MARKET ASSESSMENT AND FIELD M&V STUDY FOR COMPREHENSIVE PACKAGED A/C SYSTEMS PROGRAM

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EXECUTIVE SUMMARY

This report presents and discusses the results of a market assessment and field measurement and verification study that was conducted for the Comprehensive Packaged Air Conditioning (A/C) Systems Program that Southern California Edison Company has been implementing.

The objectives for this study were as follows:

- Perform pre- and post-installation measurements that could be used to assess A/C performance.
- Assess baseline practices and determine baseline performance for A/C installations using an appropriate control group.
- Provide early verification of energy and demand savings estimates for the measures promoted in the CPACS program.
- Ensure the results can be extrapolated to the population for CPACS 2009-2011 program planning purposes.

A major aspect of the project was to conduct field measurements of the performance of residential air conditioning units in order to quantify and assess the effects of servicing. This involved making detailed baseline measurements pertaining to the performance of a sample of HVAC packaged units, performing servicing on a subset of this sample, and then making a new set of performance measurements on the serviced units.

Data with which to assess the effects of CPACS maintenance and tune-up services were developed through field measurements for a sample of residential air conditioning units. The sample of units for the measurement work was selected from among 148 households (with a total of 168 air conditioning units) who had participated in a demand response program that SCE had implemented. The result of recruiting from this pool was that 106 households with 120 air conditioning units agreed to participate in the testing project. Performance testing was applied to these 120 HVAC units in their as-found condition. Units selected for the baseline measurement effort represented different sizes and ages.

Diagnoses of baseline faults were made for 109 HVAC units. Of these 109 units, 89 (82%) had one or more faults. The primary fault conditions were associated with refrigerant charge level and air flow level. Multiple faults for a unit are not conclusive since some faults, specifically air flow, can generate false fault diagnosis. Pre and post measurements were intended to be blind to the servicing contractor.

There were 43 units that received refrigerant charge servicing from an HVAC contractor for which ADM field staff took pre-servicing and post-servicing measurements. Conditions for the pre and post measurements were different and often made months apart. These pre- and post-servicing measurements were used to analyze changes in the EERs for the units at standard conditions before and after the servicing. The average EER for the units increased from 6.64 before servicing to 7.05 after servicing, an increase of about 6.1%. The results of a paired t-test showed that the hypothesis of no difference between the pre- and post-servicing averages could be rejected with a confidence level of 80%.

Metered data that SCE had previously collected showed that the average annual kWh usage for HVAC units was 1,303 kWh. With savings from refrigerant charge tune-ups estimated to be 6.1%, the annual kWh savings from the refrigerant charge tune-up is estimated to be 79.5 kWh.

As another part of the research on residential central air conditioning systems that was performed during this project, measurements of total duct leakage and of duct leakage to unconditioned space were made for a sample of houses. Conventional practice in measuring duct leakage is to use a duct pressurization test, usually with a standard reference positive air pressure of 25 Pascals. However, some studies have suggested that duct leakage measured at 25 Pascals may be overstating actual leakage. To examine this question, measurements of duct leakage were made at a sample of houses using three methods of measurement. Two of the methods for making the duct leakage measurements were variants of the usual duct pressurization method, with one set duct pressurization measurements made using the standard fixed 25 pascals (Pa) pressurization and a second set made by taking measurements at ½ system static pressure (SSP) for central air conditioning systems. Tracer gas infiltration testing, which is regarded as one of the more accurate methods for measuring infiltration rates, was used as a third method of measurement to provide benchmark values for duct leakage against which measurement results from the duct pressurization methods could be compared and assessed. Carbon dioxide (CO_2) was used as the tracer gas for this testing. The results of this testing showed that the correlation between the total duct leakage CFM measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.313; for duct pressurization at ½ SSP the correlation was 0.397. The correlation between the CFM of duct leakage to unconditioned space as measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.478; for duct pressurization at ½ SSP the correlation was 0.744. Measurements of duct leakage to unconditioned space made through the duct pressurization method at ½ SSP were more highly correlated with the tracer gas measurements than were measurements made at 25 Pa. These results suggest that the duct pressurization method at ½ SSP provides more accurate measurement of duct leakage to unconditioned space when using conventional measuring equipment.

Baseline measurements of total duct leakage and of duct leakage to unconditioned space were made for the sample of 109 sites for which air conditioning measurements were made. These baseline measurements were made with two duct pressurization methods (i.e., at 25 Pascals and at ½ System Static Pressure). Measurements of total duct leakage and of duct leakage after a servicing call from an HVAC contractor were made for a sample of units, also using both duct pressurization methods. For both methods of measurement, the average total duct leakage for the units decreased about 12 percent from before servicing to after servicing. The results of a paired 2-tail t-test showed that the hypothesis of no difference between the before- and after-servicing averages could be rejected with a confidence level of 80%. The improved airflow that resulted from reducing duct leakage implies an annual kWh savings of 82.1 kWh per HVAC unit.

1. INTRODUCTION

Under contract with Southern California Edison Company (SCE), ADM Associates, Inc. (ADM) has conducted a market assessment and field measurement and verification study for the Comprehensive Packaged Air Conditioning (A/C) Systems Program.

The objectives for this study were as follows:

- Perform pre- and post-installation measurements that could be used to assess A/C performance.
- Assess baseline practices and determine baseline performance for A/C installations using an appropriate control group.
- Provide early verification of energy and demand savings estimates for the measures promoted in the CPACS program.
- Ensure the results can be extrapolated to the population for CPACS 2009-2011 program planning purposes.

The scope of work for achieving these objectives was comprised of seven (7) tasks.

- Task 1 was to participate in a project initiation meeting.
- Task 2 was to develop a detailed work plan.
- Task 3 was to implement the data collection plan.
- Task 4 was to process raw data and conduct analyses for baseline and savings parameter estimates.
- Task 5 was to prepare memoranda and reports on baseline data and analyses results.
- Task 6 was to provide project management.
- Task 7 was to prepare and deliver an electronic database of measurements.

The purpose of this final report is to report and document the results of the work performed during this project. This report is organized as follows.

- Chapter 2 describes the field measurement procedures that were used for the project.
- Chapter 3 presents the results of the field measurement effort. Three major sets of measurements are tabulated and presented.
  - Baseline measurements;
  - Results of the fault diagnosis detection performed during the baseline measurements; and
  - Post-servicing field measurements.
- Chapter 4 presents and discusses an analysis of the field measurements. Final results are presented. Measurement results before and after servicing are presented for each serviced

unit along with the service measure. This chapter presents data on changes in EER resulting from proper maintenance and servicing of packaged air conditioning units.

- Chapter 5 reports on the conclusions from the work. This chapter also summarizes results, describes the lessons learned, and identifies areas for future work.

2. FIELD MEASUREMENT PROCEDUSRES

A major aspect of this project was to conduct field measurements of the performance of residential air conditioning units in order to quantify and assess the effects of servicing. This involved making detailed baseline measurements pertaining to the performance of a sample of HVAC packaged units, performing servicing on a subset of this sample, and then making a new set of performance measurements on the serviced units. This chapter describes the set of procedures that ADM staff used for making performance measurements on the units.

2.1 MEASUREMENT POINTS

As the first step in developing the field measurement procedures, ADM prepared a list of the points for which measurements were to be taken on each unit. Table 2-1 presents this list of points.

Table 2-1. Measurement Points

Point	*Abbreviation*	*Description*
1	kW_{Total}	Total electric Power to the unit, kW
2	RA_{cfm}	Return airflow rate, measured with duct blaster
3	SAT_{db}	Supply Air Temperature, dry-bulb, °F
4	SA_{rh}	Supply Air relative humidity, %
5	SA_{sp}	Supply Air duct static pressure, in Pa
6	$RMAT_{db}$	Return Air Temperature, dry-bulb, °F
7	RMA_{rh}	Return Air relative humidity, %
8	OAT	Outside Air Temperature (ambient dry-bulb), °F
9	OA_{rh}	Outside Air relative humidity (ambient), %
10	CA	Condenser Air or Air Off the Condenser Temperature, °F
11	SP	Refrigerant Suction line pressure, psig
12	LP / DP	Refrigerant Liquid Line or Discharge line pressure, psig
13	ST	Refrigerant Suction line Temperature, °F
14	LT	Refrigerant Liquid line Temperature, °F
15	Duct Leakage	Duck Leakage, in cfm
16	Outside Duct Leakage	Duct leakage to unconditioned space, in cfm

Figure 2-1 is a schematic diagram of the locations of points associated with each unit where field measurements were taken.

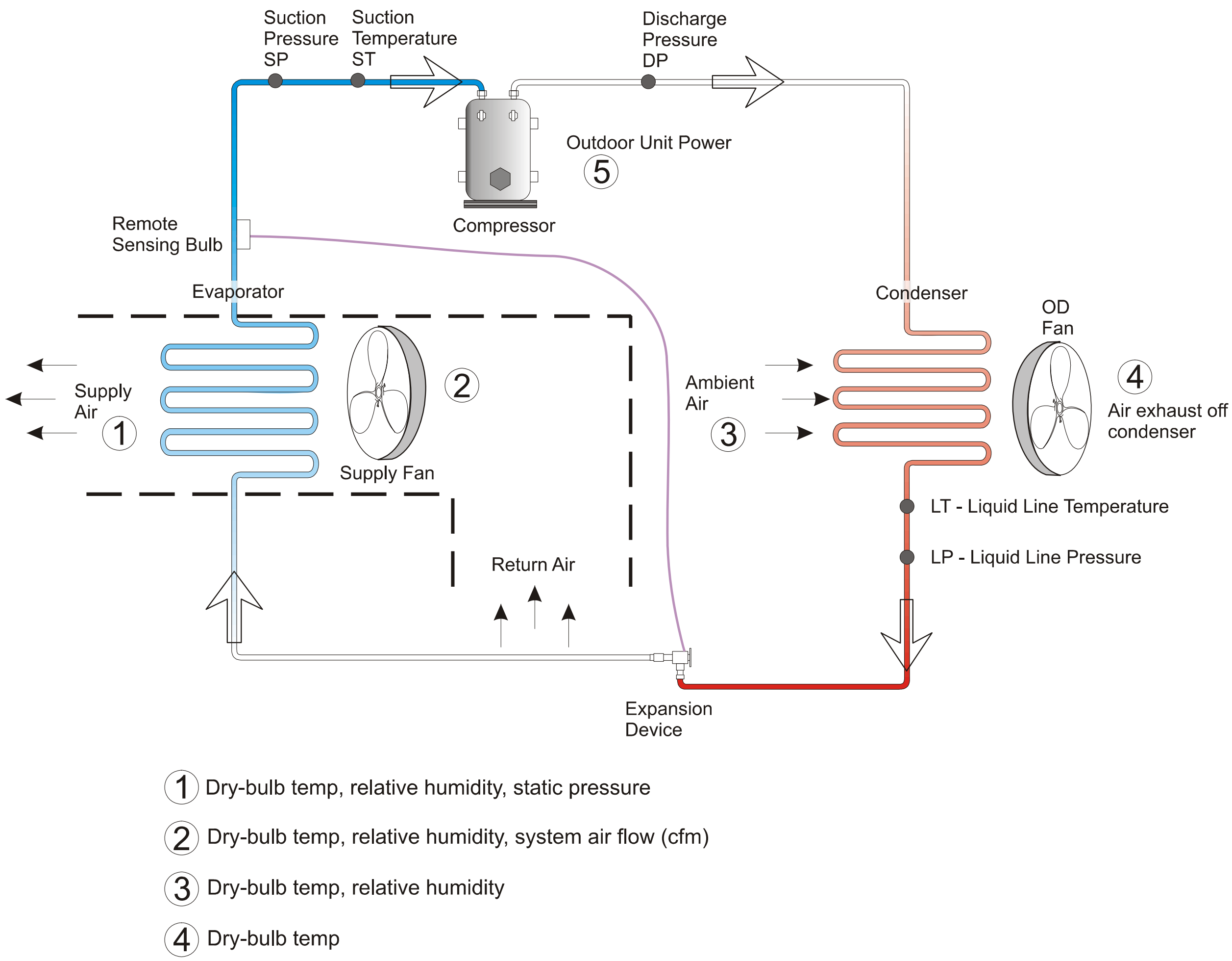

Figure 2-1. Air Conditioning Air and Refrigerant Side Measurements

2.1.1 CONDITIONS FOR MAKING PERFORMANCE MEASUREMENTS

To ensure comparability of performance measurements across units, conditions under which the performance measurements were taken were specified.

- The unit had to be in full load operation and at steady state during the performance tests. Ambient temperature had to be more than 65°F, and the unit had to have been running for 10-15 minutes or longer to ensure steady state conditions.
- If the unit was a heat pump, it had to be ensured that it was in a cooling cycle.
- All access panels had to be in place.
- The test technicians were instructed and trained to prepare and place sensors and meters such that all measurements (except airflow) could be read and recorded as quickly as possible. The meter values were recorded on a standardized data collection form. The airflow measurements had to take place immediately following all other data collection.

2.2 AIR-SIDE PERFORMANCE MEASUREMENTS

ADM staff measured the air-side performance after the unit had reached a steady state condition.

The validity of the diagnostics is based on the presumption that there is proper air flow across the coil. In practice all field diagnostics assumes there is proper air flow. An air flow of 320 cfm per ton or less is considered a low air flow rate. Air filters were removed for the air flow test. Some of the flow could be attributed to duct leakage. The air flow measurements were made at the return air register, but leakage on the return side of the duct system would provide lower air flow rates than actually occur across the evaporator coil. There is not a reliable efficient method to measure the air flow across the evaporator coil.

We measured dry-bulb temperature and relative humidity of the following air-streams using RH/Temp monitors:

- Supply Air (SA)
- Return Air (MA)

The measurements of Supply and Return Air were based on data from three measurement points in each air stream. One of the three points was located at the center of the air stream, and the other two points were located 1½ inches from the opposite edges of the duct (see *Figure 2-2*). The locations of these three points were chosen based on the analysis of a preliminary set of data collected on the units. This analysis showed that the average temperature measured at these three points could be correlated to the average of the full grid of measurement points, to a standard deviation of 0.1ºF across the ducts of all investigated units.

One remote sensor was used for each measurement location. This allowed the sensors to stabilize in their respective air streams without concerns about sensor response time when using a single probe that moved from point to point. Thus, by reducing the number of points and placing sensors for each of these points, the interval between data collection periods could be reduced to less than 10 minutes.

Dry-bulb air temperatures and relative humidity were measured using a Sper Scientific remote RH/Temp monitor model 800027. The remote sensor was placed on a seven-foot long cable. A set of fabricated rods was used to position the sensors in the appropriate locations in the air ducts. (This monitor was selected because no temperature and humidity probes on long or telescoping arms with response times of less than 10 seconds are available.)

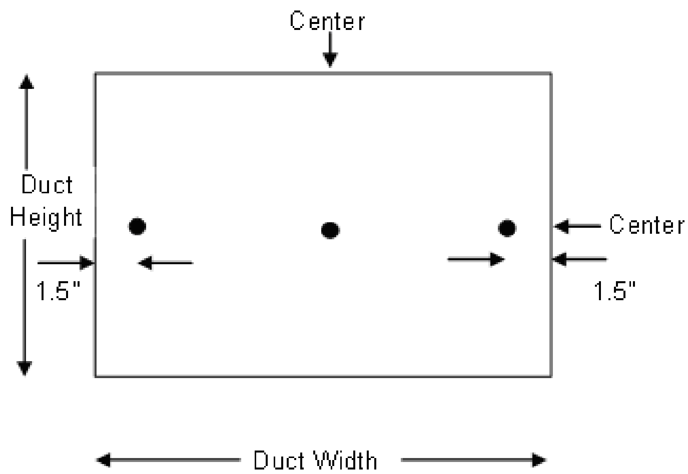

Figure 2-2. Location in Duct of Three Point Temperature and Humidity Measurements.

Using a relative humidity meter with a digital probe provided measurements with information on wet-bulb temperatures equivalent to those that would be obtained with a digital meter that provides wet-bulb temperature directly; both types of meters use the same sensor technology. The measurement procedures required using three temperature and humidity sensors that were located at different points in the air stream cross-section to provide a better overall average measurement of the airstream conditions.

Eight of the remote RH/Temp monitors were used for measurements of each unit. Three of the monitors were placed in the supply air duct, and three were placed in the mixed air chamber. One monitor was used to record ambient air temperature. Note that ambient temperature and relative humidity were recorded at the beginning of all other temperature measurements. A RH/Temp monitor was also used to measure air off the condenser. Ambient air temperature and humidity were also recorded at the end of the other temperature measurements. We measured the temperatures of ambient air and air off the condenser with radiation shielded air probes.

The temperature and humidity were measured in the supply plenum because this provided a chamber for the air to mix before being distributed to the various ducts that branch off. However there were some units, in particular closet air handlers, where the measurement was made immediately after the evaporator coil because of physical limitations.

2.3 POWER MEASUREMENTS

Electric power measurements were taken across the unit's main supply disconnect. An AEMC power meter was used to measure the electric load input to the unit. The electrical power measurements were made for the outdoor unit and for the air handler. Total power was then calculated as the sum of these two measurements. Power measurements were taken after the unit had been operating for 10 to 15 minutes and was in a steady state running condition.

