

# Measure Guideline: Air Conditioner Diagnostics, Maintenance, and Replacement

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*Alliance for Residential Building Innovation (ARBI)*

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## **Measure Guideline: Air Conditioner Diagnostics, Maintenance, and Replacement**

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## Definitions

ACCA	Air Conditioning Contractors of America
AHRI	Air Conditioning, Heating, and Refrigeration Institute
ANSI	American National Standards Institute
ARBI	Alliance for Residential Building Innovation
ASHRAE	American Society for Heating, Refrigeration, and Air Conditioning Engineers
BPI	Building Performance Institute
CD	Comprehensive Diagnostic (a procedure as defined in this document)
CEC	California Energy Commission
CFC	Chlorofluorocarbon
CFM	Cubic feet per minute
COA	Condensing Over Ambient”, or the difference between the condensing temperature and the temperature of air entering the condensing coil
DEG	Davis Energy Group
ECM	Electronically commutated motor
EER	Energy Efficiency Ratio, in Btuh per watt
EPA	Environmental Protection Agency
EST	Refrigerant evaporation saturation temperature
Ft <sup>2</sup>	Square feet
HERS	Home Energy Rating System
HFC	Hydrofluorocarbon
HVAC	Heating, ventilation, and air conditioning
kBtuh	Thousand British Thermal Units per hour
kW	Kilowatt
kWh	Kilowatt-hour
MERV	Minimum efficiency reporting value (for air filters)
NSOP	Normal system operating pressure (for airflow measurement corrections)
PD	Preliminary Diagnostic (a procedure as defined in this document)
PG&E	Pacific Gas and Electric Company
PSC	Permanent split capacitor (motor type)
SCE	Southern California Edison
SEER	Seasonal energy efficiency ratio



SHR	Sensible heat ratio
TD	Temperature difference
TSOP	Test system operating pressure (for airflow measurement corrections)
TXV	Thermostatic expansion valve
W	Watt
w.c	Water column (referring to inches of pressure)

## Executive Summary

Heating and cooling energy use represents about 54% of all energy consumed in existing residential buildings. Field studies have shown that more than half of installed air conditioning (AC) systems have significant defects, and that proper maintenance of cooling systems can reduce their energy use by as much as 50%. Since the 1990s, an immense body of knowledge on the importance of proper maintenance of heating, ventilation, and air conditioning (HVAC) systems and the impact of defects has been accumulated, and research in this area is continuing. Though some of this information has found its way into standards and utility programs, general maintenance practices continue to miss details that could realize significant energy savings at minimal cost. This guideline targets home performance contractors who are relatively inexperienced with HVAC systems, HVAC technicians who can benefit from recent research on AC system faults and diagnostic methods, and program managers who may apply this information in structuring successful HVAC tune-up programs.

Home performance contractors who are not attentive to air conditioner performance are missing the opportunity to deliver optimal energy savings to their customers. Systems that are not in obvious need of replacement may still perform far below their rated conditions. Although these systems are complicated, many faults that substantially degrade performance can be easily identified and in some cases corrected by the home performance contractor during the normal course of completing home inspections and duct testing.

Little information is available on methods to diagnose systems and particularly to detect and discriminate between multiple faults such as inadequate airflow, incorrect charge, liquid line restrictions, evaporator and condenser coil fouling, and the presence of non-condensable refrigerant contaminants. While there has been some progress with development of computer diagnostic tools for use in utility programs, a systematic method is needed to detect and identify major system faults by field technicians who do not have access to these tools.

This guideline provides information on the impact of various defects and the potential energy savings that can result from proper maintenance, and describes a two-step process for diagnosing and correcting faults. The first step involves a fundamental inspection and tests that can be completed by home performance contractors with BPI or similar training to identify or diagnose, and in some cases remediate, relatively non-technical problems such as low airflow and fouled condenser coils. The second step is designed to be implemented by experienced HVAC technicians and provides a systematic approach that can be used to identify system faults.

## Acknowledgements

Davis Energy Group (DEG) would like to acknowledge the U.S. Department of Energy Building America program and their funding and support of development of this guideline as well as research that informed it. The detailed diagnostic method described in this guideline was adapted from Appendix RA3 of the California Title 24 Residential Energy Efficiency Standards (CEC 2008). Substantial guidance with the development of diagnostic methods was provided by Jim Phillips of The Energy Savers. Also, DEG would like to acknowledge Marshall Hunt of Pacific Gas & Electric Company for the direction he provided in developing the two-step approach.

## Introduction

Space cooling and heating constitutes 54% of residential energy use nationwide (DOE 2010), and HVAC tune-ups are a potentially very cost-effective component of residential retrofits. Laboratory testing suggests that service and replacement can produce HVAC energy savings on the order of 30% to 50% (Messenger 2008). A recent sampling of more than 1,300 air conditioning (A/C) units from the Arkansas CoolSaver Program revealed that corrections in charge and airflow netted an average increase in delivered Btu/h of 38% and a peak kW demand reduction of 0.5 kW per tune-up (Kuonen 2011). However, as expressed in Kuonen's Home Energy article, "Simply put, the service practices we performed in the past to get the job done are not getting the job done today. Past practice and best practice no longer belong in the same sentence." Other programs have verified the need for air conditioner maintenance. Downey (2002) reported that of 8,873 systems tested, 65% required repairs. Mowris (2004) reported field measurements of 4,168 air conditioners where 72% of systems had improper refrigerant charge and 44% had improper airflow.

These faults are widespread due to a lack of standards, training, information, and other market barriers regarding proper evacuation and maintenance procedures. Mowris (2011a) surmises that many systems are undercharged due to long line sets where no additional refrigerant is added or where systems have leaking refrigerant over time. However, there is the possibility that low charge diagnoses may be caused by a false diagnosis of undercharge due to refrigerant restrictions, lower ambient temperatures, inaccurate analog pressure sensors, or inaccurate refrigerant line temperature sensors. For example, a system with the correct charge when loaded at high ambient temperatures (i.e., 95°F or higher) may produce a false undercharge indication when checked at lower ambient temperatures, or a false undercharge indication can be caused by inaccurate temperature sensors producing higher suction line or lower liquid line temperatures. Superheat tables available from manufacturers (Carrier 1995) or the California Energy Commission (2008) account for changes in outdoor temperature, but these tables may not predict the proper required superheat for all systems without thermostatic expansion valves (TXV) at low ambient temperatures. False diagnosis of undercharge will cause a decrease in system efficiency if refrigerant is unnecessarily added. The lack of information and training on correct procedures can result in an unintentional degradation of system performance. Incorrect charge procedures and inaccurate or badly maintained service equipment can turn a mediocre system into a poorly performing system.

This measure guideline attempts to bridge the gap between potential and realized whole house energy savings by developing field guidelines for HVAC diagnosis, repair, and equipment replacement. The intent of this document is to provide easy-to-implement, cost-effective, and reliable methods for home performance contractors and technicians to follow for the diagnosis and correction of AC system problems, particularly during the course of the energy upgrade process. It expands on ACCA Standard 4 (Maintenance of HVAC Systems) and Standard 5 (HVAC Quality Installation Specification) as they pertain to residential AC and heat pump systems, and offers a way to capture a source of energy savings that is frequently ignored in the course of completing retrofits.

This guideline primarily addresses diagnosis and repair of refrigerant systems and mechanical components but excludes information on ducting, which is addressed by another Building America measure guideline, *Building America Measure Guideline: Sealing and Insulating Ducts in Existing Homes* (Aldrich 2011). Minimizing duct leakage is critical to maintaining adequate system airflow. The diagnostic tests described in this guideline rely on the completion of duct testing and sealing prior to initiation of tests. An acceptable standard for duct tightness in existing homes is 15% of cooling system airflow at a duct pressurization of 25 Pa (CEC 2008).

Refrigerant system problems can be difficult to diagnose. The art of HVAC maintenance will continue to evolve as simpler and more accurate techniques for evaluating refrigerant charge and other diagnostic methods are developed. For example, one program implementer is recommending further research to develop a diagnostic protocol for contractors and HERS verifiers to differentiate the presence of non-condensables (air and water vapor) from a refrigerant overcharge or blocked condenser or evaporator coil or the presence of refrigerant restrictions from a refrigerant undercharge or low airflow condition (Mowris 2011a). Though there are differing opinions on diagnostics and repair procedures, the methods included in this guideline are intended to represent the current state of the art. Updates will be required as diagnostic procedures are improved and new equipment and tools emerge. Using this guideline, major problems can be identified and in some cases remedied at low cost, resulting in lower heating and cooling energy bills and improved comfort and safety for the homeowner.

In order to accommodate different levels of training and experience as well as time and cost constraints, two approaches are described. One is designed to rapidly identify and resolve major problems, and the other involves more detailed testing, diagnosis, and remediation that will help to insure that equipment is operating at its optimal performance.

Air conditioner tune-ups and repair can involve exposure to high-voltage electricity and other risks; work involving such exposures should be conducted by trained service technicians familiar with standard safety practices.

# 1 Section 1: Planning and Decision Making Criteria

## 1.1 Defining the Problem

Poor performance resulting from aging systems, partial equipment failure, poor original design and workmanship, improper service, and lack of maintenance is commonplace. These problems result in increased energy use and utility bills, reduced occupant comfort, accelerated equipment degradation, and in some cases risk to homeowners. These problems can be interrelated and difficult to correct without applying a systematic approach to diagnosis and maintenance. Deficiencies frequently found in existing homes are listed and categorized in Figure 1. In addition to these deficiencies, contractors should be cognizant of the effects that improvements to the building enclosure and ducting will have on equipment sizing so as to avoid performance degradation resulting from oversizing and excessive equipment cycling.

