing low airflow from undercharge, and refrigerant charge protocols in general cannot diagnose other faults. Laboratory tests also indicate that checking evaporator temperature split should not be performed unless air filters and coils are clean indicating the importance of checking and correcting obvious maintenance faults before checking temperature split and refrigerant charge. The systematic application of manufacturer troubleshooting procedures should be performed to determine whether or not any other faults are present before evaluating refrigerant charge. Otherwise, non-refrigerant charge faults that might have a larger impact on thermal comfort, energy efficiency, or indoor air quality might not be diagnosed or corrected. If refrigerant charge faults are identified as the most probable cause of a problem, then unit-specific

[156] Manufacturers indicate that inefficient or inadequate space heating or continuous operation of furnace can be caused by dirty air filters, restricted airflow, or too much outdoor air. Manufacturers recommend cleaning or replacing air filters, removing airflow restrictions, or adjusting economizer minimum outdoor air damper position to increase heating capacity and efficiency. Reducing or closing outdoor air damper position improves heating efficiency, economizer savings, and satisfies ASHRAE 62.1 for most buildings.

or circuit-specific high-side and low-side refrigerant charge FDD protocols should be used (if available). While some manufacturer refrigerant charge protocols include discharge and suction pressure as well as suction temperature, superheat and subcooling, most do not. Manufacturer troubleshooting procedures and refrigerant charge protocols are generally more comprehensive than generic protocols that only evaluate superheat or subcooling. Due to being tested under conditions and with faults they were not intended to diagnose, the accuracy of manufacturer refrigerant charge protocols was 48 +/- 3% based on a sample of 992 measurements. For similar reasons, the generic CEC refrigerant charge protocol accuracy was 31 +/- 4% based on 445 measurements. Manufacturer unit-specific RC protocols generally diagnose more parameters on both the high- and low-side providing greater accuracy than generic CEC RC protocols.

6.2.4 Reduce Overventilation and Unintended Outdoor Air Leakage before Conducting Refrigerant Charge Tests

Reducing overventilation and unintended outdoor air leakage should be the most important priority for HVAC maintenance programs and this important task must be conducted before performing any refrigerant charge tests. Laboratory test results indicate the importance of reducing or eliminating excess outdoor airflow by closing and sealing dampers when checking refrigerant charge diagnostics. Temperature split tests based on well-mixed return and supply drybulb and wetbulb temperatures can be used to identify low cooling capacity and efficiency due to overventilation or unintended outdoor air leakage.

6.2.5 Caution Using Subcooling Diagnostics for Units with Only Discharge Pressure Schrader Valves

Subcooling diagnostic methods should not be used on units with only discharge pressure Schrader valves. Most manufacturers of packaged units equipped with TXV expansion devices do not provide Schrader valves to measure liquid pressure required to evaluate subcooling. Therefore, generic subcooling protocols are generally not recommended for packaged unit refrigerant charge FDD. Diagnosing refrigerant charge faults with one diagnostic parameter such as subcooling cannot be used to identify other important faults that impact cooling capacity and total system efficiency such as overventilation or untended outdoor air leakage, low suction pressure, low evaporator airflow, low fan speed, dirty air filter, evaporator coil blockage, non-condensables, and metering device or filter drier restrictions.

6.2.6 Temperature Split Diagnostics

Temperature split (TS) protocols were correct 90 +/- 2% of the time when diagnosing low airflow or low sensible cooling capacity due to overventilation or other maintenance faults. Laboratory tests indicate the TS protocols can be used to evaluate low airflow and low sensible cooling capacity due to overventilation, refrigerant over/undercharge, refrigerant restrictions, non-condensables, and multiple faults.

6.2.7 FDD Protocols

Generic one-parameter refrigerant charge FDD protocols (i.e., superheat or subcooling only) frequently provide “false alarm” undercharge diagnostics and misdiagnose overcharge as correct charge when no other faults are present or when multiple faults are present. This can cause technicians to overcharge units which can reduce efficiency and compressor life. Additional research is necessary to evaluate and provide recommendations to improve FDD protocols and troubleshooting procedures.