2.4 REFRIGERANT SIDE PERFORMANCE MEASUREMENTS

Refrigerant pressures were measured using a set of refrigerant gauges on a manifold. Initial pressure measurements were made after the compressor had run 3 to 5 minutes. Another set of measurements was made during data collection of the air side measurements. These pressure measurements were made after the compressor had been running for at least 10 to 15 minutes in order for the system to stabilize.

An Extech thermometer with a remote pipe clamp-on thermocouple probe was used to measure the refrigerant suction line (ST) and liquid line (LT) temperatures. An Extech dual input digital thermometer was used to make simultaneous measurements.

The condenser air (CA) temperature was measured as the dry bulb air temperature exhausted over the condenser. The condenser over air temperature (COA) was calculated by subtracting the measured ambient temperature (OAT) from the representative condenser temperature. The condenser temperature (CT) was recorded as the saturation temperature at the LP pressure or approximated as the saturation temperature at DP-15 psig pressure.

2.5 DUCT LEAKAGE MEASUREMENTS

To measure the effects of duct repair and sealing, data are needed on air leakage from the ducts to unconditioned space. Figure 2-3 is a diagram of the general approach that was used to measure duct leakage.

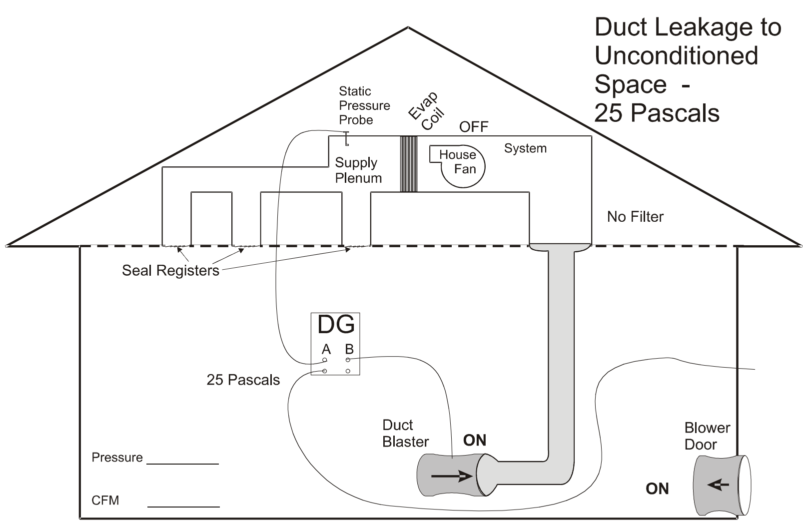

Figure 2-3. Diagram of Approach to Analyzing Duct Leakage to Unconditioned Space.

A duct blaster was used in conjunction with a digital manometer and a blower door to collect data on duct air leakage. (A duct blaster is a variable speed fan calibrated to measure air flow (CFM) for a given pressure drop across a flow sensor.)

- With the system fan turned on, the pressure difference between the supply plenum and conditioned space is measured. This is the target pressure to be maintained during the fan flow tests. If there is no access to the supply plenum, the pressure probe is placed in the nearest supply duct. The probe is ajusted to achieve the highest pressure and then firmly attach the probe (e.g., with duct tape) to ensure that it does not move during the fan flow test. The system fan is turned off.
- The duct blaster is connected and firmly sealed to the return grill to blow air into the duct system. All supply registers are sealed with register sealing tape. A plastic tube connected to the digital manometer is inserted into the duct system at the supply plenum to measure duct system pressure. Another tube connected to the digital manometer is also attached to the duct blaster at the blower fan and measures fan pressure.
- A blower door fan is placed in an outside doorway so the house can be pressurized.
- For the actual testing, the duct blaster fan is turned on, and the duct system is pressurized until stable at the supply plenum pressure determined in the first step. The digital manometer converts fan pressure to flow rate (measured in CFM), which we can cross reference with fan pressure/flow table for accuracy.
- The blower door is turned on and the speed adjusted until the pressure difference between the supply duct and the house is zero. At this pressure there is no duct leakage to the conditioned space.
- The duct blaster flow rate (in CFM) is measured and recorded. This is the duct leakage rate to unconditioned space when the system fan is on.
- The duct blaster is connected and firmly sealed to the return grill to blow air into the duct system. All supply registers are sealed with register sealing tape. A plastic tube connected to the digital manometer is inserted into the duct system at the supply plenum to measure duct system pressure. Another tube connected to the digital manometer is also attached to the duct blaster at the blower fan and measures fan pressure.

2.6 FUNCTIONAL VERIFICATION FROM AIR-SIDE PERFORMANCE TESTING

After all field measurements had been completed, the staff entered the data into a spreadsheet on a laptop computer for analysis. The software performed tests on the data and produced verification of the measured data and functional test results. The equations used to verify the data and performance calculations are described in the following discussion.

2.6.2 Input Checks

The spreadsheet first checked the measurement input data for data entry errors. With data entered properly, the spreadsheet then applied checks to determine that the following conditions were absolutely met.

$$OAT \geq 65°F$$

CA > OAT (°F)

OAT < LT (°F)

CT > CA (°F)

LP > SP (psig), if DP was measured, instead of LP use DP – 15

ET < ST < RA (°F)

ET < SA < RA (°F)

SArh > RArh (% relative humidity)

Any condition that was not met and that was not the result of an error associated with translating from measurement notes to spreadsheet was further investigated. For any of the conditions that were not met, the staff repeated the test for the specific measurement(s) associated with the failed verification. If the measurement gave the same reading, the calibration of the instrument was field checked. If the field check showed that the instrument was within the field calibration range, the original measurement was allowed to stand. If the measurement gave a significantly different result, the entire test on the unit was repeated.

The following additional, conditional checks were also made.

45 °F < SAT_{db} < 65 °F

65 °F < RAT_{db} < 85 °F

RA - SAT_{db} > 8 °F

LT < CT

CA < CT

0.7 < kW/ton < 1.6

320 < CFM/ton < 480

2.6.3 Air Side Calculations

The staff calculated the performance of the unit using the field measurements and the following formula:

Airflow Rate (CFM) = System airflow measured with Duct Blaster

Measured cooling capacity in tons = Total flow (CFM) * 60 (min/hr) * Supply air density (lbm/cuft) * (Enthalpy of mixed air (Btu/lb) – Enthalpy of supply air (Btu/lb)) / 12000 (Btu/ton)

EER = (12 * Measured cooling capacity in tons) / Total measured kW input (BTU / hour / Watts)

COP = EER / 3.413

These calculations were made using the following steps.

- The CFM of average measured airflow rate was used.
- Manufacturer's nameplate information was used to record the nominal capacity in tons.
- Psychrometric data were used to calculate the enthalpy difference in Btu/lb. of air from supply and return air streams.
- Total pounds of air mass flow rate delivered to the space were calculated from the CFM data and the psychometric chart.
- Multiplying this by the difference in enthalpy per pound of air determined the total cooling capacity of the unit.
- The calculated total cooling capacities were compared with the capacity data of the unit. Measured capacity for actual conditions should be higher than 40%, but less than 120%.
- The Energy Efficiency Ratio (EER) was calculated by dividing the unit capacity by the total unit power input per the following formula:

  EER = (Total Cooling Capacity in Btuh/Total unit power input) x 3.413

  where the total unit power input is the measured watts summed for the outdoor unit and the air handler.

2.6.4 Correction of COP to Standard Conditions

A coefficient of performance (COP) that reflected standard conditions was also calculated for each unit. Standard conditions are 95 ºF outside air temperature and 67 ºF return air wet-bulb temperature.

The coefficient of performance for standard conditions was calculated as an adjustment to the COP calculated for measurement conditions, as presented in Section 2.6.2. Staff used the following procedure to adjust the COP to standard conditions.

Manufacturers have published packaged air conditioning units' performance under varying outdoor dry-bulb temperatures (OAT) and return air wet-bulb temperature (RATwb). For example, a copy of performance data for a Carrier series 48HJ with Scroll compressor is given in Figure 2-4.[1] Some manufacturers provide the power input to the unit as a total of kW, while

[1] See Carrier, catalog #524-80032, pg 161, September 2004, available at http://www.xpedio.carrier.com/idc/groups/public/documents/techlit/48h,t-5pd.pdf.
For this model, the manufacturer used 350 cfm/ton as their standard rating condition (i.e., 5 ton unit at 1,750 cfm, 95 °F $OAT_{db,}$ RAT_{wb}, 67 °F conditions).

others provide the power input to the compressor only. In the latter case the fan power consumption at the given flow rate and a standard pressure head can be calculated and added to this to arrive at the total power input.

The ratio of Total Cooling capacity referenced in Figure 2-4 multiplied by a constant factor of 12 kBTU/ton to total power input represents the Energy Efficiency Ratio (EER) of the unit under the given OAT and RATwb:

EER = (12 * Measured cooling capacity in tons) / total measured kW input

When this EER is normalized through division by EER at standard conditions (95 ºF OAT and 67 ºF RATwb), an EER normalization factor value is calculated:

Normalization Factor = $EER_{Normalized} / EER_{Standard\ Conditions}$

This EER normalization factor is compared across units of different manufacturers within each compressor type and one representative profile is generated from a majority of the units profiled for this program during the 2005-2006 period. This factor can be used to convert the performance of the unit from a standard condition to a field condition or vice versa.

48HJ006 (5 TONS)

Temp (F) Air Ent Condenser (Edb)		Air Entering Evaporator — Cfm/BF: 1500/0.08			1750/0.09			2000/0.11			2500/0.13		
		Air Entering Evaporator — Ewb (F): 72	67	62	72	67	62	72	67	62	72	67	62
75	TC	70.8	65.4	58.5	72.5	67.3	61.1	73.0	68.4	62.8	74.8	70.3	64.8
	SHC	34.1	42.7	49.9	35.7	45.5	54.2	36.8	48.0	57.8	39.6	53.0	63.4
	kW	3.53	3.49	3.44	3.55	3.50	3.46	3.55	3.51	3.47	3.57	3.54	3.48
85	TC	68.9	63.2	55.3	70.5	65.1	57.9	72.2	66.4	60.2	73.2	68.1	62.9
	SHC	33.5	41.8	48.4	35.0	44.8	52.8	37.0	47.6	56.8	39.3	52.5	62.4
	kW	3.98	3.94	3.87	4.00	3.96	3.90	4.03	3.97	3.92	4.04	3.99	3.94
95	TC	66.8	60.6	52.4	68.3	62.5	54.3	69.3	63.8	56.6	71.2	65.6	60.6
	SHC	32.8	40.7	47.0	34.5	43.8	51.1	36.0	46.7	55.0	39.1	51.8	60.5
	kW	4.48	4.43	4.35	4.50	4.45	4.37	4.51	4.46	4.40	4.55	4.48	4.44
105	TC	64.3	57.7	49.9	65.9	59.8	51.7	66.9	61.1	54.1	68.4	62.8	58.4
	SHC	32.0	39.6	45.8	33.7	42.8	49.7	35.3	45.7	53.5	38.4	51.0	58.4
	kW	5.03	4.96	4.87	5.05	4.99	4.90	5.06	5.00	4.93	5.08	5.02	4.98
115	TC	61.5	54.8	47.3	62.8	56.7	49.1	64.0	58.2	51.6	65.4	59.9	56.1
	SHC	31.0	38.4	44.5	32.5	41.6	48.2	34.4	44.6	51.6	37.4	50.0	56.1
	kW	5.61	5.55	5.46	5.62	5.58	5.49	5.65	5.60	5.52	5.67	5.61	5.57
125	TC	58.7	51.6	44.5	59.9	53.4	46.2	60.8	54.9	49.0	62.2	56.8	53.5
	SHC	30.0	37.2	43.1	31.7	40.4	46.2	33.3	43.4	48.9	36.4	48.9	53.4
	kW	6.27	6.19	6.09	6.28	6.21	6.13	6.29	6.24	6.17	6.31	6.27	6.22

BF = Bypass factor
Ewb = Entering wet bulb temperature (°F)
TC = Total cooling (tons)
SHC = Sensible heat cooling
KW = Total power input (kilowatts)

Figure 2-4. Sample Performance Data for a Unit with Scroll Compressor

Since performance data for some manufacturers is available only for a narrow temperature range, the data were plotted in a two-dimensional plot and extrapolated using a second order algorithm to create a EER normalization factor for wider field operating conditions of 75 ºF to 125 ºF OAT and 57 ºF to 72 ºF RATwb. The CPACS program allows contractors to make measurements for refrigerant charge when the outdoor temperature is down to 55 ºF or 60 ºF dependent on the refrigerant metering device. For evaluation we allowed a minimum of only 65 ºF. The EER normalization factor was extrapolated down to 65 ºF.

Figure 2-5 shows the available data unshaded, with the extrapolated data are shown as shaded. The EER normalization factors were calculated for all three conditions defined by the manufacturer. The data for standard conditions are used for units with indoor cfm within normal conditions. The data for low-flow and high-flow conditions were used for units identified with lower and higher than standard indoor air flow conditions, respectively. Figure 2-6 presents the EER normalization factor profile for usable field conditions. Since the EER normalization factor is based on standardized condition as the denominator, at 95 ºF OAT and 67 ºF RATwb, the EER normalization factor takes a value of 1 at these conditions, as shown in Figure 2-6. We used the data from Figure 2-5 for all units, since equivalent performance data was not available for most units. This introduces an error in the EER at standard condition comparisons that are dependent on how performance varies for different makes and models and how different conditions were from standard conditions.

EER Normalization Factor of Packaged RTU

scroll	CFM	1500				1750				2000			
	RATwb	72	67	62	57	72	67	62	57	72	67	62	57
	OATdb												
	75	1.42	1.33	1.20	1.05	1.40	1.31	1.20	1.09	1.37	1.29	1.20	1.10
	85	1.25	1.16	1.03	0.88	1.23	1.15	1.03	0.92	1.23	1.14	1.05	0.95
	95	1.09	1.00	0.88	0.75	1.08	1.00	0.88	0.76	1.08	1.00	0.90	0.79
	105	0.95	0.86	0.76	0.64	0.95	0.87	0.76	0.66	0.95	0.87	0.78	0.69
	115	0.82	0.74	0.65	0.55	0.82	0.75	0.66	0.57	0.83	0.76	0.68	0.60
	125	0.71	0.63	0.55	0.47	0.71	0.64	0.56	0.48	0.72	0.65	0.59	0.52

recip	CFM	1750				2250				2400			
	RATwb	72	67	62	57	72	67	62	57	72	67	62	57
	OATdb												
	75	1.32	1.26	1.18	1.16	1.30	1.25	1.17	1.17	1.30	1.24	1.14	1.18
	85	1.18	1.12	1.06	1.05	1.18	1.12	1.07	1.07	1.17	1.12	1.05	1.07
	95	1.06	1.00	0.95	0.94	1.06	1.00	0.96	0.96	1.05	1.00	0.96	0.96
	105	0.94	0.89	0.85	0.85	0.94	0.89	0.86	0.86	0.94	0.89	0.86	0.87
	115	0.84	0.79	0.75	0.75	0.84	0.79	0.77	0.77	0.84	0.79	0.78	0.77
	125	0.75	0.71	0.67	0.67	0.74	0.70	0.68	0.68	0.73	0.69	0.69	0.68

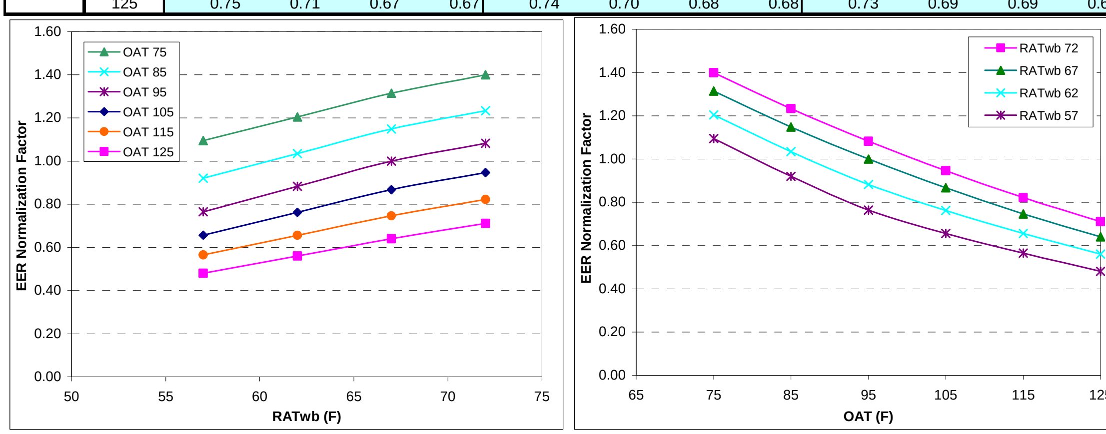

Figure 2-5. Normalization Performance Data for Usable Range of Field Conditions.