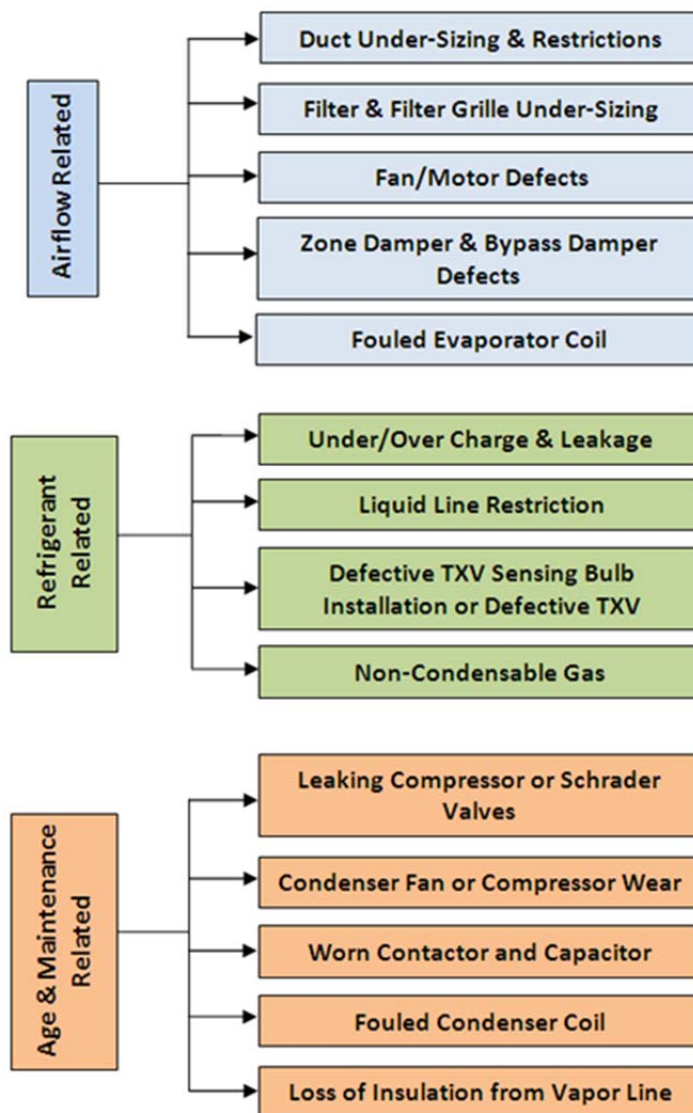


Figure 1. Primary system fault modes.

A key challenge to understanding refrigeration system performance is that there are many possible sources of degradation, and there is rarely an obvious indication which part (or often, parts) of the system is failing. Furthermore, system performance characteristics encountered in the field depend on ambient conditions and the fault status. In spite of the complexity of multiple faults and highly variable performance, HVAC technicians providing service calls are expected to improve system efficiency, often within a short scheduled time window. A systematic approach is needed to efficiently identify, diagnose and correct defects.

The complexity of the HVAC maintenance problem was explored by a recent Southern California Edison (SCE) sponsored project. Responding to lower than expected results from an HVAC maintenance program, the project team reviewed prior studies on the impact of refrigerant charge and airflow on system performance and to what extent measurement uncertainty can lead to lower than expected savings (Hunt 2010). Some conclusions were:

- Though several studies of field performance have been completed, field performance is still not as well understood as laboratory performance due to occupant behavior, service frequency and quality, and other environmental factors that are difficult to control.
- There is a general lack of diagnostic methods that ensure that the service providers will address all the problems that might exist, and that applies improvements in the correct order.
- Inaccuracies in field measurements are common, and with human factors, they have a compounding effect that can result in high levels of uncertainty in energy savings estimates. With improper methods and tools, it is difficult for a simple refrigerant charge and airflow diagnostic adjustment to be accurately performed.

## **1.2 Maintenance Approaches**

The increased focus on the energy performance of existing homes creates an opportunity to, at a minimum, identify whether a problem with AC systems exists so that the decision to make corrections can be offered to homeowners. However, most home performance contractors lack the information and training to recognize even serious problems that affect performance.

Responding to this problem, this guideline provides a two-step approach to diagnosis and repair:

- A preliminary diagnostic (PD) approach designed for home performance contractors to quickly identify system problems that can be either immediately corrected or referred to qualified HVAC technicians
- A comprehensive diagnostic (CD) approach designed for skilled HVAC technicians that ensures that systems are operating at their maximum potential.

It may be unnecessary to involve a skilled HVAC technician if the major problems are obvious and simple to resolve, such as a clogged filter or fouled condenser coil. Results of diagnostics completed by a relatively unskilled person can be referred to an HVAC technician with the skills and equipment to complete a thorough diagnosis and maintenance process to ensure systems are operating to their maximum potential. This approach avoids duplication of effort and minimizes costs. Major problems can be identified and remedied, leaving more extensive tune-ups to be completed by skilled technicians at the time of retrofit, or later as budgets and circumstances allow.

The cost to diagnose and remediate HVAC system problems is as variable as the wide range of problems seen in the field. Basic testing and diagnosis using the PD approach could be completed in 30 to 60 minutes and could cost under \$100, depending on the location and other factors (Means 2011). The identification of significant problems could lead to the need to replace the outdoor unit and indoor unit coil, which could range from under \$1,500 to over \$10,000, depending on the size and efficiency of the replacement equipment (according to the National Residential Efficiency Measures Database).

### **1.3 Energy Savings**

Just as the range of problems encountered is extremely wide, benefits accruing from maintenance are highly variable. Efforts to verify energy savings from California HVAC maintenance programs have been confounded by problems with measurement (Hunt, 2010). However, the finding of 38% energy savings for airflow and refrigerant maintenance reported for the Arkansas CoolSaver Program is encouraging (Kuonen 2011). Researchers have been attempting to find ways to use diagnostics to predict energy savings, but since multiple defects can produce similar measurable effects, specific defects cannot be identified without initiating corrective actions (Braun 2011).

The potential for energy savings through maintenance has been identified in laboratory testing that measures performance as a function of airflow, charge condition, and other parameters. Recent laboratory testing compared the effects of hot attic environment, line-set length, refrigerant charge, TXV devices, airflow, duct leakage, evaporator coil blockage, condenser coil blockage, refrigerant restrictions, measurement equipment, and non-condensables on air conditioner performance (Mowris 2012). Baseline comparisons were made to a 3-ton unit tested using ANSI/AHRI Standard 210/240 test methods. Tests showed a reduction of the “application SEER”<sup>1</sup> by 67% for a poorly installed unit with ducts installed in a hot attic with 30% duct leakage, 300 cfm per ton airflow, 10% refrigerant undercharge, and 50% condenser coil blockage. Moderate to severe non-condensables (0.3 to 1% by weight) were found to reduce the application EER and SEER by 12% to 38% and refrigerant restrictions by 30% to 59%.

### **1.4 The Consequences of No Action**

If, in the course of performing home energy retrofits HVAC maintenance is ignored, a significant energy savings opportunity will be lost. In addition to unrealized energy savings, the consequences can include:

- Reduced comfort due to inadequate airflow, poor airflow distribution, or refrigerant system and other defects that result in diminished capacity
- Poor humidity control due to faulty equipment and/or equipment short-cycling resulting from restrictions or post retrofit over-sizing
- Occupant health hazards of mold growth due to plugged condensate drains
- Low equipment reliability due to faulty components such as contactors, capacitors, condenser fan motors, and compressors

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<sup>1</sup> An “application SEER” is not a formal SEER rating but is developed using the same ANSI/ASHRAE Standard 201/240-2008 test methods used for developing official ratings. Hereafter, informal performance metrics are denoted by an asterisk.

- Electric shock hazards because of failure to identify incorrect wiring and equipment grounding
- Shortened equipment life resulting from coil freezing from low refrigerant charge or restrictions, defective TXV, improper refrigerant charge, and non-condensable air and water vapor or other contaminants in the refrigerant lines.

### **1.5 Planning for Improvements and Risk Mitigation**

Except where systems have failed catastrophically, the condition of systems cannot be determined until basic diagnostic procedures are implemented. This causes difficulty in planning for improvements and uncertainty in anticipating energy savings. For the home performance contractor, failure to conduct simple diagnostics may impose a higher risk than taking the time to investigate and at least, solve the major deficiencies. In the course of inspecting systems for performance-related problems, risks such as faulty wiring, plugged condensate drains, and imminent equipment failure can be brought to light.

There is also a market-related risk to taking no action. If homeowner expectations of energy savings and comfort are not met, news of dissatisfaction may infect the regional market for home energy upgrades. Interviewing the homeowner to determine the maintenance history of the system and to identify any comfort issues prior to proceeding with diagnostics can provide clues to possible system or design deficiencies. The two-stage approach described in the Implementation section of this document is designed to identify obvious problems with a small investment of time, and provides for efficient diagnosis and remediation by skilled HVAC technicians when significant problems are found. This approach will increase the likelihood that attention will be given to minimizing risks in HVAC systems.



## 2 Section 2: Technical Description

### 2.1 General Performance Interactions

In addition to their role in maintaining comfortable indoor temperatures, heating and cooling systems are responsible for moisture removal (latent cooling) and maintaining indoor air quality through the removal of particulates and distribution of ventilation air. If ducts are leaky and/or if there are inadequate provisions for return air, they can induce infiltration, thereby adding to the heating and cooling load. As such, defects can inhibit the ability of HVAC systems to maintain comfort, indoor relative humidity, and air quality. Poor humidity control can result in mold formation and even structural damage. The consequences of defects related to airflow and refrigerant charge are discussed in the following sections.

### 2.2 Airflow Impacts

#### *Desired Airflow*

Air conditioners are typically designed to have 350 to 400 cfm of air flowing across the indoor coil for each ton of cooling capacity. For humid climates, 350 cfm per ton is recommended to increased moisture removal. In hot-dry climates where latent loads are minimal, 500 cfm per ton is recommended (CEC 2011). As pointed out by Parker (1997), correct airflow achieves the desired balance between sensible heat transfer and moisture removal. ACCA Manual S (1995) prescribes a method for determining airflow based on the sensible heat ratio (SHR). For example, at SHRs below 0.80, an airflow is selected to achieve a supply air temperature that is 21°F below the return air temperature, whereas at SHRs above 0.85, this temperature split is reduced to 17° by increasing the airflow. The Manual S equation for calculating airflow is:

$$CFM = \text{Sensible Load} / (1.1 \times TD) \quad \text{Equation 1}$$

Where: *Sensible Load (Btuh)* = load calculated using ACCA Manual J  
*TD (°F)* = temperature difference across the cooling coil

#### *What Affects Airflow?*

In the field, system airflow is affected by the furnace or air handler equipment used, motor speed settings, installed duct system, air filter, and the condition of the air distribution system.