6.2.8 Consider Multiple Faults in FDD Procedures

Laboratory tests found 16 to 27% efficiency impacts due to the combination of multiple faults such as low charge, evaporator coil blockage (causing low airflow), and condenser coil blockage. Unintended outdoor airflow, restrictions, or non-condensables reduce efficiency even more. Predicted versus measured efficiency impacts for multiple faults are 0.3% greater indicating predicted multiple fault impacts based on summing individual impacts are roughly equivalent to measured impacts. This is an important finding since HVAC maintenance involves multiple repairs and ex ante savings are typically summed for each repair. The presence of multiple faults can mask individual faults, requiring a systematic approach to FDD protocols that follows a logical set of troubleshooting procedures that technicians can implement to properly diagnose and detect faults and make corrections to improve energy efficiency. For example, evaporator and condenser coils must be properly cleaned, new air filters installed, economizer damper position optimized, unintended outdoor air leakage repaired (through loose cabinet panels or unsealed economizer perimeter), expansion devices and filter driers checked, and evaporator temperature split checked prior to performing refrigerant charge FDD. Otherwise, refrigerant charge FDD might yield false alarms and unnecessary adjustments causing reduced or unchanged energy efficiency.

6.2.9 Require EPA Low-Loss Fittings in Programs

Tests of attachment and detachment of refrigerant hoses without EPA low-loss fittings to suction and discharge line Schrader valves indicate 0.4 to 0.5% loss of factory charge and 0.2% reduced cooling capacity and efficiency per attachment/detachment. Section 608 of the EPA Clean Air Act requires low-loss fittings which should be required under all rate-payer funded energy efficiency programs.

6.2.10 Test Field Measurement Equipment Specifications

Field measurement instrument tests found problems with suction line field measurements where tube temperatures were 25 to 40F less than outdoor ambient temperature. Smallest differences were found for specific Type-K thermocouple clamp probes with accuracy ranging from 1.1 +/- 0.6F on suction lines at 115F. The HVAC industry should work cooperatively to develop accurate FDD and field test equipment specifications to certify FDD troubleshooting procedures, probes, sensors, and test equipment accuracy.

6.2.11 Tests of HVAC Units to Improve Test Results

Additional laboratory tests are recommended for the following measures: refrigerant charge tests at varying OAT and damper positions, low airflow at varying OAT, airflow, ESP, fan speed, and damper positions, evaporator and condenser coil blockage, non-condensables, restrictions, economizer efficiency, economizer outdoor airflow at 55F and 95F unsealed and sealed. Some tests should be performed at lower outdoor wet bulb temperatures. Test conditions should be expanded to develop system performance maps for faulted equipment suitable for use in building energy simulation programs. Additional tests should be performed of multiple fault combinations.

6.2.12 Test Units in Vertical Airflow Configuration

Most packaged rooftop units are installed with vertical airflow connections, while laboratory tests are conducted with horizontal airflow connections. Testing HVAC units in the vertical configuration might be necessary to evaluate economizer performance, efficiency, and response to fault conditions and FDD protocol accuracy compared to the horizontal configuration.

6.3 Recommendations for Additional Research

6.3.1 Additional Laboratory Tests on Field Instruments

Additional field instrument tests should be conducted on low-loss fitting connections and disconnections on units with liquid pressure valves, outdoor airflow field measurement strategies, and cold weather charging hoods. Advanced FDD systems should be tested to evaluate general applicability to HVAC maintenance and troubleshooting procedures. Tests should be performed on the cold weather charging hood to evaluate accuracy and effectiveness especially with other faults including coil blockage, improper refrigerant charge, economizer outdoor air, restrictions, and non-condensables.

6.3.2 Evaluate FDD Test Equipment and Algorithms

Laboratory tests described in this report provide data for emulated single- and multiple-faults with measured system performance parameters. These data can be used to evaluate the accuracy of various FDD protocols either currently in use or under consideration by HVAC maintenance programs. On-board diagnostic systems and protocols can be similarly evaluated using the data. Tests of FDD protocols for economizer testing and evaluation are also required to supplement the FDD protocols included or under consideration for use in the HVAC maintenance programs.

6.3.3 Conduct Transient Tests

Transient non-steady-state tests of economizers need to be performed to evaluate economizer damper controls and functionality to understand economizer, sensor, and thermostat control integration. Transient tests should be conducted at part load conditions to understand part-load performance and improve engineering methods to evaluate the effects of part-load operation on system efficiency.