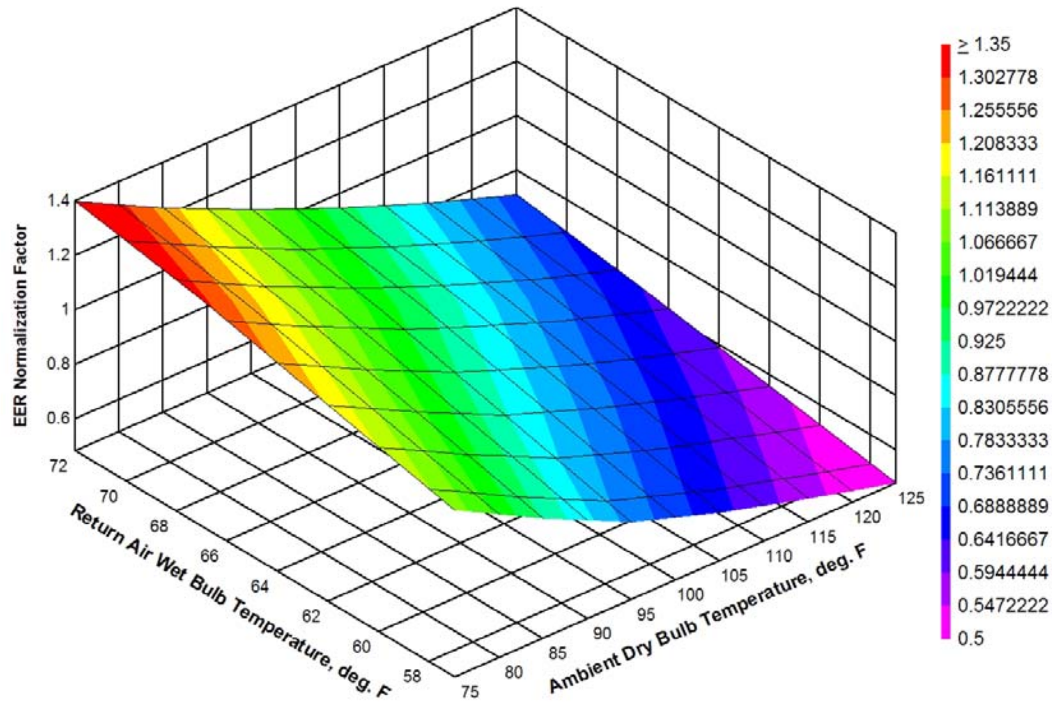

Figure 2-6. Normalization Factor Profile for the Units Equipped with Scroll Compressors.

2.6.5 Calculation of Air Properties

Air-properties were calculated that could be used in measuring the air-side performance of the test units using equations from the ASHRAE Fundamentals Handbook, with the exception of the wet-bulb calculations.

Measured dry-bulb temperature (Tdb), relative humidity (RH), altitude corrected pressure (P) were used in determining the remaining air properties.

Site Pressure, lbs/in^2, P = 14.696 * (1- 0.0000068754 * Altitude)$^{5.2559}$

Saturation Pressure, Pws = EXP{-10440.397/(Tdb+459.67) - 11.29465 - 0.027022355 * (459.67+Tdb) + 0.00001289036 * (459.67+Tdb)^2 - 0.0000000024780681 * (459.67+Tdb)^3 + 6.5459673 * LN(459.67+Tdb)}

Saturation Humidity Ratio, Ws =0.62198 * 1.0039 * Pws / (P – 1.0039 * Pws)

Degree of Saturation, DOS = RH / (1 + (1-RH) * Ws / 0.62198)

Water Vapor Pressure, Pw=Pws * RH

Humidity Ratio, W= 0.62198 * Pw / (P - Pw)

Density of Moist Air, density = 1/((0.7543*(459.67+Tdb)*(1+1.6078*W))/(P*29.921/14.696))

Enthalpy of Moist Air, enthalpy = 0.24 * Tdb + W * (1061+0.444*Tdb)

Properties of the mixed air stream were derived using the mass balance of dry air and moisture separately.

Wet-bulb temperatures were calculated from measured dry-bulb temperature and measured percent relative humidity. The site pressure adjusted for altitude is an input into the equations. The following equations for calculating wet-bulb temperature are from Jensen et al. (1990) ASCE Manual No. 70 (pages 176 & 177).

1) Compute e as (rH / 100) * 0.611 * EXP(17.27 * T / (T + 237.3))
   where T is dry bulb temp in C and
   e is ambient vapor pressure in kPa

2) Compute dewpoint temperature (Td)
   Td = [116.9 + 237.3 * ln(e)] / [16.78 - ln(e)] in C

3) Compute wet bulb temperature (Twb)
   Twb = [(GAMMA * T) + (DELTA * Td)] / (GAMMA + DELTA)
   GAMMA = 0.00066 * P where P is ambient barometric pressure in kPa
   DELTA = 4098 * e / $(Td + 237.3)^2$

2.7 DIAGNOSTICS FOR FAULT DETECTION

Several types of diagnostic testing were used to detect faults.

2.7.1 Level 1 Diagnostics

Using measurements made according to the procedures discussed above, ADM staff made the first level of diagnostic conclusions about the performance of a unit, as follows.

- If the airflow rates are outside the nominal 320-480 CFM per ton range, it is not advisable to make any conclusions about the performance of the unit.
- If the measured airflow rates are within the nominal 320-480 CFM per ton (or whatever range is specified as nominal by the manufacturer), a next level of diagnostics may be applied. Airflow rates must be established based on airflow measurements and be within the nominal operation, as a prerequisite for refrigerant side diagnostics.

2.7.2 Level 2 Detailed Diagnostics for Fault Detection

Several different parameters were calculated for use in qualitatively diagnosing the current operating condition of a unit. These parameters are defined in

Table 2-2.

Table 2-2. Definitions of Refrigerant Variables Used for Fault Diagnostics

Abbreviation	Definition
ET	Evaporator temperature, defined as T_{sat} at low side pressure
SC	Sub-cooling = Condenser Saturation Temperature – Liquid Line Temperature
SH	Superheat
ETD	Temperature drop across evaporator
CTD	Air Temperature Increase over Condenser Coils, calculated by subtracting ambient dry-bulb temperature from condensing temperature (CT)
COA	Condenser over air temperature, CT minus ambient temperature

The values for the parameters in

Table 2-2 were calculated using the following equations.

> ET, Lookup saturation temperature based on suction pressure from property table in Appendix A
>
> SC = CT – LT
>
> SH = ST – ET
>
> ETD = $RAT_{db} – SAT_{db}$
>
> CTD = CA – OAT
>
> COA = CT – OAT

Values for the parameters ET, SC, SH and COA were determined for each stage. Values for the parameters ETD and CTD were determined for a unit independent of the number of stages.

Value ranges for the parameters defined in Table 2-2 were specified and used to qualitatively diagnose the current operating condition of the unit.[2] These ranges are shown in Table 2-3. For example, if the evaporator temperature (ET) was below the range, given for normal operation, an assessment of "Low" was given.

[2] These values were developed during ASHRAE TRP 1274, where ADM worked with a committee of ASHRAE experts to establish criteria to be used for in-field diagnosis of faults for HVAC units. Note that there are still is not consensus on all the target and threshold values in fault detection; there are gray zones for some values.

Table 2-3. Range for Proper A/C Operation

Parameter	Standard Efficiency Units		High Efficiency Units[1]	
	Target	Normal Operation Range	Target	Normal Operation Range
ET (°F)	40	30 - 50	43	33 - 50
SC (°F)	15	8 – 20	10	3 – 15
SH (°F)	20[2]	-10/+10 of target value and minimum is 5 F	20[2]	–10/+10 of target value and minimum is 5 F
ETD (°F)	20	15 – 30	25	20 – 40
CTD (°F)	20	10 – 30	15	5 – 25
COA (°F)	20	10 – 30	15	5 – 25
Indoor airflow (CFM/ton)	400	320 – 480	400	320 - 480

[1] Where High Efficiency Units have SEER >=12.

[2] For TxV unit, target value is 20; for non TxV unit derive from Table 3-1..

After calculating values for the parameters, Tables 2-4a, 2-4b, and 2-4c were used to qualitatively diagnose common operational faults for HVAC units (e.g., refrigerant under or over charge, compressor valve leak, liquid-line restriction, condenser fouling and evaporator fouling problems). As an example, evaporator coil fouling causes ET to drop below the expected range, measured SH (SH_m) to drop below the expected range, and ETD to increase to above the expected range, while the other three parameters remained normal. Consequently, this combination of values for all six parameters indicates that the unit has fouled evaporator coils that need cleaning.

Table 2-4a. Operational Fault Detection and Diagnostics Matrix

Fault	***Expansion ValveType***	***ET***	***SH_m***	***CTD***	***SC_m***	***COA***	***ETD***
Inefficient Compressor	All	⇑*					
Condenser coil fouling	All			⇑	- /⇓	⇑	
Evaporator coil fouling	All	⇓	⇓				-/⇑
Refrigerant –Low charge	TXV	⇓			⇓	- /⇓	
Refrigerant –Low charge	nTXV	⇓	⇑		- /⇓		
Refrigerant – High Charge	TXV	- /⇑	- /⇓		⇑		
Refrigerant – High Charge	nTXV	- /⇑	⇓				
Refrigerant – Non condensables	All			- /⇓	- /⇑	⇑	
Liquid-line restriction	All	⇓	⇑		⇑		

Table 2-4b. Operational Fault Detection and Diagnostics Matrix, TXV Malfunction

Fault	***Type***	***SH_m***
TXV malfunction	TXV	⇑/⇓

Table 2-4c. Airflow Operational Fault Detection and Diagnostics Matrix

Fault	***Type***	***CFM per ton***
Airflow	All	⇓/⇑

Symbols: ⇑ = parameter higher than normal range,
⇓ = parameter lower than normal range,
⇑* = High range starts 15° above ET target value.
- = Parameter within normal range.
/ = Or

Sources: Braun, J.E., "Automatic Fault Detection and Diagnostics for Vapor Compression Cooling Equipment, Diagnostics for Commercial Buildings: Research to Practice", Conference Paper, Lawrence Berkeley National Laboratory, Berkeley, CA, June 1999.

If the measured superheat was high or low, with no other fault indicated, then the thermal expansion valve (TXV) was inspected.

3. FIELD MEASUREMENTS AND SERVICING

This chapter describes how the performance assessment of packaged units was conducted. In conducting the performance assessment in the field, ADM staff took the following steps:

- Select sites at which to conduct the performance assessments;
- Measure and document the performance of units at the selected sites;
- Diagnose all units needing service

Each activity is discussed in turn.

3.1 SELECTION OF UNITS FOR FIELD TESTING AND MEASUREMENTS

Data with which to assess the effects of CPACS maintenance and tune-up services were developed through field measurements for a sample of residential air conditioning units. The sample of units for the measurement work was selected from among households who had participated in a demand response program that SCE had implemented.

The pool from which to recruit households for the field testing and measurements included 148 households, with a total of 168 air conditioning units. Letters explaining the testing project were sent to all 148 households. Follow-up telephone calls were then made to the households to recruit them for the project. The result of this recruiting effort was that 106 households with 120 air conditioning units agreed to participate in the testing project.

3.2 MEASURING PERFORMANCE OF UNITS IN THE FIELD

Data on the performance of the selected units were collected according to the procedures described in Chapter 2.

3.2.1 Equipment Used

ADM staff used the following equipment for the performance testing.

- ***Sper Scientific model 800027*** relative humidity / temperature monitors with remote sensors were used to measure duct, ambient and condenser air relative humidity and temperature.
  - The temperature range is –14ºF to 122ºF, with an accuracy of ± 2ºF from the factory and a resolution of 0.1ºF.
  - All units were calibrated in an environmental chamber for an accuracy of ± 0.2ºF.
  - The relative humidity range is 20% to 99%, with an accuracy of ± 4% from the factory, with the unit calibrated at manufacturer-recommended intervals, and a resolution of 1%.
- The dual input ***Extech 421502*** type K thermocouple thermometer is used to measure refrigerant liquid line temperatures and ambient and condenser air temperatures. Three of these meters will be needed for two stage units. The range for the Type K thermocouple is –328ºF to 2500ºF, with an accuracy of ± 0.05% of reading + 0.6ºF and a resolution of 0.1ºF.

A type K thermocouple with a pipe clamp probe on a 10' cable was used for the refrigerant lines. Type K air probes will be used for the air measurements.

- A Power Meter was used to measure True RMS voltage, current, power and power factor. The model used was an ***AEMC 3910*** power meter with a current range from 1 to 500 Amps, voltage range from 0 to 600V, and power range from 30 W to 300 kW. The current accuracy is ±2% of full scale, the voltage accuracy is ±0.3%, and the power accuracy is the sum of the current and voltage accuracy (or ±2.3%). The voltage resolution is 1 Vac, and the current resolution is 0.1 Amps.
- An Ammeter was used to compare the calibration of the current reading of the power meter. The model used was a ***Fluke 33 true RMS clamp*** on ammeter. This meter has a range of 0 to 400 Amps with an accuracy of ±2% of full scale and a resolution of 0.01 Amps below 100 Amps.
- Refrigerant manifold gauges were used to measure the pressure within both the high and low side of the refrigeration system. The model used was a ***Ritchie model 41232*** that has an accuracy of ±1% of full scale. The range is 0 to 120 psi on the low side and 0 to 500 psi on the high side. These gauges are rated for R-12, R-22 and R502.
- Duct Blaster
- Blower Door

The field staff calibrated all test equipment according to manufacturer's recommendations. In addition, the staff checked all equipment periodically to ensure proper operation and that calibration was within field testable conditions.

Field calibration was accomplished by comparing the measurements made by two separate instruments.

- Calibration for the Sper RH monitors used to measure humidity was checked by comparing the humidity readings for 12 to 14 of the monitors. Any unit whose humidity reading was more than 4% different from the average reading was recalibrated using 33% and 75% humidity salts reference bottles.
- The field staff checked calibration of the equipment used to make electrical measurements, by using readings from the AEMC 3910 power meter and the Fluke 33 true RMS clamp-on ammeter. Readings from the two instruments were compared on the first A/C unit of the day. After the power had been allowed to stabilize, current measurements were made for each phase. The measurements of the two units were compared to determine whether they were within 4% of each other. If the two readings differed by more than 4%, the readings were compared with the readings from the ammeter that the HVAC technician carried. The unit farthest from the reading of the third instrument was sent to the calibration lab. Upon return from the calibration lab, the AEMC 3910 and Fluke 33 meters were compared again.
- The field staff checked calibration of the equipment used to measure refrigerant line pressure by comparing readings using the Ritchie model 41232 and the gauges the HVAC technician

carries. Readings from the two instruments were compared on the first A/C unit tested in a day. After the HVAC unit had been on for 15 minutes (to allow the unit to stabilize), pressure measurements were made with each gauge. If the measurements of the two units were not within 3% of each other, readings from both gauges of the Ritchie instrument on the low pressure side were compared. If these readings were more than 2% of each other, the gauges were sent to the calibration lab. Upon return from the calibration lab, the Ritchie and technician gauges were compared again and any differences noted for future comparisons.

3.2.2 Summary of Field Measurement Protocol

Following is a summary of steps in the field measurement protocol.

1. The outdoor AC unit was located and visually inspected. Any abnormal conditions were noted on a Measurement Form (see Appendix B).
2. Unit type, nameplate information, refrigerant type and expansion device type were documented..
3. Equipment was set up to take "near simultaneous" measurements. This included pressure gauges on all stages, clamp-on thermocouples for refrigerant lines, power meter on incoming electrical lines, temperature and humidity sensors in the supply and return air ducts, outside air and air off the condenser.
4. Unit was run for at least 10 minutes prior to recording measurements. The ambient air had ti be at least 65 ºF.
5. Any panel covers that had been removed and altered unit air flow were replaced.
6. Measurements were made from all meters listed in step 3 as quickly as possible (less than 10 minutes) and documented on the form.
7. Static pressure in the supply plenum was measured under normal operating conditions.
8. System duct air velocity for the return duct was measured and documented.
9. All supply registers were sealed with duct mask.
10. Ducts were pressurized to 25 Pascals and ½ system static pressure. Duct leakage rates were documented in cfm.
11. Blower door was installed in outside doorway and house was pressurized to 25 Pascals and ½ system static pressure. Duct leakage rates to unconditioned space (in cfm) were documented.
12. All measurement data were entered into a spreadsheet that checked data validation and applied algorithms for fault diagnosis.
13. Field staff determined if program verified all data
14. Meters and gauges were removed, any open panels were closed, and outside air intake was unsealed.

3.3 DIAGNOSES OF PERFORMANCE

After baseline measurements were made on the sampled units, the staff used the diagnostic testing procedures described in Chapter 2 to identify units for which refrigerant charging or other servicing was needed.

3.3.1 Assessment of Refrigeration Temperature Measurements

An important input to the diagnosis of refrigerant charge for air conditioning units is data on refrigerant line temperatures. To assess the sensitivity of such temperature measurements to the method of measurement, in-field measurements were made for a sample of units by using both a clamp-on thermocouple and an insulation wrapped bulb temperature sensor.