Restrictions in the air distribution system have a direct impact on airflow and include:

- Duct undersizing and excessive length, turns, fittings, kinks, and obstructed grilles
- Blower motor type
- Filter undersizing, design and type of filter, and/or filter blockage (dirt or debris)
- Cooling coil restrictions due to sizing and/or dirt or debris accumulation
- Zoned systems with incorrectly sized ducts.

**Duct Constrictions.** It is not uncommon for flex ducting to be constricted by hangers or crimped by tight bends. Section 3 (Implementation) provides a method for quickly verifying issues with supply duct sizing or restrictions that result in substantial pressure losses. The method can also detect excessive pressure loss at cooling coils, filters, and other points in the system.



**Figure 2. Examples of constricted ducts (Photo credit: R. Chitwood, reprinted with permission).**

**Blower Reaction to Increased Pressure.** Many existing systems operate under static pressure conditions that are higher than anticipated by manufacturers, resulting in decreased airflow (typical for blowers with PSC motors) or high fan watt draw (typical of ECM-driven blowers). Reduced airflow degrades air conditioner performance, which increases compressor energy use and decreases cooling capacity, and decreases the SHR of the equipment. Nearly all furnaces and air handlers installed before 1990 use constant RPM PSC motors that deliver less air as the pressure differential across them increases. However, many high efficiency furnaces and heat pump air handlers use variable speed ECM motors that vary RPM and torque to maintain a relatively constant airflow under varying static pressure conditions, but do it at the expense of increased fan power.

**Filter Pressure Loss.** Most existing HVAC systems have filter grilles or furnace filter slots that were sized for fiberglass filters. The tendency is to replace them with the newer “high minimum efficiency reporting value (MERV)” filters that are much more effective at removing small particles such as pollen. While filters with MERV ratings of 6 and higher improve indoor air quality, they also impose a much greater restriction to airflow, and result in either a decrease in airflow (with PSC blower motors) or increased power consumption (with variable speed blower motors),

### Motor Types and Characteristics

*Permanent split capacitor (PSC) motors are the most common type used in furnace and air handlers and operate at a relatively fixed RPM. As the static pressure increases they deliver less air and the power falls off slightly. Their efficiency is in the 55-70% range.*

*Electronically commutated motors (ECMs), also called brushless permanent magnet (BPM) motors, are used in higher efficiency systems. The variable speed variety has the ability to vary RPM and torque to maintain a constant airflow as static pressures increase. Others maintain a constant RPM and respond to static pressure in a similar fashion as PSC motors. The efficiency of ECM's is in the 70-80% range.*

### Calculating Filter Velocity

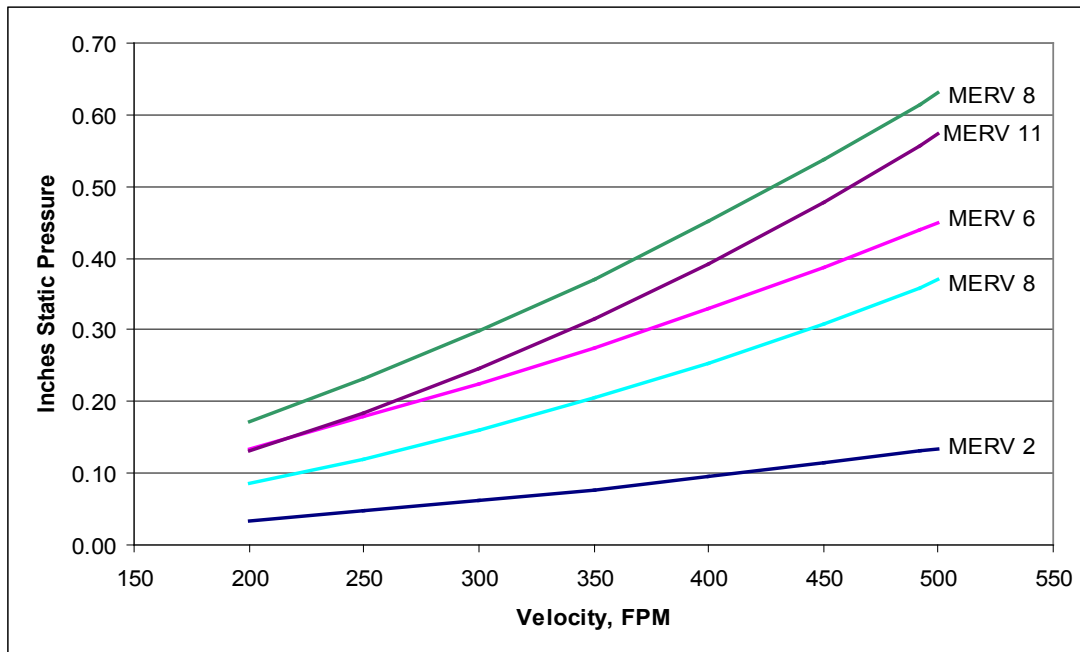
*If filter velocity exceeds 300 fpm the filter is almost certain to reduce airflow to unacceptable levels when used with high efficiency filters. To check this, the equation below can be used to calculate the filter velocity for a given system:*

$$\text{Filter Velocity} = \text{Tons} \times 57,600 / (L \times W)$$

where:

*Tons = air conditioner rated tons  
L = filter length in inches  
W = filter width in inches*

or both. Figure 3 shows the results of tests performed to determine the pressure loss for five commonly available filters (Springer 2009). The MERV 2 filter is a conventional 1 in. fiberglass type and the others are 1 in. pleated filters. Note that pressure loss does not always correspond to MERV rating, as two of the MERV 8 filters have quite different pressure drop characteristics. Unfortunately, manufacturers do not provide the pressure loss value filter on packaging, but as a general rule, velocity should be limited to 300 fpm or less to avoid excessive pressure loss.



**Figure 3. Pressure loss as a function of velocity for commonly available filters.**

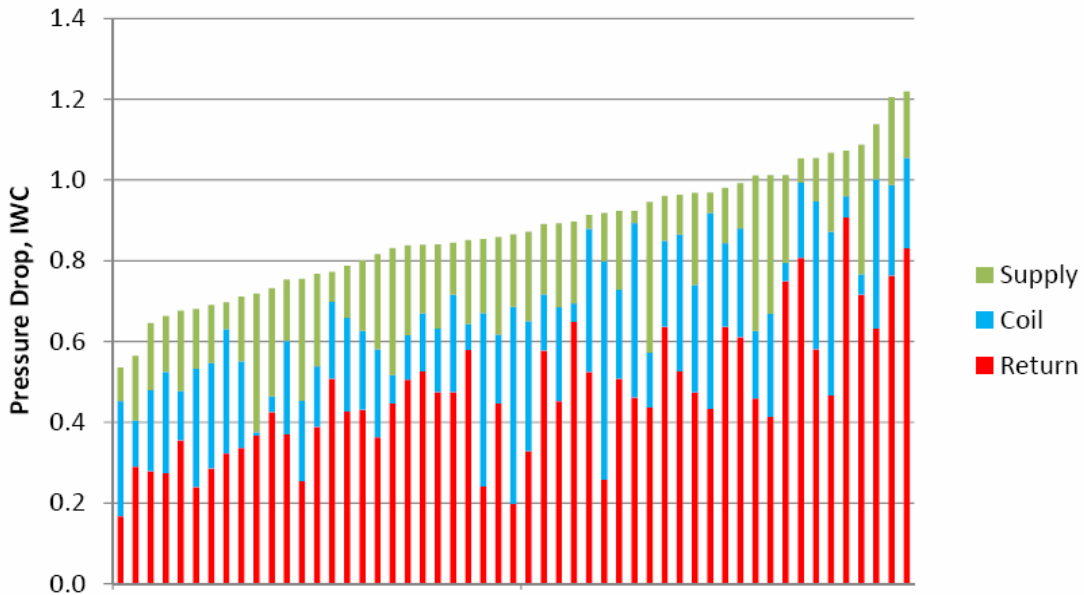
**Zone Dampers.** As the thermal integrity of building enclosures improves, it is becoming more common for contractors to install zone dampers instead of multiple systems, particularly for two-story homes. When zone dampers close, the air velocity in ducts serving the remaining zones increases, which increases overall system pressure drop and reduces the volume of air delivered. Many zoned systems utilize bypass dampers to maintain airflow across the cooling coil. When air bypasses the return-supply grilles, it results in a lower coil entering air temperature, which has a similar effect on system performance as reduced airflow. Zoned distribution systems deserve particular attention and may require modification before air conditioner charge can be properly set. Improvements could include eliminating the bypass damper and installing additional ducting to supply excess air to central living spaces.

**Duct Leakage.** It is important that ducts be tested for leaks and sealed before any other diagnostics are initiated since leakage in the return ducting will affect airflow measured at the return grille. Depending on the method used, duct leakage can result in false airflow measurements and also leads to incorrect temperature split measurements.