6.3.4 Expand Manufacturer, Model, and Size of Units Tested

The four units tested and reported in this study represent a cross-section of the equipment encountered in the HVAC maintenance programs. Laboratory tests on additional units should be conducted to expand the coverage of systems tested and applicability of results to program activities. The tests should be expanded to include heat pumps and to investigate the impact of fault conditions for cooling and heating modes. The units tested thus far are “standard efficiency” models. Tests could be expanded to examine impacts of maintenance faults on high-efficiency packaged units.

6.3.5 Tests R-410a and Alternative Refrigerant Systems

Completed tests focused on R-22 systems, since these systems represented the majority of units participating in HVAC maintenance programs. As programs start to service more R-410a systems and alternative refrigerants (NU-22, RS-44), the unique characteristics of these systems and refrigerant properties should be examined in the laboratory. In particular, R-410a systems generally use micro-channel heat exchangers (MCHE). Manufacturers claim MCHE systems may be more sensitive to incorrect charge, non-condensables, restrictions, or coil blockage than systems that use more conventional heat exchangers. MCHE systems use require 20- 40% less refrigerant and are about 10% more efficient than conventional tube and fin condensers.[157]

[157] Carrier, 2007, Commercial Documentation on Microchannel Heat Exchangers. www.carrier.com. Cremaschi,

6.3.6 Conduct Gas Heating Mode Tests on Furnaces

Current laboratory tests have focused on package HVAC systems with gas furnaces operating in the cooling mode. The impact of system faults on gas furnace efficiency should be investigated especially overventilation and excess outdoor airflow compared to optimal ASHRAE 62.1 ventilation requirements.

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8 APPENDIX A: DATA DICTIONARY

The data dictionary defines key performance metrics (entity name) derived from Intertek test files, and a glossary to define acronyms and technical terms. Key performance metrics include name, type, and range definitions and relationships to tabs in the Intertek test files.[158] Intertek test file information provides number of records and fields per tab.[159] Each Intertek test file contains 10 primary records (required by AHRI), 30 to 50 secondary records, and thousands of tertiary records. The data dictionary defines all primary and secondary Intertek data file records and some important tertiary records. Most tertiary records are used to calculate or check primary and secondary records.

Name	Type	Range	Relationship	Definition
ACCA	Character	20	NA	Air Conditioning Contractors of America www.acca.org
AHRI	Character	20	NA	Air-conditioning Heating and Refrigeration Institute www.ahrinet.org
AMCA	Character	20	NA	Air Movement and Control Association www.amca.org
ANSI	Character	20	NA	American National Standards Institute www.ansi.org
ASHRAE	Character	20	NA	American Society of Heating, Refrigerating, and Air-Conditioning Engineers www.ashrae.org
Application Rating	Number	0 to 50	Intertek Summary Tab	ANSI/AHRI Standard 210/240 or 340/360 application rating based on tests performed at Application Rating Conditions (other than Standard Rating Conditions).
AHRI "A" rating	Number	0 to 50	Intertek Summary Tab	ANSI/AHRI Standard 210/240 or 340/360 rating at steady-state operation with an ambient temperature of 95ºF [35C] and return air conditions to the cooling coil of 80ºF [26.7C] dry bulb temperature and 67ºF [19.4C] wet bulb temperature.
AHRI "B" rating	Number	0 to 50	Intertek Summary Tab	ANSI/AHRI Standard 210/240 or 340/360 rating at steady-state operation with an ambient temperature of 82ºF [27.8C] and return air conditions to the cooling coil of 80ºF [26.7C] dry bulb temperature and 67ºF [19.4C] wet bulb temperature.
AHRI "C" rating	Number	0 to 50	Intertek Summary Tab	ANSI/AHRI Standard 210/240 or 340/360 rating at steady-state dry coil operation with an ambient temperature of 82ºF [27.8C] and return air conditions to the cooling coil of 80ºF [26.7C] dry bulb temperature and 57ºF [13.9C] wet bulb temperature.
AHRI "D" rating	Number	0 to 50	Intertek Summary Tab	ANSI/AHRI Standard 210/240 or 340/360 rating at cyclic dry coil operation with an ambient temperature of 82ºF [27.8C] and return air conditions to the cooling coil of 80ºF [26.7C] dry bulb temperature and 57ºF [13.9C] wet bulb temperature to determine cooling mode cyclic degradation coefficient, Cd.