Using the two instruments, a first set of measurements was made for liquid line temperatures. There were 42 pairs of measurements in this set. Figure 3-1 compares the liquid line temperature measurements from the two methods. The average temperatures measured were 96.0 °F using the clamp-on and 94.2 °F using the insulated bulb.

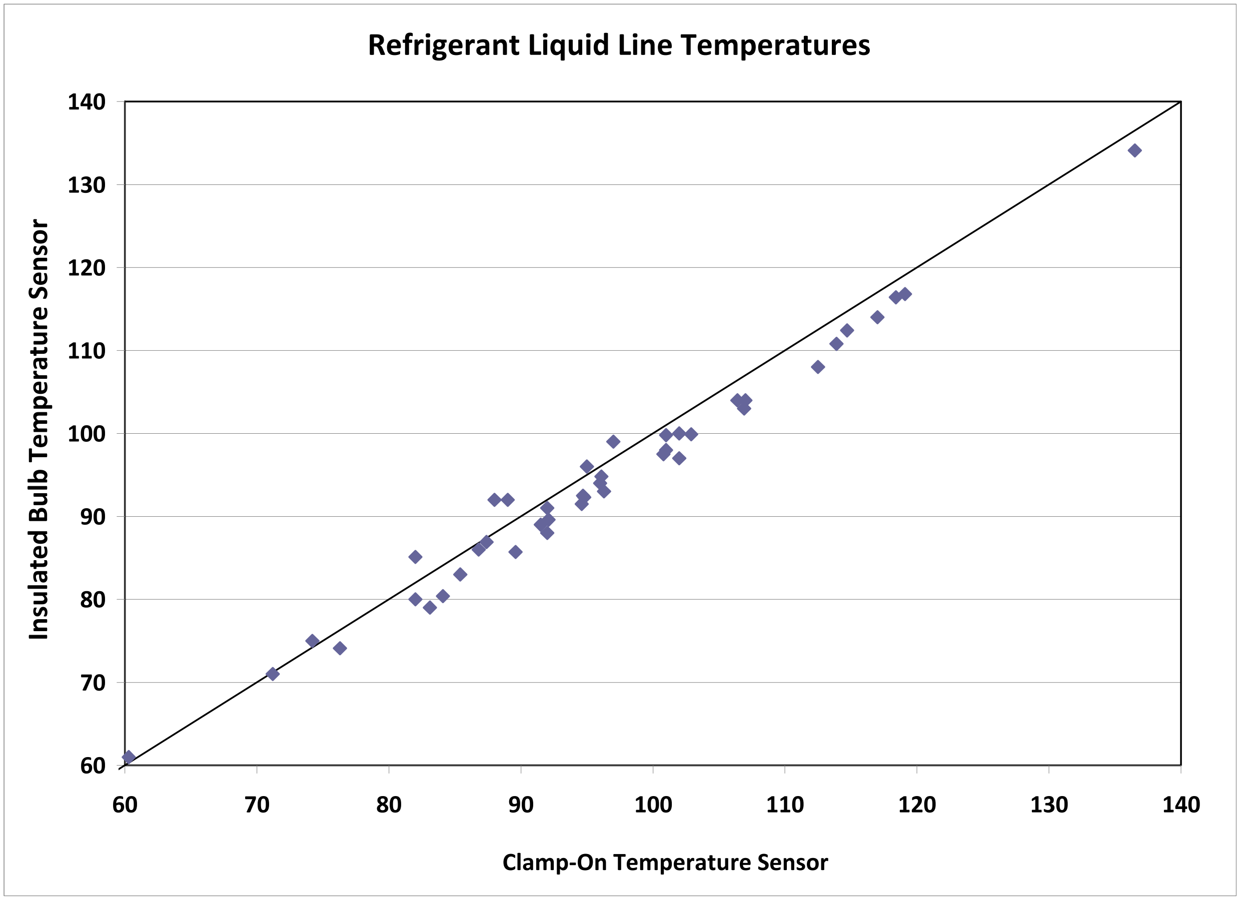

Figure 3-1. Comparison of Liquid Line Temperatures between Clamp-on Thermocouple and Insulation-Wrapped Bulb Temperature Sensor

A second set of temperature measurements was made on the low (suction) side. There were 40 pairs of measurements in this set. Figure 3-2 compares the low side temperature measurements from the two methods. The average low side temperatures measured were 57.4 °F using the clamp-on and 56.0 °F using the insulated bulb.

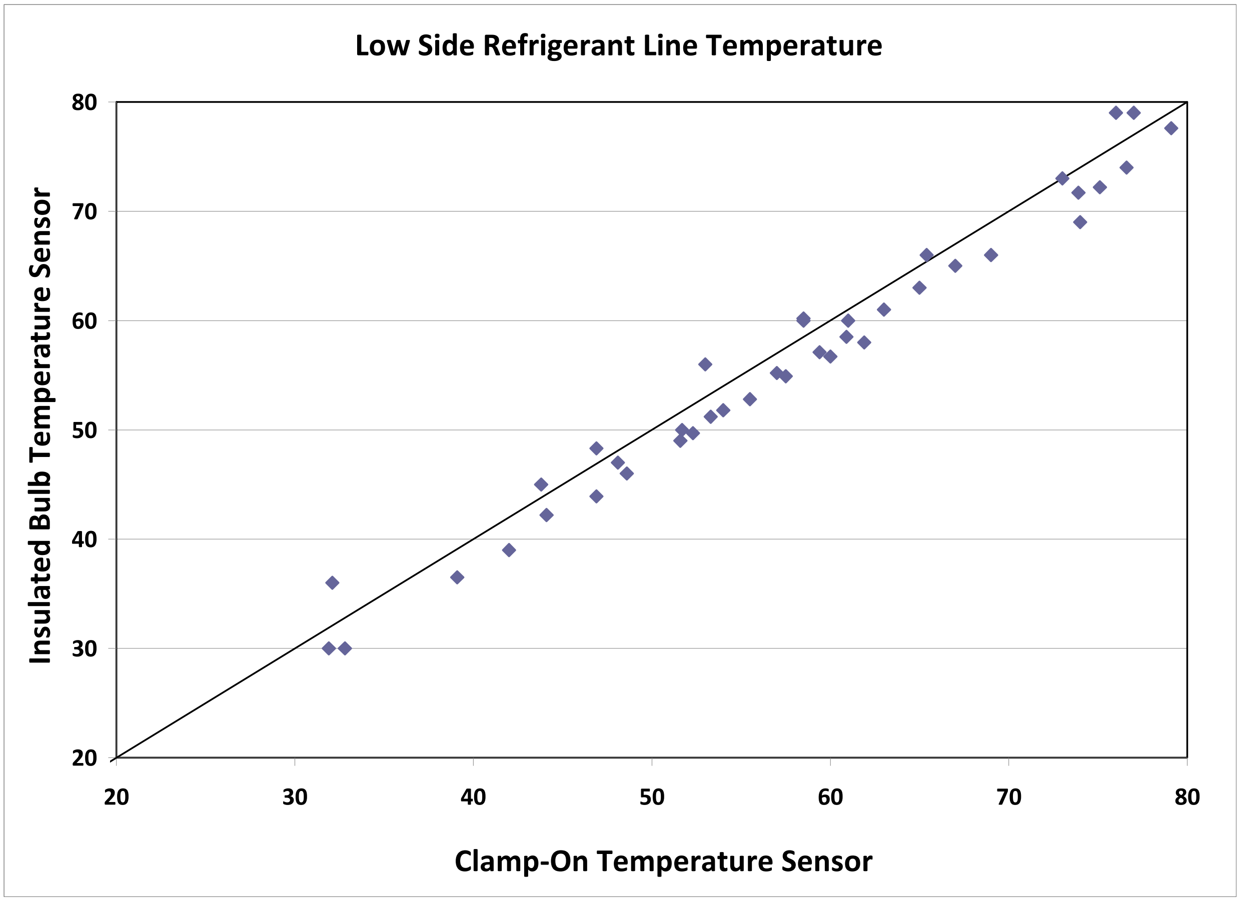

Figure 3-2. Comparison of Low (Suction) Side Temperatures between Clamp-on Thermocouple and Insulation-Wrapped Bulb Temperature Sensor

On average, the difference between the two types of temperature sensors was less than 2 °F for each set of measurements.

3.3.2 Diagnosis of Refrigerant Charge

The procedure for verifying the proper charge for the units based on the diagnostic procedures depended on whether the unit did or did not have a Thermal Expansion Valve (designated with (TXV) and (nTXV), respectively).

For units with or without TXV values, field staff used measurements of return air wet bulb temperature (RATwb), outdoor air dry bulb temperature (OATdb), suction line pressure (SP), suction line temperature (ST), as well as the implicit evaporator saturation temperature (ET) presented in Table A-1, to determine the charge condition as follows:

- Use RATwb and OATdb parameters and look-up table (

Table 3-1) to determine the required superheat (SHr).[3]

- Calculate Actual Superheat (SHm) = ST – ET

For these units, the sub-cooling (SC) levels were also assessed. A high-efficiency unit is expected to maintain the required sub-cooling (SCr) at 15°F and a standard efficiency unit at 10°F. Using the high-side pressure (LP) measurement and the measured liquid line temperature (LT), the staff determined measured sub-cooling as shown below:

- For a given refrigerant, the Condenser Saturation Temperature (CT) is the saturation temperature at the high-side pressure (LP).
- Measured sub-cooling (SCm) = CT – LT

Upon identifying SH levels and SC levels, quality codes were assigned to the following parameters:

- $SH_{Measured}$ > SHr + 10°F, "High"; $SH_{Measured}$ < SHr - 10°F, "Low"; else "Normal."
- $SC_{Measured}$ > SCr + 5°F, "High"; $SH_{Measured}$ < SHr - 7°F, "Low"; else "Normal."

In addition to the above SH and SC quality codes, the evaporative temperature (ET) and condenser temperature over air (COA) were also assigned a quality code:

- High efficiency ET threshold is 43°F, and the standard efficiency ET threshold is 40°F.
- High efficiency COA threshold is 15°F, and the standard efficiency COA threshold is 10°F.
- $ET_{Measured}$ > 50°F, "High"; $ET_{Measured}$ < ETr - 10°F, "Low"; else "Normal."
- $COA_{Measured}$ > COAr + 10°F, "High"; $COA_{Measured}$ < COAr - 10°F, "Low"; else "Normal."
- If $ET_{Measured}$ is "Low, $SC_{Measured}$ is "Low," and $COA_{Measured}$ is "Low" or "Normal," then units with TXV are diagnosed as "Low Charge."
- If $ET_{Measured}$ is "Normal" or "High," $SC_{Measured}$ is "High," and $SH_{Measured}$ is "Low" or "Normal," then units with TXV are diagnosed as "High Charge."
- If $ET_{Measured}$ is "Low," $SH_{Measured}$ is "High," and $SC_{Measured}$ is "Low" or "Normal," then units without TXV are diagnosed as "Low Charge."
- If $ET_{Measured}$ is "Normal" or "High," and $SH_{Measured}$ is "Low," then units without TXV are diagnosed as "High Charge."

[3] The values in Table 3-1 are similar to those in Title 24 documents (e.g., 2005 ACM Manual for Residential Standards) when the values are rounded off to full digit. That is, the target SH numbers in Table 3-1 and Title 24 numbers fall to the same value if full digit numbers are considered. Title 24 manuals for non-residential standards have no references to target SH data..

The super-heat table used is from Carrier, which provides the most widely used super-heat tables. Other manufacturers have SH charts specific to a model. There are no sources where all of the various make and model air conditioning unit data have been combined. Field diagnostics need to be able to utilize a table of SH values independent of the specific unit or the process will have short comings that will make it less reliable because of input problems.

Table 3-1. Required Superheat (°F) Calculator (non-TXV)

Required Superheat for units with no TXV device (nTXV)													
Measured Return Air Wet-Bulb Temperature, RATwb °F	Measured Outdoor Condenser Entering Air Dry-Bulb Temperature, OATdb °F												
	55	**60**	**65**	**70**	**75**	**80**	**85**	**90**	**95**	**100**	**105**	**110**	**115**
76	45	43	41	39	37	35	33	31	29	27	26	25	23
74	42	40	38	36	34	31	30	27	25	23	22	20	18
72	40	38	36	33	31	28	26	24	22	20	17	15	14
70	37	35	33	30	28	25	22	20	18	15	13	11	8
68	35	33	30	27	24	21	19	16	14	12	9	6	5
66	32	30	27	24	21	18	15	13	10	8	5	5	5
64	29	27	24	21	19	15	11	9	6	5	5	5	5.
62	26	24	21	19	15	12	8	5	5	5	5	5	5
60	23	21	19	16	12	8	5	5	5	5	5	5	5
58	20	18	16	13	9	5	5	5	5	5	5	5	5
56	17	15	13	10	6	5	5	5	5	5	5	5	5
54	14	12	10	7	5	5	5	5	5	5	5	5	5
52	12	10	6	5	5	5	5	5	5	5	5	5	5
50	9	7	5	5	5	5	5	5	5	5	5	5	5

Source: Charging Procedure for Residential condensing Units, Form No SK 28-01, Catalog Number 020-122, Carrier Corporation, 1994. All blank values in the original table were assigned a value of 5.

4. ANALYSIS OF RESULTS: AIR CONDITIONER PERFORMANCE

This chapter presents a summary of statistics and analysis for the data collected through the field measurements. Measurement data for before and after servicing are presented for the units for which servicing was conducted. In particular, changes in EER associated with the servicing are presented and discussed.

4.1 BASELINE CHARACTERISTICS OF UNITS TESTED

Performance testing was applied to a total of 120 HVAC units in their as-found condition. Units selected for the baseline measurement effort represented different sizes and ages.

- With respect to size, 53 units had nominal capacities of less than 4 tons and 67 units had capacities of 4 tons or more.
- With respect to age, 55 units were 10 years or less old and 65 units were over 10 years old.

Based on this size and age stratification, there were four categories into which the units fell. Table 4-1 shows the distribution of the units across these categories.

Table 4-1. Distribution of HVAC Units by Size and Age

Age of Unit	***Size of Unit***		***Totals***
	Under 4 Tons	***4 Tons or More***	
10 Years or Less	25	30	55
More than 10 Years	28	37	65
Totals	53	67	120

Of the 120 units, 76 had reciprocal compressors and 44 had scroll compressors. The distribution of the 120 units by size, age and type of compressor is shown in Table 4-2.

Table 4-2. Distribution of Sample Units by Size, Age and Type of Compressor

Size Group	***Age Group***	***Type of Compressor***		***Totals***
		Reciprocal	***Scroll***	
Under 4 Tons	***10 Years or Less***	15	10	25
Under 4 Tons	***More than 10 Years***	23	5	28
4 Tons or More	***10 Years or Less***	9	21	30
4 Tons or More	***More than 10 Years***	29	8	37
Totals		76	44	120

Units were also characterized by whether or not they were high efficiency. Out of 120 units, 31 units were high-efficiency. A distribution of the 120 units by size, age, and whether or not they were high efficiency, is shown in Table 4-3.

Table 4-3. Distribution of Sample Units by Size, Age and Whether High Efficiency

Size Group	Age Group	High Efficiency?		Totals
		No	Yes	
Under 4 Tons	10 Years or Less	18	7	25
Under 4 Tons	More than 10 Years	25	3	28
4 Tons or More	10 Years or Less	14	16	30
4 Tons or More	More than 10 Years	32	5	37
Totals		89	31	120

There were 104 units that did not have thermal expansion valves and 16 that did. The distribution of the 120 units by size, age and whether or not they had a thermal expansion valve is shown in Table 4-4.

Table 4-4. Distribution of Sample Units by Size, Age and Presence of Thermal Expansion Valve

Size Group	Age Group	Had TXV?		Totals
		No	Yes	
Under 4 Tons	10 Years or Less	21	4	25
Under 4 Tons	More than 10 Years	27	1	28
4 Tons or More	10 Years or Less	21	9	30
4 Tons or More	More than 10 Years	35	2	37
Totals		104	16	120

Using the data collected through the field measurements, EER values at standard conditions were calculated for 109 units. (Standard conditions are defined as 95ºF outside air temperature and 67 ºF return air wet-bulb temperature.) Table 4-5 reports the averages and standard deviations for the normalized EER for the different size and age groups. There is no significant differences between these groups when the standard deviation is considered.

Table 4-5. Average EERs before Servicing for Sample Units, Calculated at Standard Conditions, by Size and Age

Size Group	Age Group	Number of Units	Average Measured Baseline EER	Standard Deviation of EER
Under 4 Tons	10 Years or Less	24	7.61	2.59
Under 4 Tons	More than 10 Years	27	6.75	2.29
4 Tons or More	10 Years or Less	27	5.91	2.37
4 Tons or More	More than 10 Years	31	6.07	2.34
Totals		109	6.54	2.45

4.2 CALCULATION OF PARAMETERS FOR FAULT DIAGNOSIS

As discussed in Section 2.7.2, field staff used data collected through field measurements to calculate several different parameters for use in qualitatively diagnosing the current operating condition of a unit. These parameters were as follows:

- ET: Evaporator temperature, measured as T_{sat} at low side pressure
- SC: Sub-cooling, condenser saturation temperature at high-side subtracted by liquid line temperature
- SH: Superheat
- COA: Condenser over air temperature, condensing temperature minus ambient
- ETD: Temperature drop across evaporator
- CTD: Air Temperature Increase over Condenser Coils, calculated by subtracting ambient dry-bulb temperature from condensing temperature (CT)

The formulas for calculating the values for these parameters were presented in Section 2.7.2.