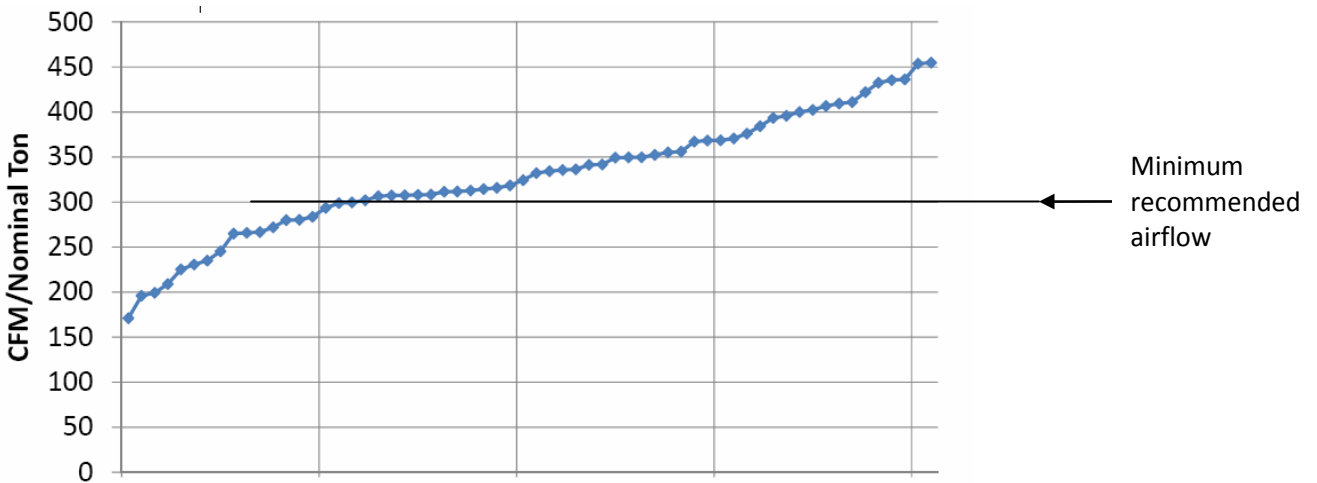
### ***Findings from the Field***

Data from California field studies suggest that system restrictions are prevalent and have a significant impact on airflow. Figures 4 and 5 display results from tests of 62 California homes built around 2007 (Wilcox 2011). Pressure measurements showed the median static pressure to

be greater than the typical manufacturer’s maximum recommended value of 0.8 in. w.c. (Figure 4). Filters were responsible for the majority of return side pressure losses, averaging 0.28 in. w.c. ACCA Manual D suggests that a standard clean furnace filter will have a pressure drop of about 0.10 in. w.c. Referring to Figure 3, achieving 0.10 in. with the best MERV 8 filter would require a velocity of about 220 fpm. For a 2-ton system at 800 cfm, the filter size would need to be about 20 in. x 25 in. to maintain this velocity.



**Figure 4. Sources of pressure drop measured in 62 California homes (Source: Wilcox 2011, reprinted with permission).**



**Figure 5. Summary of airflow measurements from 62 California homes (Source: Wilcox, reprinted with permission).**

Figure 5 confirms the connection between high pressure drop and depressed airflow rates. About two-thirds of the homes studied by Wilcox had airflow rates lower than the minimum recommended value of 350 cfm per ton. Three of the houses had airflows near or below 200 cfm per ton, which can lead to coil icing (Parker 1997).

### How Airflow Affects System Operation and Energy Use

Very low airflow (below 200 cfm per ton) can lead to system failure due to icing of the indoor coil and can shorten equipment life. When airflow falls below 350 cfm, it begins to affect the efficiency of the system as shown in Figures 6 and 7 (Parker 1997). The figures were developed by testing a typical 3-ton cooling system under ARI standard conditions. Figure 6 shows how sensible, latent, and total capacity varied, and Figure 7 shows the impact on power, capacity and effective (as opposed to rated) EER. As airflow falls, latent capacity increases and peaks at about 250 cfm/ton, but then declines. At the same 250 cfm/ton airflow, sensible capacity falls to 75%, total capacity to 85%, and EER is reduced to about 92% of the value at 400 cfm/ton.

Though the impact on EER is not enormous, the shift to lower SHR is undesirable for dry climates and can cause comfort problems. Most importantly, airflows below 350 cfm/ton render standard tests for determining refrigerant charge invalid and can lead to improper charging. Airflows below 200 cfm/ton can lead to coil icing and greatly shorten compressor life.

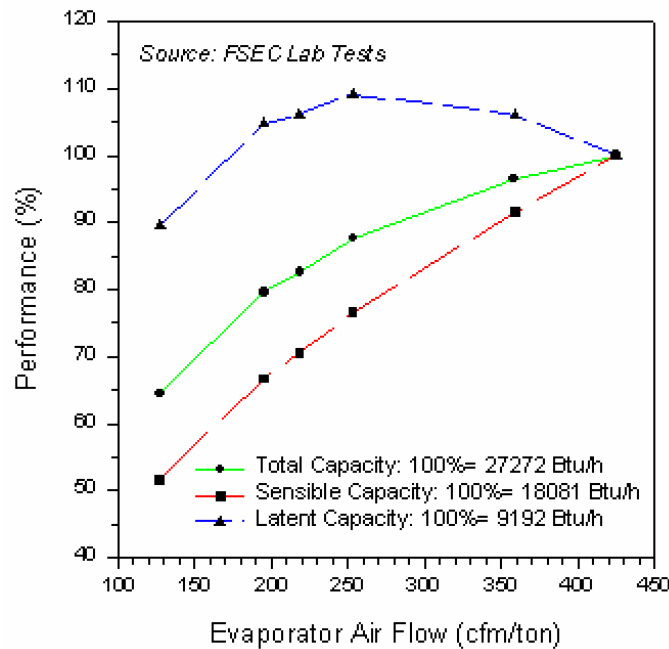
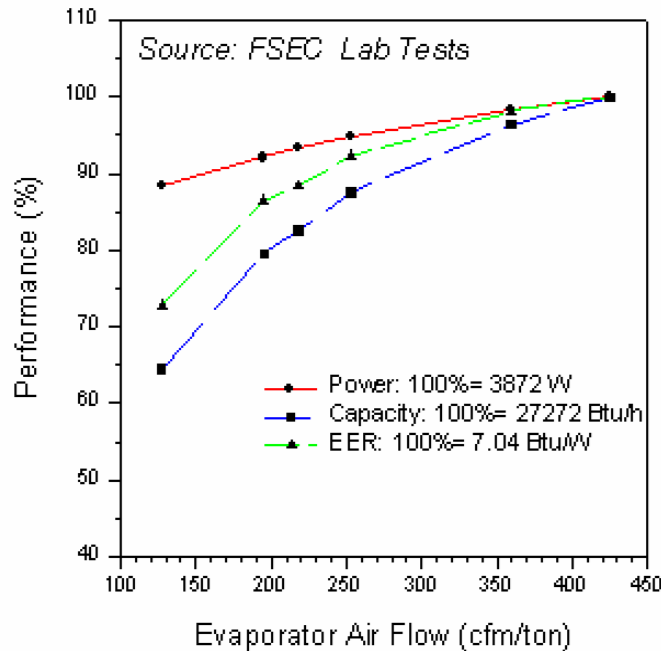


Figure 6. Impact of reduced airflow on cooling capacity (Source: Parker, reprinted with permission).



**Figure 7. Impact of reduced airflow on cooling system performance (Source: Parker, reprinted with permission).**

**Airflow Measurement Methods**

Methods for measuring airflow include the use of direct measurement devices such as flow hoods (balometer or powered types) and flow grids, and velocity measurement devices. Tests of a variety of balometers under laboratory conditions have shown they have potential errors of 20% to 30%, making them unsuitable for measuring total flow of residential systems (Wray 2002). Powered flow hoods (also known as fan flowmeters), were found to have an order of magnitude less error. They use a variable speed (ductblaster) fan to neutralize the pressure at the grille or match the pressure in the supply plenum. With a small amount of additional effort, airflow can be measured using a duct blaster at the beginning or end of a duct pressurization test by removing tape from the supply registers and running the system fan (+/- 3% error).

The Energy Conservatory TrueFlow flow grid, which inserts into the filter slot, has a measurement error of about 7%, which is acceptable for this application. Vane or hotwire anemometers can be used to measure velocity at grilles or for traverses of ducts, and can convert velocity to airflow rate. Measurement accuracy of anemometers is highly subject to the skill and diligence of the user and other factors.



**Figure 8. A powered flow hood in use.**

HVAC diagnostic procedures required by the California Energy Commission use temperature split (or difference between supply and return temperature) to check airflow across the evaporator and as a means of determining whether the refrigerant charge is valid (CEC 2008). Measurement with a calibrated airflow measurement device is preferred.

## 2.3 Refrigerant System Defects, Detection, and Correction

### ***What Can Go Wrong***

There are several faults that can occur with air conditioner refrigerant systems, and they can exist individually or in any combination creating a multitude of symptoms. Typical faults can include:

- Under or overcharge
- Refrigerant leaks
- Restrictions in the liquid line or expansion device
- Defective TXV or other expansion device
- Non-condensable gases in the system
- Condenser and evaporator fouling.

The difficulty with diagnosing refrigerant problems is that there can be multiple faults that are interactive. For example, a problem with a liquid line restriction or non-condensable contaminants in the system can lead to a false diagnosis of refrigerant charge. To arrive at a correct diagnosis requires that the system be inspected for faults in a particular order using a systematic approach that considers the environmental conditions during testing and other factors. Systematic diagnostic procedures are detailed in the Implementation section.

The challenge in performing diagnostic tests is to identify the particular source of problems that affect performance so that they can be corrected. Combinations of faults can yield false clues about the condition of refrigerant charge and can be very difficult to sort out. As reported by Purdue researchers, “evaluating fault detection and diagnosis is hard” and “false alarms happen” (Braun 2011). The methods described in this guideline are an attempt to significantly improve field diagnostic procedures and catch major problems, but there is no pretense they can produce ideal outcomes 100% of the time, even when followed precisely.