158 The Intertek test files are available at www.calmac.org.

159 Location of data in the Intertek test file tabs varies depending on number of records and fields of data collected.

Name	Type	Range	Relationship	Definition
ΔT_{fan}	Number	0 to 20	Derived from Intertek Summary Tab Data	Temperature increase of air due to fan heat excluding mechanical work causing air movement (F or C).
C1	Number	1	Intertek Compressor 1 Tab	Circuit 1 of a multiple circuit compressor air conditioning system.
C2	Number	2	Intertek Compressor 2 Tab	Circuit 2 of a multiple circuit compressor air conditioning system.
C_d	Number	0 to 1	Intertek Summary Tab	Cyclic degradation coefficient measures efficiency loss due to cycling which is the lower of tested value or default of 0.25 as defined in Appendices C and D of ANSI/AHRI 2008 Standard for Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment Standard 210/240.
CEC	Character	20	NA	California Energy Commission
CFM	Number	0 to 50000	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Cubic Feet per Minute is the measurement of volumetric flow rate of air in a duct system, dampers or through unintended leakage openings in the cabinet or ducts (ft^3/min)
CPUC	Character	20	NA	California Public Utilities Commission
CST	Number	-50 to 250	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Condenser Saturation Temperature of refrigerant based on liquid pressure measured at service valve at steady-state operation (F).
DB	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Dry Bulb Temperature of air volume measured using a thermometer freely exposed to air indicating amount of heat in air and proportional to mean kinetic energy of air molecules. Temperature is measured in degrees Fahrenheit (°F or F), Celsius (°C or C), or Kelvin (K).
DEER	Character	20	NA	Database for Energy Efficiency Resources www.deeresources.com
DP	Number	0 to 600	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Discharge Pressure of refrigerant leaving compressor measured at steady-state operation (psig).
DT	Number	0 to 400	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Discharge Temperature of refrigerant leaving compressor measured at steady-state operation (F).

Name	Type	Range	Relationship	Definition
$\dot{E}_{loss\ fan}$	Number	0 to 10000	NA	Mechanical heat loss of fan heat added to air stream as it passes across the fan motor excluding mechanical energy causing airflow (W)
Economizer	Character	20	Derived from Intertek Power Tab Data	Electro-mechanically controlled damper system attached to a packaged HVAC system designed to provide minimum outdoor airflow per ASHRAE 62.1 when OAT is greater than economizer changeover setting and maximum outdoor airflow to save energy and cool conditioned space instead of compressor-based cooling when OAT is less than the changeover setting.
EER or EER_A	Number	0 to 50	Intertek Summary Tab	Energy Efficiency Ratio of total (sensible plus latent) cooling capacity (Btu/hour) divided by total system power (Watts) at steady-state operation and AHRI Standard Rating Conditions of at 95F OAT and indoor conditions of 80F DB and 67F WB (Btuh/Watt).
EER_B	Number	0 to 50	Intertek Summary Tab	Energy efficiency rating of total (sensible plus latent) cooling capacity (Btu/hour) divided by total system power (Watts) at 82F OAT and indoor conditions of 80F DB and 67F WB (Btuh/Watt).
EER^*	Number	0 to 50	Intertek Summary Tab	Application energy efficiency ratio of total (sensible plus latent) cooling capacity (Btu/hour) divided by total system power (Watts) at steady-state operation and application conditions (Btuh/W).
EER^*_m	Number	0 to 50	Intertek Summary Tab	Measured multiple-fault application sensible energy efficiency ratio (Btuh/W).
EER^*_o	Number	0 to 50	Intertek Summary Tab	Baseline non-fault application sensible energy efficiency ratio of sensible cooling capacity (Btu/hour) divided by total system power (Watts) at steady-state operation and application conditions (Btuh/W).
EER^*_p	Number	0 to 50	Derived from Intertek Summary Tab Data	Predicted application sensible energy efficiency ratio for multiple faults of sensible cooling capacity (Btu/hour) divided by total system power (Watts) at steady-state operation and application conditions (Btuh/W).
EER^*_s	Number	0 to 50	Intertek Summary Tab	Application sensible energy efficiency ratio of sensible cooling capacity (Btu/hour) divided by total system power (Watts) at steady-state operation and application conditions (Btuh/W).
ε_i	Number	0 to 1	Derived from Intertek Summary Tab Data	Single-fault to non-fault sensible efficiency impact ratio (dimensionless).
$\Delta\varepsilon$	Number	0 to 1	Derived from Intertek Summary Tab Data	Measured minus predicted divided by measured application sensible energy efficiency ratio (dimensionless).
ESP	Number	-10 to 10	Intertek Summary Tab	External Static Pressure measurement of total airflow resistance based on outlet minus inlet air pressure (i.e., flow resistance of filters, grills, coils and ductwork) created by equipment blower fan (inches H2O).