Summary statistics for the calculated values of ET, SC, SH, COA, ETD, and CTD are presented in Table 4-6.

Table 4-6. Summary Statistics for Calculated Values of ET, SC, SH, COA, ETD and CTD

	Non TXV	***TXV***	***All***
	ET		
N	94	15	109
Mean ET	37.23	34.73	36.89
Standard Deviation of ET	8.23	13.03	9.01
	SC		
N	94	15	109
Mean ET	15.87	11.42	15.26
Standard Deviation of ET	8.80	6.65	8.65
	SH		
N	94	15	109
Mean ET	17.82	26.65	19.04
Standard Deviation of ET	15.77	19.92	16.58
	COA		
N	94	15	109
Mean ET	23.65	15.87	22.58
Standard Deviation of ET	8.10	6.48	8.32
	ETD		
N	94	15	109
Mean ET	21.42	18.65	21.04

Standard Deviation of ET	4.73	5.40	4.89
CTD			
N	94	15	109
Mean ET	12.94	8.73	12.36
Standard Deviation of ET	4.70	4.24	4.85

As described in Section 2.7.2, value ranges for these various parameters were specified and used to qualitatively diagnose the current operating condition of the unit. (These ranges were shown in Table 2-3.) Based on a comparison of the measured value of a parameter to its target value, an assessment was made of whether the measured value was "Low," "Normal," or "High". The distributions of these assessments are shown in Table 4-7.

Table 4-7. Distribution of Assessment Values for ET, SC, SH, COA, ETD and CTD (Total n = 109)

	Low	***Normal***	***High***
ET	19	86	4
SC	18	57	34
SH	31	37	41
COA	3	83	23
ETD	19	85	5
CTD	22	87	0

The driving conditions under which the measurements were conducted are characterized in the following figures.

- Figure 4-1 presents the outdoor air dry bulb temperature and the outdoor air relative humidity conditions under which unit diagnoses were performed.
- Figure 4-2 presents the corresponding return air wet bulb temperature and condenser air temperature conditions.
- Finally, Figure 4-3 shows the supply air dry bulb temperature and the return air dry bulb temperature for the unit diagnoses.

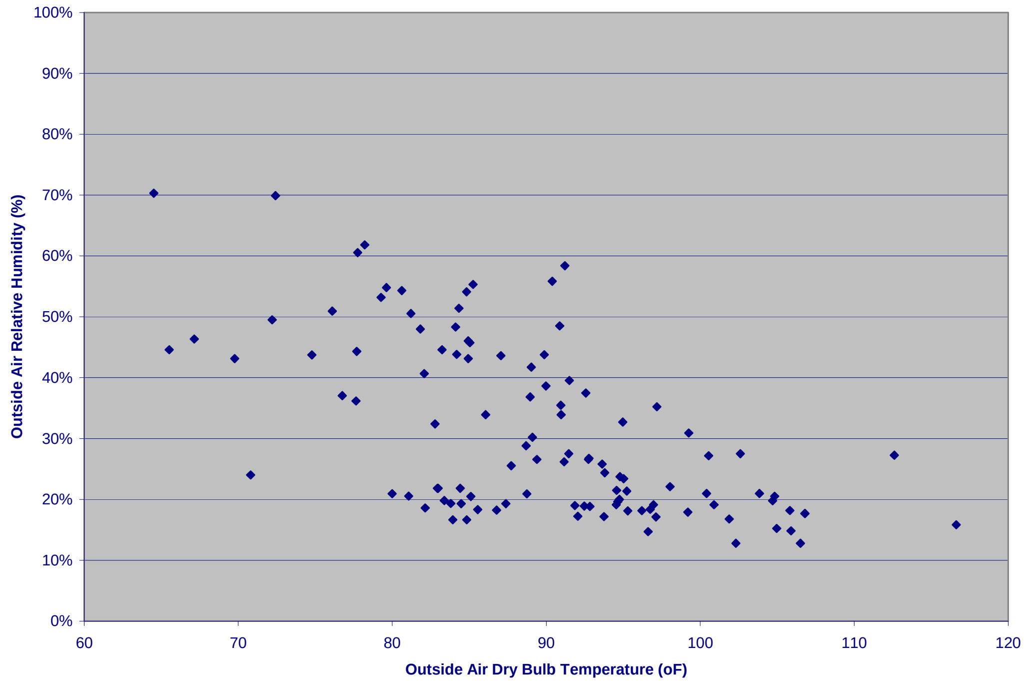

Figure 4-1. Outdoor Air Dry Bulb Temperature and Outdoor Air Relative Humidity

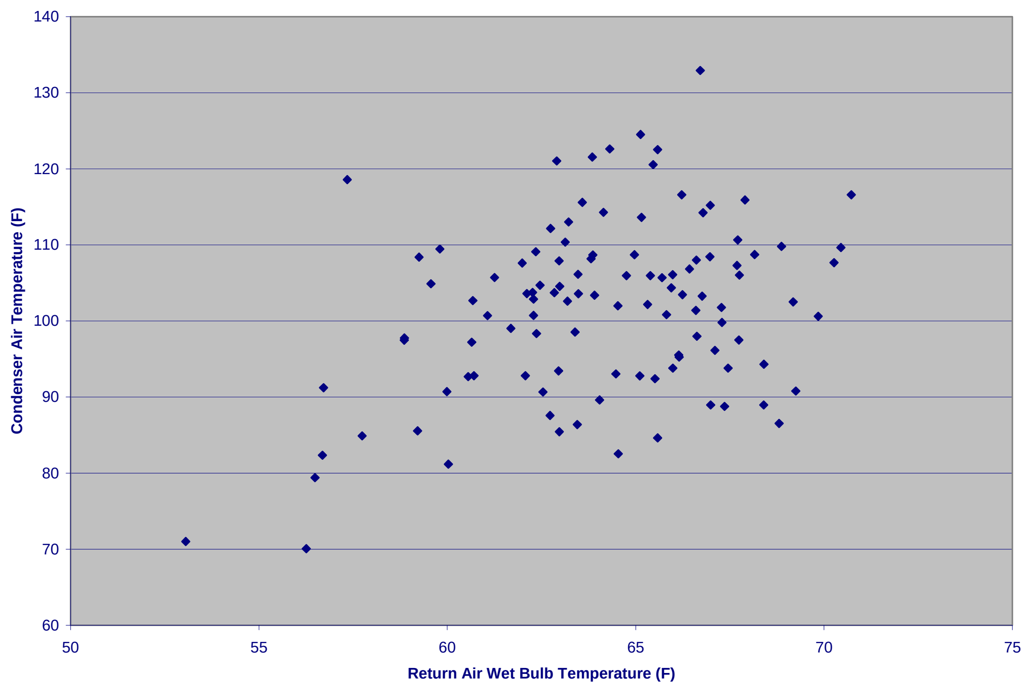

Figure 4-2. Return Air Wet Bulb Temperature and Condenser Air Temperature

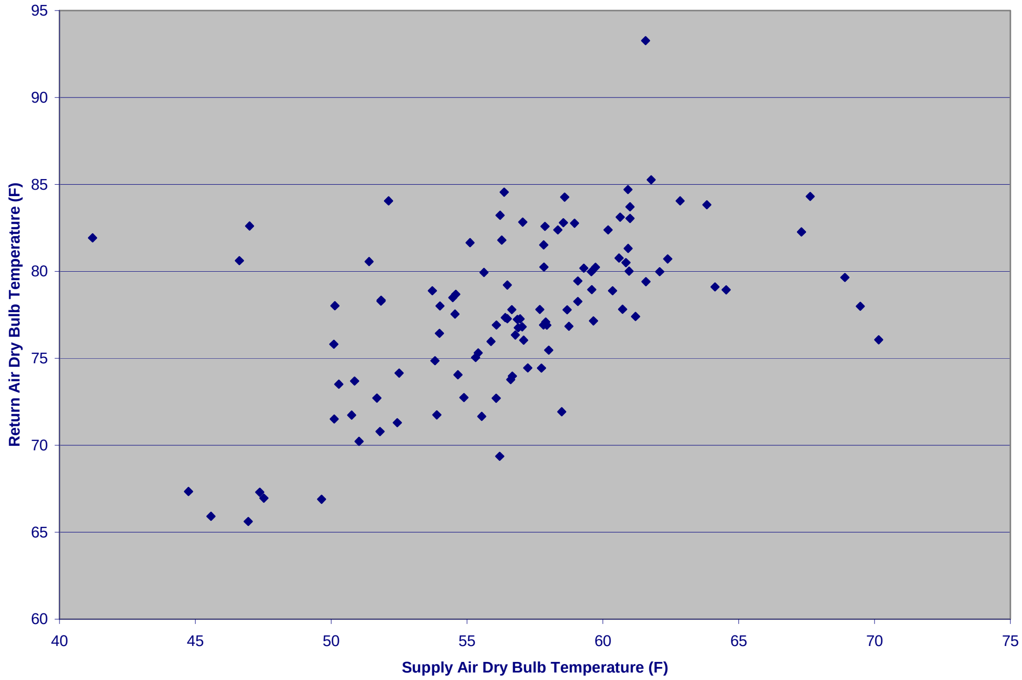

Figure 4-3. Supply Air Dry Bulb Temperature and Return Air Dry Bulb Temperature

4.3 RESULTS OF FAULT DIAGNOSES

The diagnostic procedures for detecting faults are based on the presumption that there is proper air flow across the coil. Therefore, for determining what servicing was required, air flow was measured first. If those measurements showed that there was a problem with the air flow, those problems were fixed before further measurements were made. The values that were calculated for the parameters ET, SC, SH, COA, ETD and CTD were used to qualitatively diagnose common operational faults for HVAC units (e.g., refrigerant under or over charge, compressor valve leak, liquid-line restriction, condenser fouling and evaporator fouling problems). Other than air flow, most of the diagnostic faults are independent of each other.

Baseline measurements can only be made with a unit in its existing condition. Diagnostics of multiple faults from one baseline test is not conclusive.

In practice, servicing should be based on a hierarchy of fault detection. A guideline for servicing of faults is provided in Table 4-8. Consistent with diagnosis results, servicing begins with items 1 through 4 in Table 4-8. After these items have been serviced, the unit is retested and another full set of measurements is made. These measurements are used to determine whether the unit needs a refrigerant charge adjustment (i.e., item 5 in Table 4-8 is checked). After a charge adjustment is made, the unit is again retested and another full set of measurements made. If necessary, servicing is performed for item 6 (although this item is not part of measurement diagnostic procedures).

Table 4-8. Hierarchy of Servicing Faults

Item	Fault Type
1	Incorrect Supply fan air flow across evaporator coil.
2	Non-condensibles in refrigerant line.
3	Condenser fouling.
4	Expansion device and liquid line restrictions.
5	Refrigerant charge incorrect.
6	Miscellaneous – tighten fan belt, mounting bolts, clean and drain condensate pan and line.

4.3.1 Results of Fault Diagnoses: All Units Tested

Table 4-9 reports the numbers of different types of faults that were determined for the units tested. Of the units tested, 72% had no or one fault. Only 28 % of the baseline units had more than one fault identified; for all of those units one of the faults was with the air flow. Low air flow can cause diagnosis to report false positives for other faults. Diagnosis of multiple faults by the evaluation team is not conclusive since faults should be corrected according to the hierarchy table 4-8. However, limitations prevent servicing by the evaluation team and multiple fault diagnosis may only be indicative and not conclusive of multiple problems.

Table 4-9. Results of Fault Diagnostic Testing (Total n = 109)

Type of Fault	***Number of Units with Fault***
Inefficient Compressor?	4
Refrigerant Flow Restriction?	4
Condenser Fouling?	0
Evaporator Fouling?	2
Charge Problem?	
High Charge	30
Low Charge	6
Non-condensable	12
Airflow?	
High CFM	2
Low CFM	59
Total number of units diagnosed	109

Table 4-10 reports the numbers of units with different numbers of faults. Of the 109 units for which diagnoses of faults were made, 89 were diagnosed with at least one fault. This table also presents average standardized EER values for units tested by number of faults. The standardized EERs are calculated from baseline measurements that have been normalized to standard conditions.

Table 4-10. Numbers of Faults Detected and Average Standardized EER, Rated and Measured Cooling Capacity and Measured kW Input for Units Tested

Number of Faults	Number of Units with Faults	Baseline EER for Standard Conditions		Rated Cooling Capacity, tons		Measured Cooling Capacity, tons		Total Measured kW Input	
		Average	Standard Deviation	Average	Standard Deviation	Average	Standard Deviation	Average	Standard Deviation
None	20	8.20	2.48	3.38	0.63	2.58	0.88	3.66	1.03
One	58	6.45	2.15	3.80	0.77	2.16	0.69	4.14	1.07
Two	27	5.81	2.38	3.93	0.90	1.93	0.75	4.11	0.93
Three	4	4.35	3.20	4.13	0.63	1.81	1.29	4.74	1.89
Total # of Units	109	6.54	2.45	3.77	0.79	2.17	0.79	4.07	1.07

As discussed in Section 3.3.1, data on refrigerant line temperatures is an important input to the diagnosis of refrigerant charge for air conditioning units. In-field measurements of refrigerant line termperatures were made for a sample of units by using both a clamp-on thermocouple and an insulation wrapped bulb temperature sensor. The data showed that there was a difference of about 1.8°F between the two temperature measurements.

The refrigerant line temperatures are used only in the tune-up diagnostics. Thus, the different temperature measurement techniques affect EER estimation only when the refrigerant line termperature measured through the two techniques is on the threshold of changing the diagnostics. To gauge the effect of the 1.8ºF difference in temperatures, diagnostics were run using two different refrigerant line temperatures for 43 units that received servicing from an HVAC contractor for which ADM field staff took pre-servicing and post-servicing measurements. The original diagnostics were determined using the temperatures measured through the use of the clamp-on thermocouple sensors. The diagnostics were then rerun with the liquid line temperature equal to the original measurement minus 1.8ºF and the low side suction line temperature substituted with the original measurement minus 1.4ºF.

For the 43 units, using the different temperature changed the baseline diagnostics from no problem to high charge for two units. Of the 16 units that originally had a high charge diagnosis, it was estimated that an additional 0.1 pounds of refrigerant should be removed for nine; there was no difference for the other seven units. Of the three units that originally had a low charge diagnosis, none of the diagnostics were changed because modified refrigerant line temperatures were used.

4.3.2 Results of Fault Diagnoses: Units Classified by Size and Age

Table 4-11 reports the numbers of different types of faults that were determined for the units tested categorized by age and size.

Table 4-11. Results of Fault Diagnostic Testing for Units Categorized by Age and Size

Unit Size (Tons)	***High Charge***	***Low Charge***	***No Problem***	***Non-condensibles***	***Total***
			Unit Age: 10 Years or Under		
2.0	1			1	2
2.5	1			1	2
3.0	3	1	11	1	16
3.5	1		3		4
4.0	2	1	9	1	13
5.0	3		9	2	14
Subtotals	11	2	32	6	51
			Unit Age: Over 10 Years		
2.5	1			2	3
3.0	4	2	6		12
3.5	2	2	5	3	12
4.0	11		11	1	23
5.0	1		7		8
Subtotals	19	4	29	6	58
Totals	30	6	61	12	109

Table 4-12 summarizes the number of faults detected for the units tested when the units are categorized by age and size.

Table 4-12. Number of Faults per Unit for Units Categorized by Age and Size

	Number of Faults				
Unit Size (Tons)	***None***	***One***	***Two***	***Three***	***Total***
			Unit Age: 10 Years or Under		
2.0		1	1		2
2.5		1	1		2
3.0	10	4	2		16
3.5	3	1			4
4.0	1	8	3	1	13
5.0	1	6	6	1	14
Subtotals	15	21	13	2	51
			Unit Age: Over 10 Years		
2.5		2	1		3
3.0	2	8	2		12
3.5	2	7	2	1	12
4.0		15	7	1	23
5.0	1	5	2		8
Subtotals	5	37	14	2	58
Totals	20	58	27	4	109

4.3.3 Refrigerant Charge Adjustments

Table 4-13 shows information of the refrigerant charge adjustments made for air conditioning units with servicing company records; also shown in the table are pre- and post-servicing measurements. Pre- and post-servicing measurements were made on different days from the actual servicing. A calculated estimate of the charge adjustment amount is also provided. There are 28 units for which all three sets of data are available. Calculation of estimated charge adjustment is based on a different algorithm than the charge diagnosis. Charges adjustments less than 4 ounces are generally not implemented, and it represents less than 5% of the charge for a typical system. Diagnosis of high charge with no calculated adjustment indicates only a marginally high charge diagnosis. The method of diagnosis used in the evaluation (see chapter 3) is more sophisticated than the Carrier method generally used by A/C service contractors.