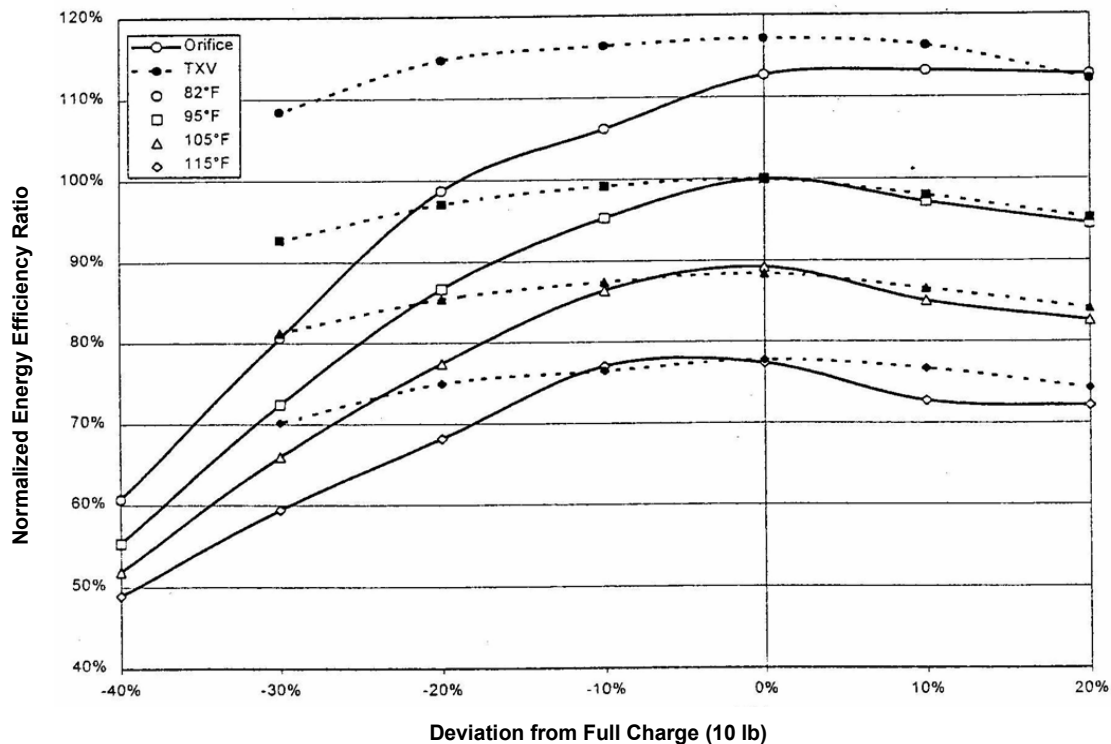
### ***The Importance of Correct Refrigerant Charge***

The amount of refrigerant in an air conditioner affects both the efficiency and capacity of the system. In addition to energy and capacity impacts, incorrect charge can also damage equipment. If the system is overcharged, there is a chance that liquid refrigerant will not completely evaporate and could slug the compressor. If there is too little charge in the system, the low suction line pressure and corresponding saturation temperature of the refrigerant can lead to ice formation on the evaporator, which restricts heat transfer, increases airflow resistance, and reduces airflow. This will further reduce air conditioner performance and can shorten compressor life (Siegel 2001).

Tests of more than 4,000 residential cooling systems in California indicated that about 34% were undercharged, 28% were overcharged, and only 38% had the correct charge (Proctor 1998). Additional field data for residential cooling systems gathered by Blasnik et al. (1996) and Proctor (1997) indicated that an undercharge of 15% is common. Temple (2006) identified how many units of a sample of 289 were within 4°F of the nominal charge for subcooling and 8°F for superheat. Of these, 28% had an acceptable charge, 31% were high, and 41% were low. Mowris (2004) reported field measurements of 4,168 air conditioners where 72% of systems had improper refrigerant charge and 44% had improper airflow. These studies point to the conclusion

that a large percentage of systems are operating below their rated efficiencies<sup>2</sup>. It should be noted that the accuracy of large sets of field measurements may be brought into question by technicians using analog pressure gauges and pipe temperature measurement instruments that might cause false undercharge diagnostics.

The impact of under and overcharge on EER is shown in Figure 9, which was developed from data obtained by Pacific Gas & Electric Company at their San Ramon test facility using systems with orifice and TXV expansion devices. These results verify that TXVs can mitigate the effects of undercharge provided they are properly installed. If the TXV sensing bulb does not have good thermal contact with the suction line, refrigerant charge can be misdiagnosed.



**Figure 9. Normalized EER versus charge and outside temperature (Source: P G & E, reprinted with permission).**

### ***The Effect of Non-Condensable Gases***

The presence of non-condensable gases in refrigerant systems is not given the attention it deserves. The existence of non-condensable gases (NCGs), such as air, nitrogen, or argon in refrigerant systems results in performance reductions. These gases, which have low boiling points, remain in gaseous phase under operating conditions, and will remain trapped in the condenser tubing. Detailed tests of air conditioners in California homes revealed that two of seven had refrigerant contaminated with NCGs. The efficiency of one of these systems was increased by 19% and the other by 35% when the existing refrigerant was removed and replaced with pure refrigerant (Proctor 2011). Contaminants can be left in systems as a result of inadequate evacuation or can enter the system during charge tests if proper procedures are not

<sup>2</sup> It should be noted that the accuracy of large sets of field measurements may be brought into question by technicians using analog pressure gauges and pipe temperature measurement instruments that might cause false under-charge diagnostics.



followed. A CEC report (CEC 2011) describes a scenario where the installer pressurized the inside coil and line set with 20 psig of nitrogen but failed to fully remove it prior to releasing the refrigerant into the system, resulting in a 42% reduction in sensible EER. Failure to frequently change the oil on a vacuum pump can prevent the pump from achieving the desired 240 micron<sup>3</sup> vacuum needed to properly evacuate refrigerant lines, resulting in nitrogen left in the line during charging.

### **Fault Detection and Diagnosis**

Detection of faults such as liquid line restrictions, faulty TXVs, and the presence of non-condensables require several diagnostic measurements, including:

- **Temperature split** is the difference between return air and supply air dry bulb temperatures measured upstream of the blower and downstream of the cooling coil. Temperature split is evaluated using a table (see Table 2). This table was developed by Carrier Corporation and modified for use in California’s Title 24 standards (CEC 2008). Proper temperature split (within +/-3°F) should be further validated with refrigerant charge diagnostic tests since proper temperature split can be found in systems that are overcharged or that contain non-condensables (Mowris 2011a). The temperature split method can be used to diagnose low capacity, which can be caused by refrigerant undercharge or liquid line restrictions.
- **Evaporator coil airflow.** Accurate measurement of airflow is needed to eliminate the uncertainty that test results may be a result of airflow that is below manufacturers’ specifications.
- **Superheat and subcooling.** For proper diagnosis, both superheat and subcooling should be measured. Other defects besides under/overcharge can yield false indications of charge condition, leading to a diagnosis that would result in incorrect actions to remediate the problem.
- **Condensing temperature over ambient (COA),** or the difference between the condensing saturation temperature and the temperature of the air entering the condensing coil is useful in differentiating between overcharge and the presence of non-condensables.
- **Evaporator saturation temperature (EST)** can provide clues about defects other than improper refrigerant charge, such as the presence of non-condensables and liquid line restrictions.

#### **Definitions**

*Superheat is the difference between the suction line refrigerant temperature and the vapor saturation temperature at the measured pressure.*

*Subcooling is the difference between the liquid line refrigerant temperature and the liquid saturation temperature at the measured pressure.*

The use of these measurements in diagnosing problems is described in the Implementation section of this document.

The importance of using accurate, well calibrated test equipment cannot be overemphasized as a means of avoiding false diagnostics and inappropriate application of refrigerant charge. Findings

<sup>3</sup> “Micron” refers to microns of Mercury. One micron is equivalent to 0.0046 psia (14.7 psia is standard atmospheric pressure).

from a study of the uncertainty of HVAC measurements concluded that with the methods and tools commonly used today, a simple refrigerant charge adjustment cannot be accurately implemented, measured, and verified. It is impossible to eliminate all sources of measurement uncertainty, but they should be mitigated where possible (for example, by using higher accuracy, calibrated instruments). The effects of measurement uncertainty need to be better understood by contractors and technicians (Hunt 2010).

### **Methods for Setting Refrigerant Charge**

When new air conditioners are installed, the correct charge can be approximated by weighing in a prescribed amount of refrigerant that takes into account the length and size of the refrigerant lines between the condensing unit and the cooling coil. This method can also be used with existing systems if they are fully evacuated (e.g. to 240 microns) beforehand, and is useful when outdoor temperatures are too low to apply superheat or subcooling methods. Manufacturers generally recommend verification of the charge using the methods described below.

The standard method for measuring refrigerant charge involves the use of pressure gauge sets and clamp-on temperature sensors to measure subcooling and/or superheat. Fixed orifice systems require measurements of outdoor dry bulb and return air wet bulb temperatures, and charge is typically set based on superheat. However, subcooling should also be measured to insure that the expansion device is receiving a solid stream of liquid.

Proper refrigerant charge for systems with TXVs is typically determined by measuring subcooling, but measuring superheat will verify that the TXV is operating correctly. Measured values are compared against target values provided by manufacturers.

Refrigerant charge should be checked while temperature conditions are favorable, that is when the outdoor temperature is between 70°F and 100°F and the indoor temperature is between 70°F and 80°F (Carrier 2010). Provided the indoor dry bulb temperature can be maintained above 70°F, refrigerant charge adjustments for systems with TXVs may be made at outdoor temperatures as low as 65°F (Mowris 2010, CEC 2008). Systems without TXVs should not be tested at outdoor temperatures below 70°F.

### **Diagnostic Methods for Assessing Refrigerant Charge**

The most accurate way to verify proper refrigerant charge is to measure subcooling and superheat using manifold gauges. If proper airflow has been verified and is greater than 300 cfm per ton, the temperature split across the cooling coil can provide an indication that there is a charge problem. This diagnostic method can identify undercharge conditions in the absence of an EPA-certified technician, but should be followed up with refrigerant charge tests and other diagnostics.

For systems that are very low on charge, refrigerant leaks can be detected by pressurizing them with nitrogen. Systems should be able to maintain a pressure of 150 psig for 10 minutes. Also, test equipment is available for detecting refrigerant leaks.