Name	Type	Range	Relationship	Definition
ESPI	Character	20	NA	Efficiency Savings and Performance Incentives report.
EST	Number	-50 to 250	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Evaporator Saturation Temperature of refrigerant based on the suction pressure measured at steady-state operation (F).
FDD	Character	20	NA	Fault Detection Diagnostic method, protocol, or procedure used to evaluate HVAC system faults.
Fingers	Character	20	NA	Technicians establish minimum outdoor damper position using 1, 2, or 3 fingers corresponding to the following diameters: 1-finger 0.74 inch (1.88 cm), 2-fingers 1.289 inches (3.27 cm), and 3-fingers 1.972 inches (5.01 cm).
h_r	Number	0 to 100	Intertek G100958166-100-XX Tab	Enthalpy of return air from conditioned space; a thermodynamic quantity equivalent to the total heat content of air consisting of internal energy (u) plus product of pressure (p) times volume (v) (Btu/lbm or J/kg).
h_{mx}	Number	0 to 100	Intertek G100958166-100-XX Tab	Enthalpy of mixed air leaving fan based on supply air humidity ratio and air temperature minus temperature increase due to fan (Btu/lbm or J/kg).
h_{oa}	Number	0 to 100	Intertek G100958166-100-XX Tab	Enthalpy of outdoor air (Btu/lbm or J/kg).
HVAC	Character	20	NA	Heating, Ventilating, and Air Conditioning
IEER	Number	0 to 50	Intertek IEER Calculation file	Integrated Energy Efficiency Ratio is defined in AHRI Standard 340/360-2007 and uses indoor conditions of 80F drybulb and 67F wetbulb and the following outdoor drybulb conditions: 95F (100%), 71F (75%), 68F (50%), and 65F (25%). In January 2010 IEER replaced the Integrated Part Load Value (IPLV) as the part load energy efficiency descriptor for all commercial unitary products rated above 65,000 Btu/h. See http://www1.eere.energy.gov/buildings/appliance_standards/pdfs/ac_hp_rfi_noda.pdf. Also see ASHRAE Standard 90.1-2007, Energy Standard for Buildings Except Low-Rise Residential Buildings, October 2007.
Intertek Summary Tab	Character	20	NA	Intertek test file summary tab of AHRI performance data (40 to 50 records and 6 fields).
Intertek Comments Tab	Character	20	NA	Intertek test file comments (20 records and 6 fields).
Intertek Detail1 Tab	Character	20	NA	Intertek test file detail data (20 records and 10 fields).
Intertek Power Tab	Character	20	NA	Intertek test file power usage data (100 records and 10 fields).
Intertek Compressor 1 Tab	Character	20	NA	Intertek test file compressor 1 refrigerant system data (100 records and 10 fields).

Name	Type	Range	Relationship	Definition
Intertek Compressor 2 Tab	Character	20	NA	Intertek test file compressor 2 refrigerant system data (100 records and 10 fields).
Intertek Indoor TC Tab	Character	20	NA	Intertek test file refrigerant system backup thermocouple temperature measurement data (100 records and 10 fields).
Intertek Outdoor TC Tab	Character	20	NA	Intertek test file outdoor, evaporator-inlet-air, evaporator-outlet-air, mixed-air thermocouple temperature measurement data (100 records and 10 fields).
Intertek G100958166-100-XX Tab	Character	20	NA	Intertek test file of 4-second time-series measurement data in 500 to 1000 rows and 200 records in rows 1 and 2 of columns A through GA. Average steady-state values are summarized in Intertek Summary, Detail1, Power, Compressor 1, Compressor 2, Indoor TC and Outdoor TC tabs. For cyclic tests the following columns are relevant: Column V = Net Sensible Cap, Column AC = Total Power, Column AD = Air Side Cap, Column AH = Air Side EER.
Intertek G100958166-100-XX Tab	Character	20	NA	Intertek test file data record fields (2000 records and 2 fields).
IOU	Character	20	NA	Investor Owned Utility
IPLV	Number	0 to 50	Intertek IPLV Calculation file	Integrated Part Load Value was developed by AHRI to describe performance of a chiller or air conditioner derived from equipment efficiency tests at various capacities. ASHRAE Standard 90.1 specifies minimum IPLV for equipment. IPLV is calculated based on efficiency at capacities of 100%, 75%, 50%, and 25%. IPLV = 0.01A + 0.42B + 0.45C + 0.12D, where A = EER at 100% load, B = EER at 75% load, C = EER at 50% load, and D = EER at 25% load.
ISP	Number	-10 to 10	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Inlet Static Pressure of air entering evaporator (inches of H2O)
IWC	Number	-10 to 10	NA	Inches Water Column (IWC) pressure equal to 0.0734824 inches of mercury (Hg) equal to 0.0360912 psig (inches of H20).
LT	Number	0 to 200	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Liquid Temperature of refrigerant leaving the condenser measured at the service valve at steady-state operation (F).