There are some pre and post diagnostics in Table 4-13 that are not consistent with the documentation provided by the service contractors. The evaluation team was not allowed to witness the servicing of the units so documentation of servicing could be verified. The study was designed to be blind to the servicing contractors.

Table 4-14 tabulates the data in Table 4-13 to allow comparison between the results of the servicing company and the pre-servicing measurements made for this study. The service company’s action for refrigerant charge only matched the baseline diagnosis 32% of the units. The most common difference occurred where charge was added but the baseline evaluation diagnosed the charge to be correct. The evaluation diagnosed three units with non-condensibles, however this is not conclusive from the testing methodology and the service companies were not obligated to evacuate refrigerant to correct the problem as part of the CPACS program. There is speculation that the service companies may not always identify the correct refrigerant metering device which can cause a mis-diagnosis of charge.

There were 46 units for which there were records from the servicing company on refrigerant charge adjustments. For these 46 units, refrigerant was added for 22 units, removed for 13 units and not changed for 11 units. Table 4-15 shows the average charge adjustment amounts (added or removed) as reported in the records of the servicing company.

Figure 4-4 plots the refrigerant charge adjustment in ounces versus unit size in tons for 35 units for which charge was adjusted. Figure 4-5 shows the refrigerant charge adjustment in ounces plotted against nameplate unit charge amount in ounces for 29 units for which the charge was adjusted and nameplate data was available.

Table 4-13. Summary of Refrigerant Charge Adjustments

Unit ID#	Tons	Service Company		Pre-Service Measurement		Post-Service Measurement	
		Charge	Charge Adjustment (Oz.)	Charge Diagnosis	Calculated Adjustment (Oz.)	Charge Diagnosis	Calculated Adjustment (Oz.)
SCE002	4	No Change	0	Non-condensibles	Na	OK	
SCE003	4	High Charge	-21	High Charge	8	Low Charge	14
SCE008	5	Low Charge	14	OK		Low Charge	11
SCE012	5	High Charge	-82	High Charge	15	OK	
SCE014	3	No Change	0	High Charge	14	High Charge	
SCE023	4	Low Charge	2	High Charge	14	OK	
SCE025	3.5	High Charge	-18	High Charge	17	Low Charge	14
SCE027	4	High Charge	-22	High Charge	6	OK	
SCE032	3.5	High Charge	-19	High Charge	1	OK	
SCE035	3.5	High Charge	-13	Non-condensibles	Na	OK	
SCE040	4	High Charge	-8	High Charge	17	OK	
SCE041	3.5	Low Charge	8	OK		OK	
SCE042	5	Low Charge	14	OK		High Charge	
SCE043	5	High Charge	-13	OK		OK	
SCE069	4	Low Charge	3	OK		OK	
SCE071	5	Low Charge	10	OK		OK	
SCE083	3.5	No Change	0	OK		OK	
SCE089	3.5	High Charge	-3	High Charge	1	High Charge	
SCE097B	?	Low Charge	37	High Charge	12	High Charge	
SCE097 A	3.5	No Change	0	OK		OK	
SCE106	2	High Charge	-4	Non-condensibles	na	OK	
SCE108	2	Low Charge	8	OK		OK	
SCE109	4	High Charge	-4	OK		High Charge	
SCE110 A	2	High Charge	-4	OK		OK	

SCE140 A	4	Low Charge	81	OK		OK	
SCE143	5	Low Charge	16	High Charge		OK	
SCE146	4	No Change	0	High Charge	3	OK	
SCE147	4	Low Charge	4	OK		OK	

Table 4-14. Comparison of Servicing Company Diagnosis and This Study's Pre-Servicing Measurements

		Service Company Diagnosis			Totals
		Add Charge	Remove Charge	No Change	
Evaluation Pre-Servicing Measurements	Low Charge	0	0	0	0
	High Charge	3	7	2	12
	Ok	8	3	2	13
	Non-Condensibles	0	2	1	3
	Totals	11	12	5	28

Table 4-15. Refrigerant Charge Added or Removed

Charge	Average Amount (Oz.)	Standard Deviation (Oz.)	Number of Units
Added	16.3	17.8	22
Removed	-17.1	20.7	13

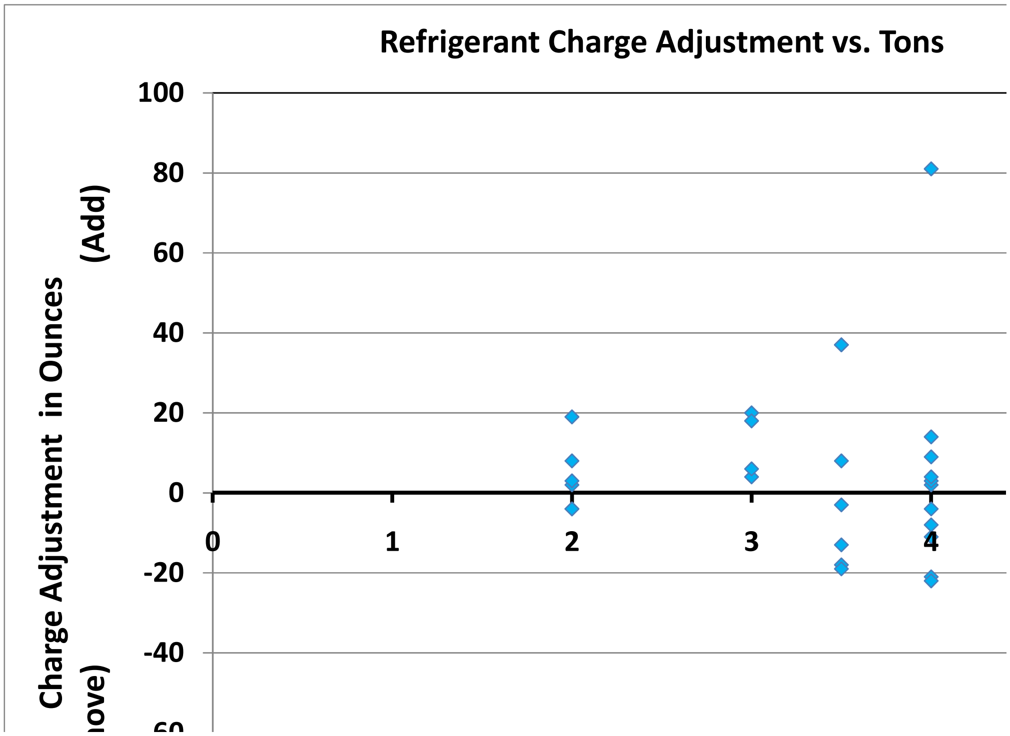

Figure 4-4. Amount of Refrigerant Charge Adjustment versus Unit Size (Tons)

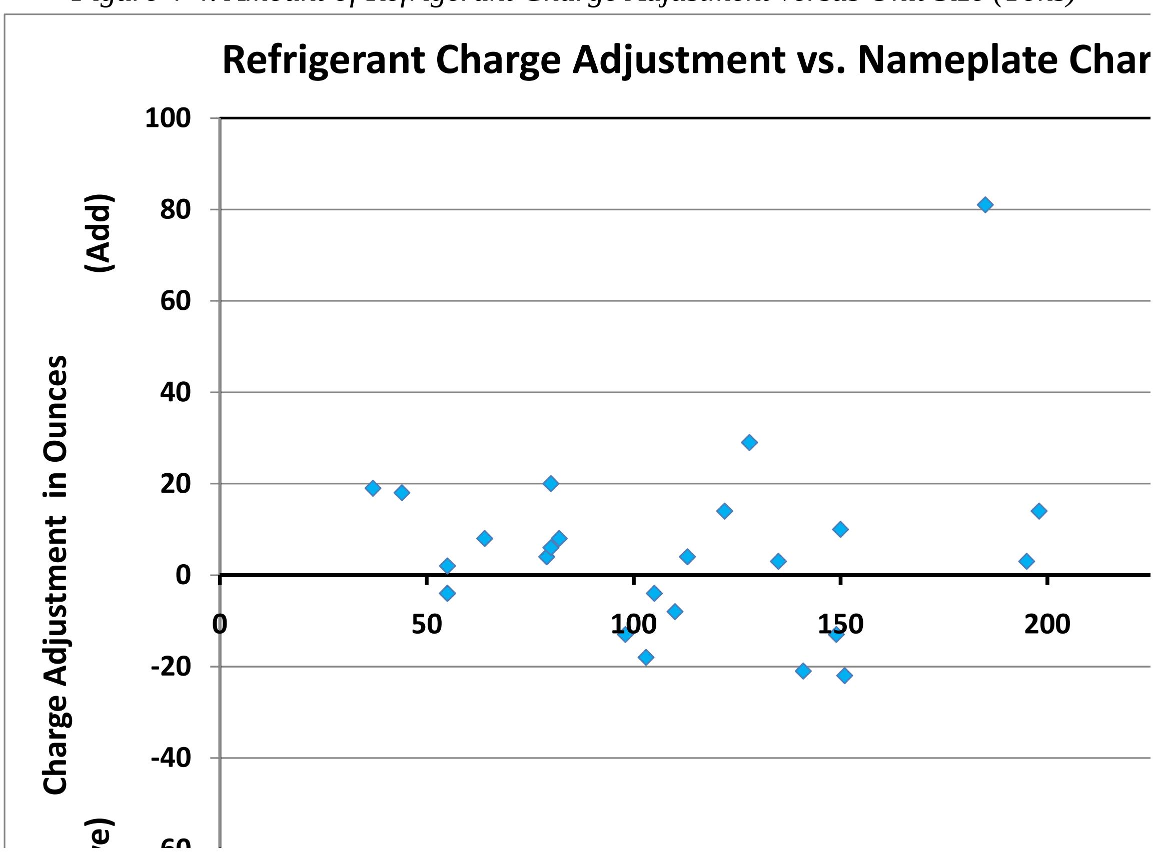

Figure 4-5. Amount of Refrigerant Charge Adjustment versus Nameplate Charge

4.4 EFFECTS OF REFRIGERANT CHARGE SERVICING

Only two types of servicing were reported by the HVAC contractor for the AC units addressed in this study: (1) adding or removing refrigerant charge and (2) duct leakage repair. The effects of refrigerant charge servicing are addressed in this section, while the effects of duct leakage repair are addressed in Chapter 5.

4.4.4 Effects of Refrigerant Charge Servicing on EERs

There were 43 units that received refrigerant charge servicing from an HVAC contractor for which ADM field staff took pre-servicing and post-servicing measurements. These measurements provide data for assessing changes in the performance of the units.

The pre- and post-servicing measurements were first used to analyze changes in the EERs for the units at standard conditions before and after the servicing. Figure 4-6 provides a graphical comparison of the pre- and post-servicing normalized EER for the 43 units using procedures discussed in Section 2. Table 4-16 reports the results of a paired t-test that was performed on the data for the units. The average EER for the units increased from 6.64 before servicing to 7.05 after servicing, an increase of about 6.1%. However, the results of the paired t-test show that the hypothesis of no difference between the pre- and post-servicing averages can be rejected only with a confidence level of 80%.

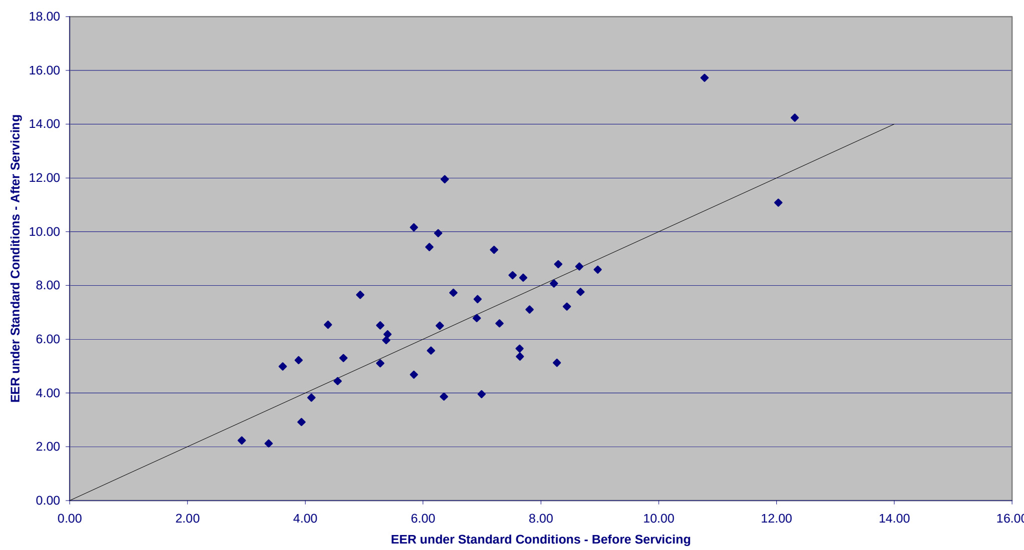

Figure 4-6. Comparison of Before- and After-Servicing EERs (EER Calculated for Standard Conditions)

Table 4-16. Results of Paired t-test on Units with EER Calculated from Before- and After -Servicing Data

	Before-Servicing EER	***After-Servicing EER***
Mean	6.64	7.05

Standard Deviation	2.13	2.87
Observations	43	43
Pearson Correlation	0.72	
Hypothesized Mean Difference	0.00	
Degrees of freedom	42	
t Stat	-1.341	
P(T<=t) one-tail	0.094	
t Critical one-tail	1.682	
P(T<=t) two-tail	0.187	
t Critical two-tail	2.018	

The effects of refrigerant charge servicing on the performance of air conditioning units categorized by age is shown in Table 4-17. Units less than 10 years old had an average performance improvement of 9% while older units, ten or more years old, only had an average performance improvement of 4%.

The effects of refrigerant charge servicing on the performance of air conditioning units categorized by size is shown in Table 4-18. The 2.5-ton units show a large average performance improvement, but these are based on a sample of only 3 units. The 4-ton units actually show a small decrease in performance, but this may not be statistically significant.

Table 4-17. EERs Before and After Refrigerant Charge Servicing for Units Categorized by Age

	Age	Mean EER under Standard Conditions	Standard Deviation of EER	Number of Observations
Before-Servicing	< 10 years	6.80	1.89	17
	10 + Years	6.54	2.30	26
	Total	6.64	2.13	43
After-Servicing	< 10 years	7.41	2.89	17
	10 + Years	6.81	2.89	26
	Total	7.05	2.87	43

Table 4-18. EERs Before and After Refrigerant Charge Servicing for Units Categorized by Size

	Unit Size (Tons)	Mean EER under Standard Conditions	Standard Deviation of EER	Number of Observations
Before-Servicing	2.5	5.21	1.25	3
	3.0	7.20	2.67	8
	3.5	7.16	1.85	8
	4.0	6.97	2.13	15
	5.0	5.61	1.85	9
	Total	6.64	2.13	43
After-Servicing	2.5	7.71	3.69	3
	3.0	7.75	3.43	8
	3.5	7.66	3.68	8
	4.0	6.86	2.35	15
	5.0	5.98	2.35	9
	Total	7.05	2.87	43

4.4.5 kWh Savings from Refrigerant Charge Servicing

Data on the kWh usage for the units that were serviced were collected through end use metering from the summer of 2005 to the summer of 2008. For 2007, there were 159 units for which there were end use metered data, running from 1/08/2007 to 11/08/2007. These data were used to determine the annual kWh usage for the units for 2007.[4] The average annual kWh usage per AC unit was 1,303 kWh with a range from 0 kWh (for 5 units) to 5,229 kWh.

[4] kWh usage was imputed for sites that did not have end use metered data for the entire period. The imputed data added 3.5% more energy use than what was already in the dataset.

As described in the previous section, savings from refrigerant charge tune-ups were estimated to be 6.1%. Thus, with a baseline kWh usage of 1,303 kWh per year, the annual kWh savings from the refrigerant charge tune-up is estimated to be 79.5 kWh.

5. ANALYSIS OF RESULTS: DUCT TESTING AND SEALING

As part of research on residential central air conditioning systems that was performed during this project, measurements of total duct leakage and of duct leakage to unconditioned space were made for a sample of houses. The results from analysis of the duct leakage measurements are presented in this chapter.

5.1 COMPARISON OF DUCT LEAKAGE MEASUREMENT METHODS

Conventional practice in measuring duct leakage is to use a duct pressurization test, usually with a standard reference positive air pressure of 25 Pascals. However, some studies have suggested that duct leakage measured at 25 Pascals may be overstating actual leakage. To examine this question, measurements of duct leakage were made at a sample of 21 existing houses using three methods of measurement.

Two of the methods for making the duct leakage measurements were variants of the usual duct pressurization method. The duct pressurization was performed by connecting a Duct Blaster® to the return side of the system. Total duct leakage was measured with the registers sealed and the Duct Blaster® pressurizing the duct system. Total Duct leakage was then measured for two sets of test conditions.

- One set of duct pressurization measurements was made using the standard fixed 25 pascals (Pa) pressurization.
- A second set of duct pressurization measurements was made by taking measurements at ½ system static pressure (SSP) for central air conditioning systems. SSP is a measurement of static pressure at the supply side plenum of the duct system when the supply fan is on and operating with registers in their normal position. This pressure is unique for each system. The rationale for using a modified SSP is to replicate the conditions that produce the weighted average pressure that the duct sees at the leakage locations.