**Figure 10. A digital manifold gauge.**

### **Evaporator Coil Fouling and Damage to Distributor Tubes**

Fouling of the evaporator reduces airflow, will eventually cause the coil to freeze over, and may damage the compressor. If fouling affects one or more coil circuits, the circuit receiving less airflow will boil off less refrigerant. If a TXV is installed, it will throttle back, starving the rest of the coil. Superheat will be in the correct range, but capacity and efficiency will be reduced. Coil fouling can be checked by removing the coil inspection cover. The first indication may be low airflow and high pressure drop. If access to the evaporator is difficult, measuring the pressure drop across the coil will indicate significant fouling, but minor fouling can affect coil performance without substantially increasing pressure drop. For an evaporator that is properly sized and clean, the pressure differential should not be more than 0.25 in. w.c. (about 62 Pa). Though rare, kinked or blocked distributor tubes will reduce cooling capacity. A method for detecting this problem is provided in Attachment C.



**Figure 11. Examples of condenser and evaporator coil fouling.**

## **2.4 Repair or Replace?**

### **Condenser Replacement**

In some cases replacement of the air conditioner condensing unit and indoor coil is necessary to restore proper function or to head off an impending equipment failure. In such cases, improving efficiency becomes secondary to providing comfort. Other indications that the condensing unit should be replaced include:

- The system is more than 15 years old (especially if it contains R-12 refrigerant)
- The condensing unit is oversized due to incorrect initial sizing or reduced loads due to improvements to the building enclosure
- Compressor amps are higher than rated and/or the compressor is noisy or failing
- The condenser coil is damaged (fins compressed or corroded).

Replacement affords an opportunity to reduce cooling costs by installing a more efficient system. Federal standards for air conditioners were set at SEER 10 in

### **The Evolution of Refrigerants**

*Chlorofluorocarbon (CFC) refrigerants including R-12 were phased out of production in 1996 because of their high ozone depletion potential (ODP). R-22, an HCFC, began being phased out in 2010 and is being replaced by R-410a, an HFC. By virtue of not containing chlorine, R-410a has zero ODP. R134a, another HFC, was also used in some systems in the 1990's, but is being phased out due to its high global warming potential.*

1987 and raised to SEER 13 in 2006. To meet ENERGY STAR<sup>®</sup> specifications, split system air conditioners must have a SEER rating of 14.5 or higher and an EER of 12 or higher. A decision tree for guiding the replacement decision is provided in Attachment C.

It is typical for the evaporator coil and expansion valve to be replaced at the same time that the condenser is replaced. Most coils designed for R-22 are not rated for the higher pressures used with R410A refrigerant. At the time of replacement, both the size of the system and the size of the indoor coil should be re-evaluated. If improvements were made to the home since the previous system was installed, proper load and equipment sizing should be performed using ACCA Manual J, D, and S sizing methodologies. Equipment and operating costs can be saved by installing a correctly sized system.

Installing an indoor coil that is one size larger than the condensing unit (for example installing a 3-ton coil with a 2-ton condenser) will improve airflow by reducing coil pressure drop<sup>4</sup>. The larger coil surface area will also improve system efficiency but will increase the SHR and may not be appropriate for humid climates. At time of replacement, the sizing and tightness of ducts and particularly the size of the filter grille should be evaluated and upgraded as needed.

Many of the same defects that are seen in existing systems can appear in newly installed systems if proper procedures for insuring adequate airflow and providing the correct refrigerant charge that is free of non-condensables are not taken. Air, nitrogen and other gases that are left in the system and mixed with the refrigerant can have severe impacts on performance. Non-condensables accumulate in the condenser and blanket heat transfer surfaces, which impacts condenser coil performance and efficiency. ASHRAE (2010) recommends evacuation to 240 microns of mercury (Hg) to remove non-condensable air or nitrogen. Removal of moisture is a much more difficult task. According to ASHRAE (2010) “Excess moisture in refrigeration systems may lead to freeze-up of the capillary tube or expansion valve. Contaminants can cause valve breakage, motor burnout, and bearing and seal failure. Except for freeze-up, these effects are not normally detected by a standard factory test. Therefore, it is important to use a dehydration technique that yields a safe moisture level without adding foreign elements or solvents. In conjunction with dehydration, an accurate method of moisture measurement must be established. Many factors, such as the size of the unit, its application, and the type of refrigerant, determine the acceptable moisture content.”

Proper evacuation is important for R22 and much more important for R410A since the synthetic polyolester (POE) oil is highly hygroscopic and moisture in R410A systems will cause significant problems (Maier 2009). Installation of a liquid line filter drier is required for all systems and a new filter-drier must be properly installed whenever a system is opened. When POE oils are exposed to moisture and heat, they may react, forming acid that is harmful to the system. If a R410A POE oil system is open for the same amount of time service technicians are used to having R22 mineral oil systems open, there is a much greater chance of moisture contamination of the oil. It is imperative that technicians keep containers of POE oils sealed, except when the oils are actually being dispensed. POE oils should also be stored properly in their original container because many plastics used to package oils are permeable to moisture. It is also important to keep compressors and systems closed, except when work is actually being

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<sup>4</sup> Codes in some locations require that evaporators and condensing units be matched and that their ratings be included in the AHRI database.

done on the equipment, and to filter out undesirable contaminants. This can be achieved with proper installation and service techniques as well as the use of correct filters and driers.

### **The Value of High Efficiency Systems**

If replacement is required, installing an air conditioner that has an efficiency that exceeds the national standard of 13 SEER can be a good investment. Table 1 lists annual energy savings and costs for 15 and 18 SEER air conditioners that were calculated from BEopt simulations of a typical 1800 ft<sup>2</sup> house in five climate zones. Savings and costs are relative to a SEER 13 system. Attachment B lists all the assumptions that were used in this analysis.

These estimates suggest that replacement with a SEER 15 system may be justified, particularly if there is a source of low interest financing. For example, for a 5%, 15-year loan, a SEER 15 system would save enough energy to produce a positive cash flow in all but the marine climate zone. In this case, analysis of a SEER 18 system yielded negative cash flows in all climate zones<sup>5</sup>.

### **Replacement of Other Components**

**Furnace.** If the air conditioner is being replaced, replacement of the furnace should be considered.

Furnace efficiency has improved significantly over the past 15 years, and the older the furnace, the greater the risk that cracks in the heat exchanger could introduce carbon monoxide into the house. The cost to replace the furnace when the air conditioner is replaced will be lower than if it is done independent of the air conditioner because the cooling coil must typically be removed in both cases. Some newer coils can accept either R-22 or R-410A, and should be labeled as such.

**Blower Motor.** With many furnaces it is possible to replace the existing PSC motor with a more efficient brushless DC permanent magnet (ECM) motor. Because the ECM motors are usually 1-2 in. longer than PSC motors, the dimensions of the furnace should be checked before a new motor is ordered. The larger motor will not fit in some of the slim furnaces that are found in the field.

To determine the energy savings and economic viability of replacing a PSC motor with an ECM motor, a Regal Beloit (Genteq) “Evergreen” motor was obtained and tested in a 1970’s vintage furnace. Energy use of the two motors was used to estimate energy savings as function of operating hours. A detailed description of the test and results is provided in Attachment B. Findings indicate that, unless it is necessary to replace a defective motor, energy savings will probably not offset the incremental cost if amortized at 5% and 15 years.

**Table 1. Estimated Energy Savings, Replacement Costs, and Simple Paybacks for 15 SEER vs. 13 SEER Air Conditioner.**

<b>Electric Utility Savings</b>	<b>15 SEER</b>
Hot-Dry	\$82
Hot-Humid	\$95
Mixed-Humid	\$59
Marine	\$11
Cold	\$29
<b>Incremental Cost</b>	
Hot-Dry	\$360
Hot-Humid	\$360
Mixed-Humid	\$300
Marine	\$180
Cold	\$240
<b>Simple Payback, years</b>	
Hot-Dry	4.4
Hot-Humid	3.8
Mixed-Humid	5.1
Marine	16.5
Cold	8.2

<sup>5</sup> Discrepancies between modeling results of typical homes and utility survey data could raise questions about the validity of these results. See Attachment B.

Replacement is easier to justify in buildings that have very long blower run times, and run time depends on the climate, the integrity of the thermal enclosure, temperature setpoints, and other factors.

**Condensing Unit Components.** Generally when a compressor fails, the HVAC contractor replaces the entire condensing unit. However, contractors should assess the practicality of replacing only the compressor. The size and condition of the condensing coil is a determining factor of efficiency, and replacing compressors in older systems that have large coils in good condition could be much more cost-effective than full replacement. If the evaporator coil is designed for R-22 refrigerant, then replacing the compressor would also eliminate the cost of trading out the R-22 evaporator coil with one rated for R-410A.

Parker (2005) researched the energy savings from improved condenser fan and exhaust configuration, as well as replacement of PSC motors with brushless DC motors. He concluded fan improvements could reduce motor power by about 49 W (26%), and combined with motor replacement could reduce power by about 100 W. More efficient and quiet condenser fan blades are commercially available.

### 3 Section 3: Implementation

Two approaches to air conditioner diagnosis and repair are provided: the Preliminary Diagnostic (PD) approach, for quickly identifying and correcting serious system design or installation problems without the need for special training or test equipment; and the Comprehensive Diagnostic (CD) approach that provides a procedure and methodology for qualified HVAC technicians to accurately identify system defects using the proper diagnostic tools. Measurement of “temperature split” is used as a key determinant that there are significant refrigerant problems and that the CD method must be employed.

It is important that duct testing and sealing is done in advance, as return duct leakage can result in false airflow readings and incorrect temperature split readings. For proper procedures on testing and sealing ductwork in existing homes, refer to the *Building America Measure Guideline: Sealing and Insulating Ducts in Existing Homes* (Aldrich 2011).

Some states, such as Florida, have specific requirements regarding who is qualified to perform work on HVAC systems. While tasks described in the PD method require little training, it may be necessary to have a licensed HVAC contractor perform this work.