Name	Type	Range	Relationship	Definition
LP	Number	0 to 600	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Liquid Pressure of refrigerant leaving the condenser measured at the service valve at steady-state operation (psig).
NC	Character	20	NA	Non-condensable air or water vapor enter the refrigerant system through low-side leaks or are not removed by proper evacuation to 500 microns Hg or 0.009668 psig and remain trapped inside the condenser tubing. Non-condensables cover interior tube surface area and reduce heat transfer causing increased discharge pressure and compressor power air. Water vapor mixes with refrigerant oils causing sludge which cause restrictions and acid formation which cause compressor failure.
non-TXV	Character	20	NA	Piston metering device or single/multiple fixed-orifice or piston refrigerant expansion valve(s) installed on liquid line before evaporator.
OA	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Outdoor Air volumetric flow rate of air into the system primary airflow through operable (i.e., economizer) or fixed dampers or cabinet.
OAF	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Outdoor Air Fraction defined as the fraction of outdoor air intake flow or return airstream of the system primary airflow per ASHRAE 62.1.
ΔOAF	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	ΔOAF % equals unsealed minus sealed OAF %.
OAF_e	Number	0 to 1	Derived from Intertek Summary Tab Data	Outdoor Air Fraction of air based on specific enthalpy entering return air stream through economizer, relief damper, or cabinet as a fraction of total airflow (dimensionless).
OAF_m	Number	0 to 1	Derived from Intertek Summary Tab Data	Outdoor Air Fraction based on measured mixed-air drybulb temperature entering unit through economizer as a fraction of total airflow (dimensionless).
OAF_t	Number	0 to 1	Derived from Intertek Summary Tab Data	Outdoor Air Fraction based on drybulb temperature measurements entering unit through economizer, relief damper, or cabinet as a fraction of total airflow (dimensionless).
OAL	Number	-10 to 10000	Derived from Intertek Summary Tab Data	Outdoor Air Leakage of air into the return air stream (scfm)

Name	Type	Range	Relationship	Definition
OAT	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Outdoor Air Temperature (F)
QI	Character	20	NA	Quality Installation includes correct design, installation, final testing, documentation, and education.
QM	Character	20	NA	Quality Maintenance inspection and maintenance requirements to preserve a system's ability to achieve acceptable thermal comfort, energy efficiency, and indoor air quality.
Restriction	Character	20	NA	Restrictions (typically blocking filter drier or expansion device) reduce refrigerant mass flow, efficiency and capacity and are generally caused by moisture, copper particles, flux/brazing residue, sludge, or particulates in refrigerant system when installed, manufactured, or opened for repair.
RTD	Character	20	NA	Resistance Temperature Detector
RTU	Character	20	NA	Roof Top Unit refers to a commercial packaged HVAC system located on the roof of a building.
RTU1	Character	20	NA	7.5-ton TXV unit (1 of 2) equipped with a thermostatic expansion valve refrigerant expansion device.
RTU2	Character	20	NA	7.5-ton TXV unit (2 of 2) equipped with a thermostatic expansion valve refrigerant expansion device.
RTU3	Character	20	NA	7.5-ton non-TXV unit equipped with a fixed-orifice refrigerant expansion device.
RTU4	Character	20	NA	3-ton TXV unit equipped with a thermostatic expansion valve refrigerant expansion device.
RTU5	Character	20	NA	3-ton non-TXV unit equipped with a fixed-orifice refrigerant expansion device.
SC	Number	0 to 100	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Subcooling temperature measuring heat removed from refrigerant after it has changed to a liquid below its saturation temperature and equal to the condenser saturation minus liquid line temperature (F).
scfm	Number	0 to 50000	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Standard cubic feet per minute is the volumetric flow rate of air corrected to "standardized" conditions of 70F [21C] drybulb temperature, 29.92 in. Hg (inches of Mercury) [101.3 kPa] atmospheric pressure, and density of 0.075 lb/ft3 [1.2 kg/m3] (ft^3/min).
SEER	Number	0 to 50	Intertek Summary Tab	Seasonal Energy Efficiency Ratio (SEER). The total heat removed from the conditioned space during the annual cooling season, expressed in Btu's, divided by the total electrical energy consumed by the air conditioner or heat pump during the same season, expressed in watt-hours.