Duct leakage to unconditioned space as well as total duct leakage was measured under both sets of conditions. A blower door was setup in an exterior doorway and was used to pressurize the house to the same pressure as the ducts. Duct leakage to unconditioned space was measured at 25 Pa and ½ SSP, when possible. In some cases leaky house envelopes did not allow pressurization of the house to the target duct pressures.

Tracer gas infiltration testing, which is regarded as one of the more accurate methods for measuring infiltration rates, was used as a third method of measurement to provide benchmark values for duct leakage against which measurement results from the duct pressurization methods could be compared and assessed. Carbon dioxide (CO_2) was used as the tracer gas for this testing.

For the tracer gas infiltration testing, CO_2 was released from a portable tank into a home at the return register with the system fan on to distribute the gas. Carbon dioxide meters with attached loggers were used to measure the CO_2 levels in parts per million (ppm). Concentrations of CO_2 were monitored with the system fan off and on. Two loggers were installed at two locations, one on a chair near the return register and the other most often in the living room or master bedroom. The loggers recorded CO_2 levels every 30 seconds. The volume of the house was determined by releasing a measured volume of CO_2 into the house, recording the peak concentration, and calculating the active net air volume. In all cases CO_2 levels remained well within safe limits. The tracer gas method used the natural system pressure when the supply fan is on, while the ½ SSP is based on an actual system measurement during normal operation.

Duct leakage measurements were made with all three methods for 21 houses. In addition, there were three houses where measurements could be made with the tracer gas method and with duct pressurization at ½ SSP, but not with duct pressurization at 25 Pascal. Accordingly, comparisons of the three methods are based on the measurement data from 21 houses.

For total duct leakage, Figure 5-1 compares the measurements from the two duct pressurization methods against the tracer gas measurements. Table 5-1 reports summary statistics for the different measurements. (Measurements reported in the graphs are calculated from data collected from loggers placed near the returns at each site.)

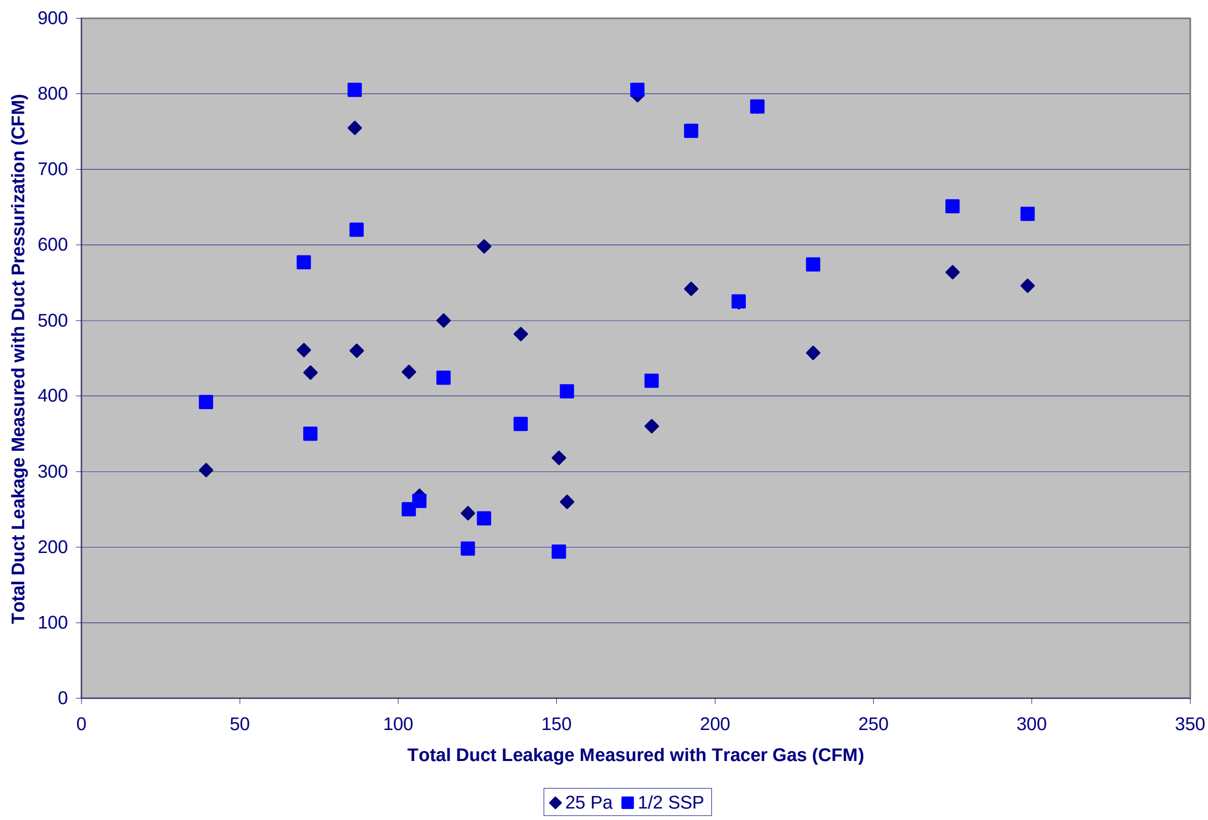

Figure 5-1. Total Duct Leakage (CFM) As Measured with Different Methods

Table 5-1. Summary Statistics for Total Duct Leakage Measurements (n= 21)

Method	***Measured CFM***	
	Average	***Standard Deviation***
CO_2 Tracer Gas Infiltration Testing	149.8	68.7
Duct pressurization at 25 Pascal	480.3	162.3
Duct pressurization at ½ SSP	487.0	205.2

The correlation between the total duct leakage CFM measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.313; for duct pressurization at ½ SSP the correlation was 0.397.

Duct leakage to unconditioned space was also calculated for the 21 test houses. The cubic feet per minute (CFM) of duct leakage to unconditioned space was calculated as:

$$CFM_{\text{Leakage to unconditioned space}} = CFM_{\text{Fan On}} - CFM_{\text{Fan Off}}$$

CFM for both fan-on and fan-off conditions is calculated as follows:

$$CFM = (ACH*Volume)/60$$

ACH = Air Changes Per Hour

$$Volume = \frac{\text{Volume of } CO_2 \text{ Injected} \times 1{,}000{,}000}{\text{PPM After Injection} - \text{PPM Before Injection}}$$

ACH was calculated by best fitting a curve a best fit to the CO_2 decay data over the monitored period. (The calculated ACH minimizes the chi-square error term of the fitted values.) An example of the CO_2 measured data along with the fitted ACH is shown in Figure 5-2.

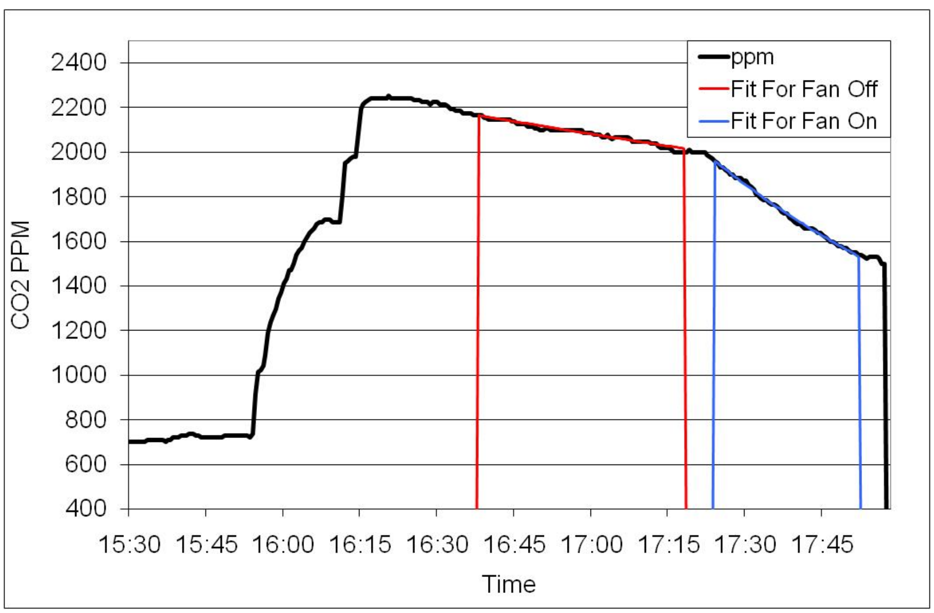

Figure 5-2. Example of Determining ACH through Curve Fitting

For duct leakage to unconditioned space, Figure 5-3 compares the measurements from the two duct pressurization methods against the tracer gas measurements. Table 5-2 reports summary statistics for the different measurements of duct leakage to unconditioned space.

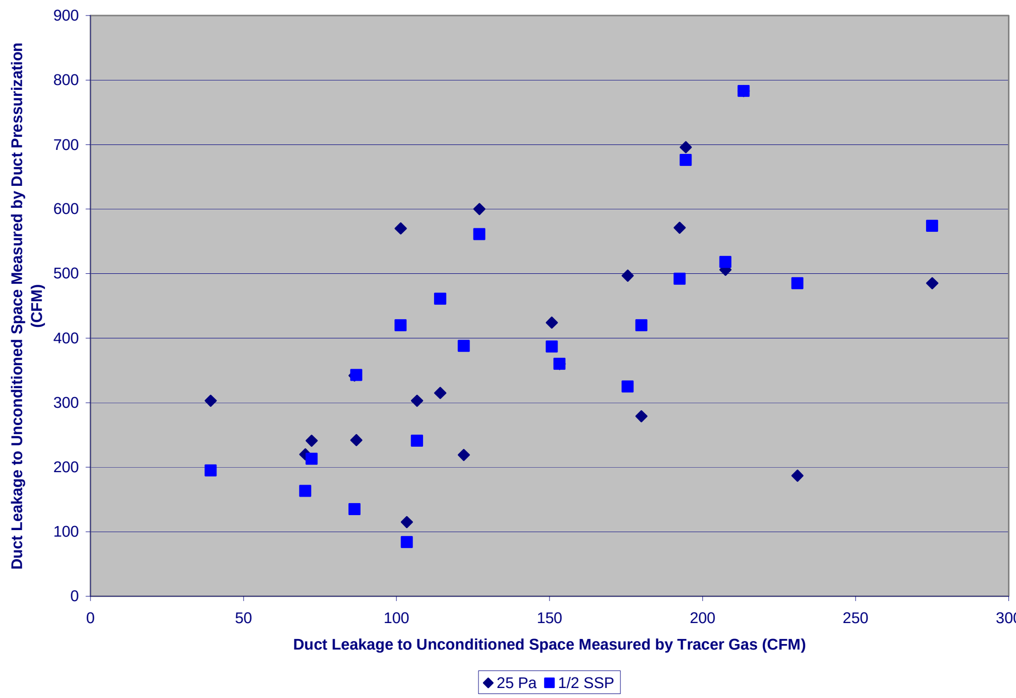

Figure 5-3. Duct Leakage to Unconditioned Space (CFM) As Measured with Different Methods

Table 5-2. Summary Statistics for Measurements of Duct Leakage to Unconditioned Space (n= 21)

Method	Measured CFM	
	Average	Standard Deviation
CO_2 Tracer Gas Infiltration Testing	143.0	61.5
Duct pressurization at 25 Pascal	393.2	179.9
Duct pressurization at ½ SSP	391.6	180.4

The correlation between the CFM of duct leakage to unconditioned space as measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.478; for duct pressurization at ½ SSP the correlation was 0.744. Measurements of duct leakage to unconditioned space made through the duct pressurization method at ½ SSP were more highly correlated with the tracer gas measurements than were measurements made at 25 Pa. These results suggest that the duct pressurization method at ½ SSP provides more accurate measurement of duct leakage to unconditioned space when using conventional measuring equipment.

Although measurements with the duct pressurization method made at ½ SSP are an improvement over measurements made at 25 Pa, additional studies are needed to determine if there is even a better predictor than using ½ SSP. Other ratios of system static pressure could be measured and the results compared to those from the tracer gas method. The modified SSP could be based on other on-sites measurements besides supply pressure and could include return plenum pressure and pressures at the registers.

5.2 BASELINE MEASUREMENTS OF DUCT LEAKAGE

Baseline measurements of total duct leakage and of duct leakage to unconditioned space were made for the sample of 109 sites for which air conditioning measurements were made. These baseline measurements were made with both duct pressurization methods (i.e., at 25 Pascals and at ½ SSP). The results of those measurements are reported in this section.

5.2.1 Baseline Measurements of Total Duct Leakage

Table 5-3 summarizes the baseline measurements of total duct leakage that were made using the two duct pressurization methods. The overall average for total duct leakage is similar for the two methods. However, the number of sites where measurements at 25 Pa could be made was smaller than the number where measurements at ½ SSP could be made.

Table 5-3. Summary Statistics for Baseline Measurements of Total Duct Leakage

Size of AC Unit (Tons)	Number of Houses Measured	Total Duct Leakage		
		Average (CFM)	Standard Deviation (CFM)	CFM per Ton
		Measurements Made at 25 Pa		
2.0	2	291.0	65.1	145.5
2.5	5	353.8	167.0	141.5
3.0	26	371.8	149.9	123.9
3.5	12	420.9	156.6	120.3
4.0	30	480.2	179.4	120.0
5.0	17	482.7	168.1	96.5
All	92	431.3	169.4	
		Measurements Made at ½ SSP		
2.0	2	165.5	46.0	82.85
2.5	5	323.0	191.4	129.2
3.0	28	369.8	198.3	123.3
3.5	16	378.6	197.3	108.2
4.0	36	499.4	207.6	124.9
5.0	21	490.6	140.7	98.1
All	108	431.8	200.2	

5.2.2 Baseline Measurements of Duct Leakage to Unconditioned Space

Table 5-4 summarizes the baseline measurements of duct leakage to unconditioned space that were made using the two duct pressurization methods. The overall average for duct leakage to unconditioned space is similar for the two methods. However, the number of sites where measurements at 25 Pa could be made was smaller than the number where measurements at ½ SSP could be made.

Table 5-4. Summary Statistics for Baseline Measurements of Duct Leakage to Unconditioned Space

Size of AC Unit (Tons)	Number of Houses Measured	Total Duct Leakage		
		Average (CFM)	Standard Deviation (CFM)	CFM per Ton
Measurements Made at 25 Pa				
2.0	2	179.5	10.6	89.8
2.5	5	247.8	195.6	99.1
3.0	25	289.3	154.0	96.4
3.5	10	270.6	156.6	77.3
4.0	27	367.4	182.7	91.8
5.0	15	323.3	141.4	64.7
All	84	313.2	165.0	
Measurements Made at ½ SSP				
2.0	1	157.0	N/A	78.5
2.5	5	190.8	128.0	76.3
3.0	27	279.6	167.4	93.2
3.5	13	253.3	135.5	72.4
4.0	36	383.5	185.7	95.9
5.0	21	349.0	146.5	69.8
All	103	321.2	172.2	

5.3 BEFORE AND AFTER MEASUREMENTS OF DUCT LEAKAGE

Measurements of total duct leakage and of duct leakage to unconditioned space before and after a servicing call from an HVAC contractor were made for a sample of units. These before- and after-servicing measurements were made with both duct pressurization methods (i.e., at 25 Pascals and at ½ SSP). The results of those measurements are reported in this section.

5.3.3 Before- and After-Servicing Measurements of Total Duct Leakage

Table 5-5 reports the results of a paired t-test that was performed on the before- and after-servicing data for total duct leakage. For both methods of measurement, the average total duct leakage for the units decreased about 12 percent from before servicing to after servicing. However, the results of the paired 2-tail t-tests show that the hypothesis of no difference between the before- and after-servicing averages can be rejected only with a confidence level of 80%.

Table 5-5. Results of Paired t-test on Change in Total Duct Leakage Calculated from Before- and After-Servicing Data

	Before-Servicing CFM	After-Servicing CFM

Measurements Made at 25 PA		
Mean	447.0	393.2
Standard Deviation	151.6	258.5
Observations	35	35
Pearson Correlation	0.394	
Hypothesized Mean Difference	0	
Degrees of freedom	34	
t Stat	1.311	
P(T<=t) one-tail	0.099	
t Critical one-tail	1.691	
P(T<=t) two-tail	0.199	
t Critical two-tail	2.032	
Measurements Made at ½ SSP		
Mean	433.4	381.5
Standard Deviation	209.1	289.1
Observations	43	43
Pearson Correlation	0.572	
Hypothesized Mean Difference	0	
Degrees of freedom	42	
t Stat	1.411	
P(T<=t) one-tail	0.083	
t Critical one-tail	1.682	
P(T<=t) two-tail	0.166	
t Critical two-tail	2.018	

5.3.4 Before- and After-Servicing Measurements of Duct Leakage to Unconditioned Space

Table 5-6 reports the results of a paired t-test that was performed on the before- and after-servicing data for duct leakage to unconditioned space. For both methods of measurement, the average CFM for duct leakage to unconditioned space decreased just under 30 percent from before servicing to after servicing. Moreover, the results of the paired 2-tail t-tests show that the hypothesis of no difference between the before- and after-servicing averages can be rejected with a confidence level over 99%.