#### 3.1 The Preliminary Diagnostic (PD) Method **Objectives and Scope**

The PD Method is designed for use by the home performance contractor and has three primary objectives:

- Find major defects
- Fix what can be fixed given available skills, tools, and time
- Identify what information should be referred to an HVAC technician if further repairs are needed.

The scope of the PD method includes:

- Observation of the general condition of the system and obvious maintenance needs
- Verification of proper airflow and quick diagnosis of the source of airflow restrictions if airflow is too low
- A temperature split test to identify low system capacity, efficiency, and overall condition.

If problems are identified, the contractor can inform the homeowner of completed repairs and/or provide a recommendation and referral for further diagnostics and maintenance.

It is understandable for home performance contractors to strive to minimize their time on site to keep costs down. Less than one hour is required to complete the PD process, provided duct and seal has already been completed. The PD process first evaluates overall system performance by measuring supply-return temperature difference (temperature split) to diagnose airflow and capacity problems. The process either ends with this test or continues with an airflow test to isolate the problem if airflow is too low.

### **Skills and Tools Required**

The PD is designed to fit the skill level of most home performance contractors who have had BPI training, and to only require test equipment that is likely to be in their toolbox. The following equipment is required:

- Digital manometer with hoses and static pressure probes
- Duct blaster, flow grid, or accurate balometer
- Digital thermometer, preferably with two probes that can simultaneously measure supply and return temperatures (see Attachment D for specifications)
- Digital relative humidity device that can also report wet bulb temperature, or a psychrometric chart or tool (available as smart phone “app”)
- Portable drill and 5/16 in. high speed bit
- Aluminum tape for covering holes drilled into ducts
- Tape measure, screwdrivers, nut drivers, lights, ladders, and other commonly carried tools.

#### **Accuracy of Test Equipment**

*Accuracy of test equipment can vary widely resulting in false diagnosis. Refer to Attachment C for recommended specifications.*

A hose with a spray nozzle and coil cleaning solution will also be useful if the outdoor coil requires cleaning.

### **Measuring Airflow**

A duct blaster can be used as a powered flow hood to measure airflow, and measurements can be completed as an extension of duct pressurization tests by simply removing the seals from the registers and leaving the duct blaster attached to the return grille. A flow grid (see Figure 12) or an accurate balometer may also be used. The supply plenum pressure (NSOP) must be measured before starting tests; for the duct blaster, it is needed to match the plenum pressure; and for the flow grid and balometer, to adjust the measured airflow using the equation below:

$$\text{Corrected airflow} = \text{Measured airflow} \times \sqrt{\text{NSOP}/\text{TSOP}} \text{ where:}$$

NSOP = normal system supply operating pressure (at supply plenum)  
 TSOP = tested system operating pressure (with flow grid or balometer in place)

More detailed information on airflow measurements is provided in Attachment C. Note that newer furnaces and air handlers may be equipped with a variable speed motor that will attempt to overcome high static pressures to deliver a constant cfm. It is not necessary to calculate corrected airflow for these systems.



**Figure 12. The Energy Conservatory TrueFlow flow grid.**

#### **IMPORTANT!**

*For systems with gas furnaces, a combustion safety check should be completed in accordance with BPI specifications before starting any work.*

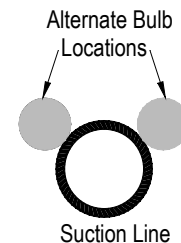


### **Before Starting Tests**

Conduct a general review of the system to identify the location and general condition of the furnace or air handler, condensing unit, thermostats, return air grille(s) and filter(s), and other characteristics such as the number of controlled zones and whether a bypass damper is installed. Use Table A-1 to record this information. If possible, interview the owner to determine the age and service history of the system and to obtain information on any comfort problems experienced. If any electrical or other safety hazards are observed, they should be corrected before beginning.

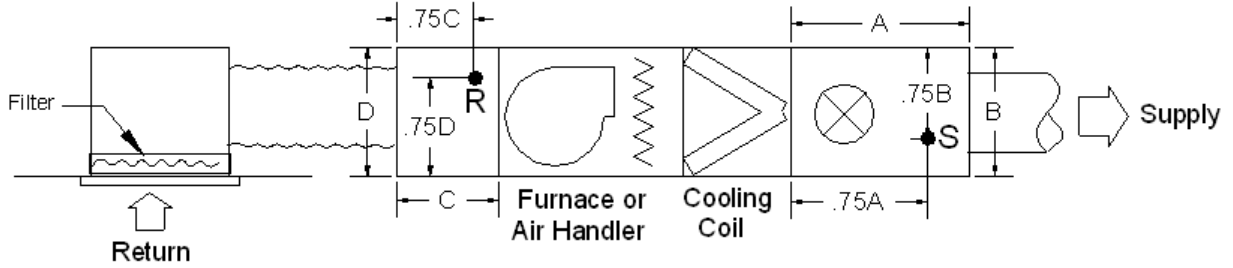
### **Preliminary Diagnostic Steps**

1. Since duct leakage affects airflow and system pressures, ideally ducts should be sealed to a maximum leakage of 15% of total airflow or 10 cfm per 100 ft<sup>2</sup> of conditioned floor area before any testing is initiated. If duct test and seal (DTS) was completed, or if leakage was measured without sealing, record the duct leakage in Table A-1.
2. Determine the condensing unit size from the nameplate. Model numbers of most systems include a 24, 30, 36, etc. to indicate the size of the system in kBtuh. Divide by 12 to obtain the size in tons.
3. Check the condenser coil (outdoor unit) for fouling. If the coil is clearly plugged, pull the fuse, remove panels from the unit to gain access to the inside surface of the coils and use biodegradable coil cleaning solution and a hose with a nozzle to clean the coils. Refer to Attachment C for detailed instructions on coil cleaning methods.
4. Check the condition of the insulation on the suction line and note its location.
5. If the system has a TXV, verify that the sensing bulb is securely fastened to a horizontal section of the suction line with a copper strap, in contact over its entire length, and is insulated with a single wrap of insulation with 50% overlap (see Figure 13).
6. Determine the filter velocity at nominal conditions (400 cfm/ton) using:



**Figure 13. Proper position of TXV sensing bulb.**

- $\text{Velocity} = \text{Tons} \times 57,600 / (\text{L} \times \text{W})$  where L and W are the filter dimensions in inches
  - If the velocity exceeds 300 fpm or the filter is obviously fouled, remove it for the duration of the tests and note that a larger filter grille or slot must be provided.
7. Drill 5/16 in. holes at points R (return) and S (supply) as shown in Figure 14.



**Figure 14. Temperature and pressure test points.**

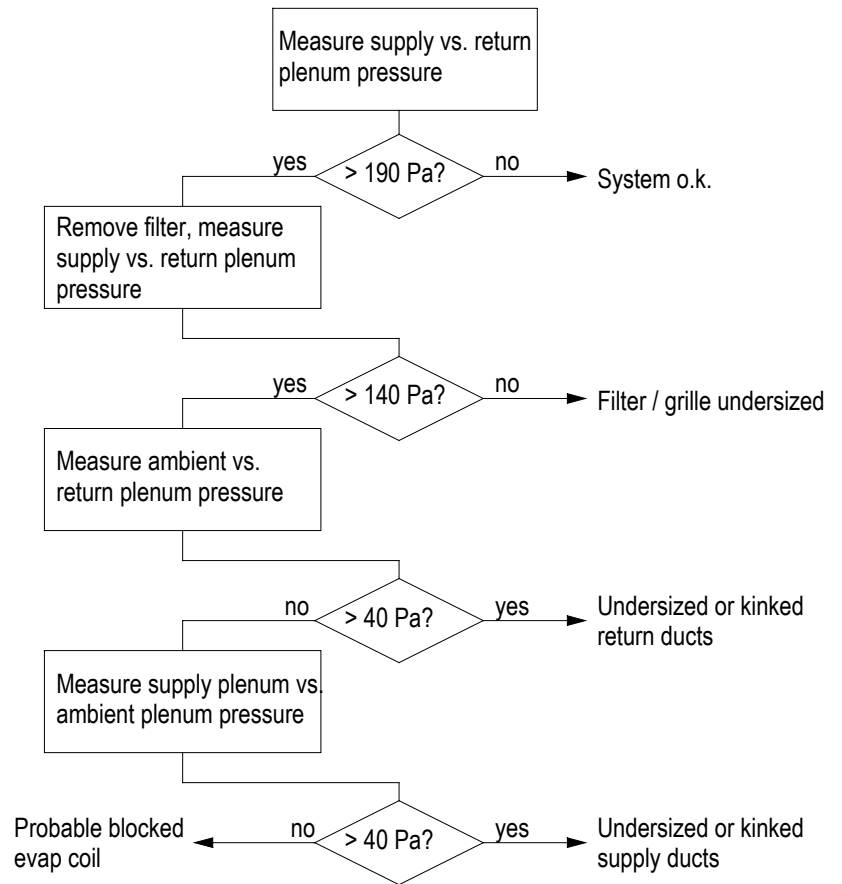
8. Start the system fan by removing the condensing unit power fuse and setting the thermostat(s) to call for cooling. If the system has multiple zones, manually close the bypass damper (if present), activate all zone thermostats, and verify that there is airflow at registers in each zone.
9. Measure airflow and make corrections as needed using supply plenum pressures (NSOP and TSOP). Enter results in Table A-1. If the airflow is less than 300 cfm per ton, proceed to the next step, otherwise jump to Step 11. For detailed instructions on airflow measurements, see Attachment C.

10. Using a manometer, complete the pressure loss diagnostic as shown in Figure 15. The first measurement point is connected to “input” and the second to “ref” of the manometer. For example, for the first test connect the supply probe to “input” and the return probe to “ref”.

The “ambient probe can be located in the attic or indoor space. If the equipment is located in an unconditioned attic or basement, the access hatch or door should be left open to equalize pressures between the house and the equipment space.