Name	Type	Range	Relationship	Definition
SH	Number	-10 to 100	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Superheat temperature measuring heat added to refrigerant after it has changed to a gas above its saturation temperature and equal to suction line temperature minus evaporator saturation temperature (F).
SP	Number	-10 to 200	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Suction Pressure of refrigerant entering compressor measured at steady-state operation (psig).
ST	Number	-10 to 200	Intertek Compressor 1 or 2 Tab or Intertek G100958166-100-XX Tab	Suction Temperature of refrigerant entering compressor measured at steady-state operation (F).
Ton (cooling)	Number	0 to 1000	Derived from Intertek Summary Tab Data or Intertek G100958166-100-XX Tab	One ton of cooling is defined as the heat energy removed from one short ton of water (2,000 pounds) to produce one ton of ice at 32F (0°C) in 24 hours. The energy required for the phase change of liquid water at 32F (0C) into solid ice at 32F is referred to as the heat of fusion which is 144 Btu/lb multiplied by 2,000 lbs of water or 288,000 Btu of energy over a 24 hour period requires 12,000 Btu/hour to make one ton of ice in one day. The Btu is the energy required to raise one pound (lb) of water one degree Fahrenheit (F).
TS	Number	-10 to 100	Derived from Intertek Summary Tab Data or Intertek G100958166-100-XX Tab	Temperature Split difference of evaporator inlet air drybulb minus outlet air drybulb temperature (F).
TXV	Character	20	NA	Thermostatic Expansion Valve controls the amount of refrigerant flow into the evaporator thereby controlling superheating at the outlet of the evaporator based on a sensing bulb installed on the suction line of the refrigeration system.
$\dot{W}_{fan}$	Number	0 to 50000	Intertek Power Tab	Watts of electric power used by the indoor fan
WB	Number	-50 to 150	Intertek Summary Tab or Intertek G100958166-100-XX Tab	Wet Bulb temperature of a volume of air if cooled to saturation (100% relative humidity) by evaporation of water vapor into the air with latent heat supplied by the volume of air. Wet bulb is the lowest air temperature under ambient conditions by evaporation of water only. At 100% relative humidity, wet-bulb temperature equals dry-bulb temperature.

Name	Type	Range	Relationship	Definition
WHPA	Character	20	NA	Western HVAC Performance Alliance includes engineers, contractors, energy efficiency, facility, and property management organizations, researchers, educators, utilities, and regulatory agencies whose decision-maker-level appointees work with one another to improve energy efficiency.
x_a	Number	0 to 1	Derived from Intertek Summary Tab Data	Evaporator airflow ratio decrease due to evaporator coil blockage (dimensionless).
x_p	Number	0 to 1	Derived from Compressor 1 or 2 Tab Data	Discharge pressure ratio increase due to condenser coil blockage (dimensionless).
x_r	Number	0 to 1	NA	Refrigerant charge per factory charge ratio (dimensionless).
y_e	Number	0 to 1	Derived from Intertek Summary Tab Data	Sensible efficiency impact of evaporator coil blockage based on evaporator airflow ratio decrease (dimensionless).
y_c	Number	0 to 1	Derived from Intertek Summary Tab Data	Sensible efficiency impact of condenser coil blockage based on discharge pressure ratio increase (dimensionless).
y_o	Number	0 to 1	Derived from Intertek Summary Tab Data	Sensible efficiency at pre-existing refrigerant charge per factory charge ratio (dimensionless).
y_r	Number	0 to 1	Derived from Intertek Summary Tab Data	Sensible efficiency impact at refrigerant charge per factory charge ratio (dimensionless).
y_{rca}	Number	0 to 1	Derived from Intertek Summary Tab Data	Sensible efficiency impact of refrigerant charge measure based on difference of y_r minus y_o (dimensionless).