Table 5-6. Results of Paired t-test on Change in Duct Leakage to Unconditioned Space Calculated from Before- and After-Servicing Data

	Before-Servicing CFM	***After-Servicing CFM***
Measurements Made at 25 PA		
Mean	309.1	220.4
Standard Deviation	130.3	126.4
Observations	30	30
Pearson Correlation	0.081	
Hypothesized Mean Difference	0	
Degrees of freedom	29	
t Stat	2.789	
P(T<=t) one-tail	0.005	
t Critical one-tail	1.699	
P(T<=t) two-tail	0.009	
t Critical two-tail	2.045	
Measurements Made at ½ SSP		
Mean	289.0	211.8
Standard Deviation	162.8	171.2
Observations	42	42
Pearson Correlation	0.448	
Hypothesized Mean Difference	0	
Degrees of freedom	41	
t Stat	2.847	
P(T<=t) one-tail	0.003	
t Critical one-tail	1.683	
P(T<=t) two-tail	0.007	
t Critical two-tail	2.020	

5.4 KWH SAVINGS FROM REDUCING DUCT LEAKAGE

From Table 5-4, there were 103 houses where duct leakage to unconditioned space was measured at ½ SSP. The weighted average size of the air conditioning units for these houses was 3.79 tons. At a nominal 400 cfm per ton the expected system airflow is 1,516 cfm. The duct leakage to unconditioned space for measurements at ½ SSP went from 289.0 cfm for baseline to 211.8 cfm for post-servicing. The effective system airflow to the space increased from 1,227 cfm to 1,304 cfm, an increase of 6.3%. This improved airflow implies an annual kWh savings of 82.1 kWh per AC unit that results from duct repair.

5.5 KWH SAVINGS FROM REFRIGERANT TUNE-UP AND REDUCING DUCT LEAKAGE

Estimates of the kWh savings from refrigerant tune-ups were presented in Section 4.4.5, while Section 5.4 presented estimates of savings from duct repairs. However, if households received both types of servicing, there will be interactive savings effects.

Out of 46 houses for which tune-up diagnostics and duct leakage measurements were made, there were 23 units (50%) that indeed received both charge adjustment and duct repair service.[5]

The combined kWh savings for refrigerant charge adjustment (6.1%) and duct repair (6.3%) is 12.0%. With an average kWh usage of 1,303 kWh for air conditionings, this implies an annual kWh savings of 156.4 kWh per AC unit that results from both services.

[5] Of the 46 units, there were 12 units (26%) that received charge adjustment only, 9 units (20%) that received only duct repair, and 2 units (4%) that received neither service.

6. CONCLUSIONS

Under this research project, ADM Associates, Inc. conducted a field performance assessment of residential packaged air conditioning units. Using a pre-defined set of diagnostic procedures, ADM field staff made in-field measurements of HVAC units' performance. These measurements were then used to diagnose faults in the operation of the units. In addition, field staff measured leakage from the air conditioning duct systems; both total duct leakage and duct leakage to unconditioned space were measured and analyzed.

Baseline measurements were made on 109 units to diagnose any faults with the units. Out of the 109 units tested, 89 were diagnosed as having some fault condition. The primary fault conditions were associated with charge level and air flow level.

There were 43 units that received servicing from an HVAC contractor for which ADM field staff took pre-servicing and post-servicing measurements. The pre- and post-servicing measurements were used to analyze changes in the EERs for the units at standard conditions before and after the servicing. The average EER for the units increased from 6.64 before servicing to 7.05 after servicing, an increase of about 6.1%. However, the results of a paired t-test showed that the hypothesis of no difference between the pre- and post-servicing averages can be rejected only with a confidence level of 80%.

There are some areas where further research on diagnostic testing could be conducted. The sensitivity of the measurement points should be evaluated prior to selection of the field measurement protocols in order to minimize field measurement points. Due to the turbulent environment in the ducting of packaged rooftop units, the measurement of airflow is an area that could benefit from additional research. After a set of protocols with consistent and repeatable results under a single set of conditions has been developed, these must be tested under varying conditions in order to refine the normalization of EER to standard conditions. Whether or not the EER normalization factor is consistent across unit types should also be assessed. Testing of units under the same conditions for pre and post servicing will give a more reliable measure of performance improvement due to servicing. These efforts would provide more refined diagnostic procedures for application to future studies of large numbers of units.

For some of the houses, measurements were also made of total duct leakage and of duct leakage to unconditioned space. Conventional practice in measuring duct leakage is to use a duct pressurization test, usually with a standard reference positive air pressure of 25 Pascals. However, some studies have suggested that duct leakage measured at 25 Pascals may be overstating actual leakage. To examine this question, measurements of duct leakage were made at a sample of 21 existing houses using three methods of measurement.

- Two of the methods for making the duct leakage measurements were variants of the usual duct pressurization method.

- One set of duct pressurization measurements was made using the standard fixed 25 pascals (Pa) pressurization.
- A second set of duct pressurization measurements was made by taking measurements at ½ system static pressure (SSP) for central air conditioning systems. SSP is a measurement of static pressure at the supply side plenum of the duct system when the supply fan is on and operating with registers in their normal position. This pressure is unique for each system. The rationale for using a modified SSP is to replicate the conditions that produce the weighted average pressure that the duct sees at the leakage locations.

- Tracer gas infiltration testing, which is regarded as one of the more accurate methods for measuring infiltration rates, was used as a third method of measurement to provide benchmark values for duct leakage against which measurement results from the duct pressurization methods could be compared and assessed. Carbon dioxide (CO_2) was used as the tracer gas for this testing.

For this comparison of measurement methods, duct leakage measurements were made with all three methods for 21 houses. The results were as follows.

- The correlation between the total duct leakage CFM measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.313; for duct pressurization at ½ SSP the correlation was 0.397.
- The correlation between the CFM of duct leakage to unconditioned space as measured with the CO_2 tracer gas method and with duct pressurization at 25 Pascal was 0.478; for duct pressurization at ½ SSP the correlation was 0.744. Measurements of duct leakage to unconditioned space made through the duct pressurization method at ½ SSP were more highly correlated with the tracer gas measurements than were measurements made at 25 Pa.

The results from the comparison of duct leakage measurement methods suggests that additional research might include conducting more testing of the different methods.

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APPENDIX A. SATURATION TEMPERATURE OF REFRIGERANT

For a given measured pressure and the system fluid (refrigerant) the saturation temperature is looked-up from the property table (Table A-1) below.

Summary Table A-1. Pressure – Temperature Chart

PSIG	TEMPERATURE °F						
	Yellow	Green	Green	Blue	Purple	Teal	White
	REFRIGERANT – (SPORLAN CODE)						
	12 (F)	22 (V)	124 (M)	134a (J)	502 ®	A750 or 507 (P)	717 (A)
5*	-29	-48	3	-22	-57	-60	-34
4*	-28	-47	4	-21	-55	-58	-33
3*	-26	-45	6	-19	-54	-57	-32
2*	-25	-44	7	-18	-52	-55	-30
1*	-23	-43	9	-16	-51	-54	-29
0	-22	-41	10	-15	-50	-53	-28
1	-19	-39	13	-12	-47	-50	-26
2	-16	-37	16	-10	-45	-48	-23
3	-14	-34	18	-8	-42	-46	-21
4	-11	-32	21	-5	-40	-44	-19
5	-9	-30	23	-3	-38	-41	-17
6	-7	-28	26	-1	-36	-39	-15
7	-4	-26	28	1	-34	-38	-13
8	-2	-24	30	3	-32	-36	-12
9	0	-22	32	5	-30	-34	-10
10	2	-20	34	7	-29	-32	-8
11	4	-19	36	8	-27	-31	-7
12	5	-17	38	10	-25	-29	-5
13	7	-15	40	12	-24	-27	-4
14	9	-14	41	13	-22	-26	-2
15	11	-12	43	15	-20	-24	-1
16	12	-11	45	16	-19	-23	1
17	14	-9	46	18	-18	-21	2
18	16	-8	48	19	-16	-20	3
19	17	-7	49	21	-15	-19	4
20	18	-5	51	22	-13	-17	6
21	20	-4	52	24	-12	-16	7
22	21	-3	54	25	-11	-15	8
23	23	-1	55	26	-9	-14	9
24	24	0	57	27	-8	-12	11
25	25	1	58	29	-7	-11	12
26	27	2	59	30	-6	-10	13
27	28	4	61	31	-5	-9	14
28	29	5	62	32	-3	-8	15
29	31	6	63	33	-2	-7	16
30	32	7	65	35	-1	-6	17
31	33	8	66	36	0	-4	18
32	34	9	67	37	1	-3	19
33	35	10	68	38	2	-2	19
34	37	11	69	39	3	-1	20
35	38	12	71	40	4	0	21
36	39	13	72	41	5	1	22
37	40	14	73	42	6	2	23
38	41	15	74	43	7	3	24
39	42	16	75	44	8	4	25
40	43	17	76	45	9	4	26
42	45	19	78	47	11	6	28
44	47	21	80	49	13	8	29
46	49	23	82	51	15	10	31
48	51	24	84	52	16	11	32
50	53	26	86	54	18	13	34

PSIG	TEMPERATURE °F						
	Yellow	Green	Green	Blue	Purple	Teal	White
	REFRIGERANT – (SPORLAN CODE)						
	12 (F)	22 (V)	124 (M)	134a (J)	502 ®	A750 or 507 (P)	717 (A)
60	62	34	95	62	26	21	41
62	64	35	97	64	27	22	42
64	65	37	98	65	29	24	44
66	67	38	100	66	30	25	45
68	68	40	101	68	32	26	46
70	70	41	103	69	33	28	47
72	71	42	104	71	34	29	49
74	73	44	106	72	36	30	50
76	74	45	107	73	37	32	51
78	76	46	109	75	38	33	52
80	77	48	110	76	40	34	53
85	81	51	114	79	43	37	56
90	84	54	117	82	46	40	58
95	87	56	120	86	49	43	61
100	90	59	123	88	51	45	63
105	93	62	126	90	54	48	66
110	96	64	129	93	57	51	68
115	99	67	132	96	59	53	70
120	102	69	135	98	62	55	73
125	104	72	138	100	64	58	75
130	107	74	140	103	67	60	77
135	109	76	143	105	69	62	79
140	112	78	146	107	71	64	81
145	114	81	148	109	73	66	82
150	117	83	150	112	75	68	84
155	119	85	152	114	77	70	86
160	121	87	154	116	80	72	88
165	123	89	157	118	82	74	90
170	126	91	159	120	83	76	91
175	128	92	161	122	85	78	93
180	130	94	163	123	87	80	95
185	132	96	165	125	89	82	96
190	134	98	167	127	91	83	98
195	136	100	169	129	93	85	99
200	138	101	171	131	95	87	101
205	140	103	173	132	96	88	102
210	142	105	175	134	98	90	104
220	145	108	178	137	101	93	107
230	149	111	182	140	105	96	109
240	152	114	185	143	108	99	112
250	156	117	188	146	111	102	115
260	159	120	192	149	114	105	117
275	163	124	196	153	118	109	121
290	168	128	201	157	122	113	124
305	172	132	205	161	126	117	128
320	177	136	209	165	130	120	131
336	181	139	213	169	133	124	134
350	185	143	217	172	137	127	137
365	188	146	221	176	140	130	140

PSIG	Yellow 12 (F)	Green 22 (V)	Green 124 (M)	Blue 134a (J)	Purple 502 ®	Teal A750 or 507 (P)	White 717 (A)
52	55	28	88	56	20	15	35
54	57	29	90	57	21	16	37
56	58	31	91	59	23	18	38
58	60	32	93	60	24	19	40

PSIG	Yellow 12 (F)	Green 22 (V)	Green 124 (M)	Blue 134a (J)	Purple 502 ®	Teal A750 or 507 (P)	White 717 (A)

Table headers: TEMPERATURE °F; REFRIGERANT – (SPORLAN CODE)

Source: Sporlan Valve Company, Form 1-301.

Air Conditioning Residential Performance & Duct Leakage Measurement Form — **SCE Project - 703**

Date:
Customer Name: ID #: SCE###
Address: City: State: CA
Measurement team: SCE Meter #

# of A/C units: # of Floors Conditioned Floor Area in Sq. Ft

A/C Unit Identifier: Location:

Inspection of Unit Data (Describe condition of unit and conditions of location)

On level pad	Clearance at condenser	Fused electric connections
Corrosion	Excess vibration/noise	Missing filters, panels, other
Dirty condenser	Bent condenser coil fins	Problems w/ refrigerant line tube
Dirty evaporator	Bent evaporator coil fins	Condensate drain pan condition
Dirty air filters	Belt tension	Cracks in belts
Odors around the unit (condenser/evaporator/drain pipe)		Appears in good condition

Type of unit: A/C only, Heat Pump, A/C w/ elec. Heat, A/C w/ gas heat, Other
Refrigerant: R-22, R-410A, Other, Amount (lbs)
Expansion Device Type: TXV, Feed Piston, Capillary Tube
Distance (feet) between Condenser and evaporator

Unit Name Plate Data: Readable, Non-readable
Make: Model:
Type: Std. Eff. Hi-Eff. Serial No.:

Compressor Type: Reciprocal, Scroll
Make: Model: Size:
Compressor: Volts Amps Hp:
Outdoor Fan: Volts Amps Hp:

Sensible Capacity (kBTU/hr): Latent Capacity (kBTU/hr):
Tons: COP or EER or SEER: Year / Age

Indoor Fan: Volts Amps Hp:
Make Model SN:

Other Notes:

Take Pictures of: 2 Full views of unit, Nameplate (if readable), Problems or Trouble spots

Southern California Edison Comprehensive Field Assesment Program for Packaged A/C Systems V5

		Air Stream conditions		
Test Condition :		Supply Air	CFM	**SACFM**
			DBTemp	**SATdb**
	Baseline		RelHumid	**SARH**
			WBTemp	**SATwb**
ID #	SCE###	Return Air	CFM	**RACFM**
Region	SCE Area		DBTemp	**RATdb**
Date			RelHumid	**RARH**
Time			WBTemp	**RATwb**
UnitAge,yr		Ambient	Initial DBT	**OATdbi**
Unit Size,ton		(Outdoor Air)	Final DBT	**OATdbf**
ExpDevice			RelHumid	**OARH**
Comp type			WBTemp	**OATwb**
HiEfficienUnit		condenser	air exhaust	**CA**
Refrigerant				
Outdoor and acceptable Test Conditions				
Minimum CFM Requirements				
Measured Performance Parameter Check				
The calculated EER of the Unit is			**or COP of**	
Measured Capacity of the unit in kBTU			**or in Ton**	
The measured EER of the unit under standard conditions would be			**or COP of**	

Refrigerant Side conditions			**Stage-1**	
High-side	Pressure	**DP**		
		LP		
	lineTemp	**LT**		
	satTemp	**CT**		
Low-side	Pressure	**SP**		
	lineTemp	**ST**		
	satTemp	**ET**		
Evaporator	Temp drop	**ETD**		
nTXV	Required SH	**SHr**		
	Measured SH	**SHm**		
TXV	Required SC	**SCr**		
	Measured SC	**SCm**		
Total Measured kW input				
Diagnostic Assessments				
		Stage 1	**Stage 1**	
Parameter	Target	Measured	**Diagnosis**	
ET	**40**			
SC	**15**			
SH	**#REF!**			
COA	**20**			
ETD	**20**			
CTD	**20**			
Selected Detectable Fault Status				
Inefficient Compressors				
Refrigerant Flow restriction				
Condenser fouling				
Evaporator fouling				
Charge Related Problem				
TXV problem				
Air Flow				

Blue Cells are for data entry
Red Cells are output or results

Field Assesment Data Input Form V5

SCE ### | Blue Cells are for data entry

Duct Leakage to Unconditioned Space

	Static Pressure, Pascals	Airflow Rate, cfm	Alternate Pressure	Calc. cfm
System Static Pressure, Pascals (supply plenum)				
Total Duct Leakage - at ½ SPP or ?				
Duct Leakage Rate to Unconditioned Space - at ½ SPP or ?				
Total Duct Leakage - at 25 Pascals				
Duct Leakage Rate to Unconditioned Space - at 25 Pascals				

Temperature and Humidity Measurements

					Calibrated		Average		Wet-Bulb
Location	**Point**	**Meter ID#**	**Temperature, °F**	**Rel. Humidity (%)**	**Temp, °F**	**RH (%)**	**Temp, °F**	**RH (%)**	**Temp, °F**
Initial Ambient	-								
Return Air	1								
Return Air	2								
Return Air	3								
Supply Air	1								
Supply Air	2								
Supply Air	3								
Air Off Cond. (CA)	-								
Final Ambient	-								

One-Time Electrical Measurements:

Conditions	Vac	Amps	kW	Power Factor
Outdoor Unit Power				
Air Handler Power				
Sum Total				

Refrigerant Side Measurements:

Location	Liquid Line	Low Side (Suction)	High Side (Discharge)	
Pressure (psig)				
Line Temperature(°F)				Clamp-on
Line Temperature(°F)				Bulb w/ insulation