The static pressure probes should be inserted perpendicular to the airflow to a depth that produces the highest pressure.

The pressure drop of the



**Figure 15. Pressure loss diagnostic flow chart.**

evaporator coil could be directly measured by drilling another hole between the blower unit and the coil, but this is not recommended as it would risk causing permanent damage to the coil and releasing refrigerant.

It is possible that problems identified during this test can be corrected on the spot, for example, straightening a kinked duct. If this is the case, repeat airflow measurements to verify that the corrective action was successful.

If the system includes a variable speed fan motor and high pressure losses are suspected even though the airflow exceeds 300 cfm, the pressure loss diagnostic can be used to identify problems that, if corrected, will reduce fan energy use and noise.

11. Measure temperature split. Re-insert the condensing unit fuse and wait 15 minutes or more for the system to stabilize. Then use the digital thermometer to measure and record the dry bulb air temperature at points R and S. Also measure the wet bulb temperature at R or at the return grille. The probes should be centered in the air stream. Take measurements after two minutes, or after the readings have stabilized. Enter all three temperatures in Table A-1. Subtract the supply temperature (S) from the return air dry bulb temperature (R) to calculate the temperature split, and enter it in Table A-1.

Over the range of indoor dry bulb temperatures of 76°F–80°F, corresponding wet bulb temperatures must be 50°F–60°F in order to obtain target temperature split values (see Table 2). If indoor wet bulb temperature is too low, it may be increased by opening windows or running the heating system; otherwise the test will have to be delayed until weather conditions are more favorable.

12. Refer to Table 2 and select the Target Temperature Split that corresponds to the return air dry bulb and wet bulb measurements, and enter the target temperature in Table A-1. The measured temperature split should be within  $\pm 3^\circ\text{F}$  of values listed in the table. If the temperature split is more than  $3^\circ\text{F}$  below the target value, one or more defects are causing a reduction in capacity. Discontinue testing and refer the system to a qualified technician for further diagnostics and repairs. A high temperature split can indicate low airflow.

If the system fails the airflow measurement tests or the temperature split tests, or if other significant problems are uncovered during the PD process, provide the HVAC technician with a completed PD Form (Table A-1). This information will allow him/her to focus on the critical problems and avoid duplication of effort. Work to improve airflow may include installing a larger filter grille, long radius elbows, turning vanes, or larger ducting. The HVAC technician should re-check airflow and temperature split before proceeding with refrigerant charge tests.

**Table 2. Target Temperature Split (Return Dry Bulb – Supply Dry Bulb)**  
 Source: California Energy Commission (CEC 2008).

		Return Air Wet-Bulb (°F) ( $T_{return, wb}$ )																											
		50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	
Return Air Dry-Bulb (°F) ( $T_{return, db}$ )	70	20.9	20.7	20.6	20.4	20.1	19.9	19.5	19.1	18.7	18.2	17.7	17.2	16.5	15.9	15.2	14.4	13.7	12.8										
	71	21.4	21.3	21.1	20.9	20.7	20.4	20.1	19.7	19.3	18.8	18.3	17.7	17.1	16.4	15.7	15.0	14.2	13.4	12.5									
	72	21.9	21.8	21.7	21.5	21.2	20.9	20.6	20.2	19.8	19.3	18.8	18.2	17.6	17.0	16.3	15.5	14.7	13.9	13.0	12.1								
	73	22.5	22.4	22.2	22.0	21.8	21.5	21.2	20.8	20.3	19.9	19.4	18.8	18.2	17.5	16.8	16.1	15.3	14.4	13.6	12.6	11.7							
	74	23.0	22.9	22.8	22.6	22.3	22.0	21.7	21.3	20.9	20.4	19.9	19.3	18.7	18.1	17.4	16.6	15.8	15.0	14.1	13.2	12.2	11.2						
	75	23.6	23.5	23.3	23.1	22.9	22.6	22.2	21.9	21.4	21.0	20.4	19.9	19.3	18.6	17.9	17.2	16.4	15.5	14.7	13.7	12.7	11.7	10.7					
	76	24.1	24.0	23.9	23.7	23.4	23.1	22.8	22.4	22.0	21.5	21.0	20.4	19.8	19.2	18.5	17.7	16.9	16.1	15.2	14.3	13.3	12.3	11.2	10.1				
	77	-	24.6	24.4	24.2	24.0	23.7	23.3	22.9	22.5	22.0	21.5	21.0	20.4	19.7	19.0	18.3	17.5	16.6	15.7	14.8	13.8	12.8	11.7	10.6	9.5			
	78	-	-	-	24.7	24.5	24.2	23.9	23.5	23.1	22.6	22.1	21.5	20.9	20.2	19.5	18.8	18.0	17.2	16.3	15.4	14.4	13.4	12.3	11.2	10.0	8.8		
	79	-	-	-	-	-	24.8	24.4	24.0	23.6	23.1	22.6	22.1	21.4	20.8	20.1	19.3	18.5	17.7	16.8	15.9	14.9	13.9	12.8	11.7	10.6	9.4	8.1	
	80	-	-	-	-	-	-	25.0	24.6	24.2	23.7	23.2	22.6	22.0	21.3	20.6	19.9	19.1	18.3	17.4	16.4	15.5	14.4	13.4	12.3	11.1	9.9	8.7	
	81	-	-	-	-	-	-	-	25.1	24.7	24.2	23.7	23.1	22.5	21.9	21.2	20.4	19.6	18.8	17.9	17.0	16.0	15.0	13.9	12.8	11.7	10.4	9.2	
	82	-	-	-	-	-	-	-	-	25.2	24.8	24.2	23.7	23.1	22.4	21.7	21.0	20.2	19.3	18.5	17.5	16.6	15.5	14.5	13.4	12.2	11.0	9.7	
	83	-	-	-	-	-	-	-	-	-	25.3	24.8	24.2	23.6	23.0	22.3	21.5	20.7	19.9	19.0	18.1	17.1	16.1	15.0	13.9	12.7	11.5	10.3	
	84	-	-	-	-	-	-	-	-	-	25.9	25.3	24.8	24.2	23.5	22.8	22.1	21.3	20.4	19.5	18.6	17.6	16.6	15.6	14.4	13.3	12.1	10.8	

## 3.2 The Comprehensive Diagnostic (CD) Method

### ***Objectives and Scope***

The Comprehensive Diagnostic (CD) Method provides guidance to the HVAC technician and is designed to detect and remedy most HVAC performance deficiencies by applying accurate measurements and repairs in the correct sequence. This method will allow the technician to zero in on problems efficiently so as to maximize energy savings impact and minimize time on the job. If a preliminary diagnostic (PD) was not previously completed, the scope of the CD Method includes a PD.

### ***Skills and Tools Required***

The CD Method assumes that the technician performing the work has proper training and experience and well-maintained equipment. Skills in diagnosing refrigerant system defects require a mix of art and science and improve with experience. In addition to commonly used tools and materials, the technician should have the following:

- Manifold set (digital preferred)
- Digital thermometer, preferably with two probes that can simultaneously measure supply and return temperatures
- Digital relative humidity device that can also report wet bulb temperature, or a psychrometric chart or tool (available as smart phone “app”)
- Digital thermometer with clamp-on thermocouples for measuring line temperatures (if not included with manifold set)
- Vacuum pump (well maintained) and micron pressure gauge
- Scale for weighing refrigerant
- Refrigerant recovery system
- R-22 and R-410a refrigerant
- Digital manometer with hoses and static pressure probes
- Flow grid, fan powered flow hood , or balometer (if airflow was not previously measured and verified to be greater than 300 cfm per ton)
- Clamp-on amp meter
- Condenser air exit restrictor (for cold weather testing)

If evacuation of the system is necessary, a four-valve gauge manifold with one 3/8 in. hose for connection to the vacuum pump and a separate hose to the refrigerant cylinder is recommended to speed evacuation. The valve core depressors should be removed to minimize restrictions. Measurement inaccuracies have a compounding effect that can result in incorrect refrigerant charge or misdiagnosis. Temperature measurement equipment should meet the specifications listed in Attachment D.

### **Refrigerant Charge Test Overview**

This procedure<sup>6</sup> is designed to identify and correct refrigerant under/overcharge, liquid line restrictions, defective expansion devices, and the presence of non-condensables. Charging procedures typically only require measurement of superheat for fixed orifice expansion devices and subcooling for systems with TXVs. However, identification of defects other than under/overcharge requires measurement of both subcooling and superheat, as well as condensing temperature over ambient temperature (COA). Methods for obtaining these measurements are described, as well as methods for using these data to diagnose defects.

#### **Need a Refrigerant Refresher?**

*To review information on safe refrigerant charging and recovery techniques, see:  
[www.epatest.com/608/manual/Manual.htm](http://www.epatest.com/608/manual/Manual.htm)*

All measurements and calculations should be recorded in Table A-2 so they can be used for later diagnosis. Whenever adjustments to refrigerant charge or other corrections are made, the system should be allowed to run 15 minutes before tests are repeated. Systems should not be tested when the outdoor temperature is below 55°F, and ideally should be tested at temperature and humidity conditions that are close to operating conditions.

### **Before Starting Tests**

Identifying obvious problems at the outset will minimize the number of diagnostic variables and save time. Conduct a general review of the system to identify the location and general condition of the furnace or air handler, condensing unit, thermostats, return air grille(s) and filter(s), the number of controlled zones, the type of refrigerant and expansion valve used (fixed orifice or TXV), and the length and location of the refrigerant lines. If a TXV is used, insure the bulb is properly secured to the suction line and insulated. This information should be entered in Table A-1. If any wiring or electrical hazards are observed, they must be corrected before continuing. If a bypass damper is installed, lock it in the fully closed position. Inspect the condenser coil and, if necessary, clean it using the procedure described in Attachment C.

Review prior testing or maintenance completed and fill in any missing information in Table A-1. Ducts should be pressure tested and sealed to 15% of system airflow or 10 cfm per 1,000 ft<sup>2</sup> of conditioned floor area before proceeding.

### **Test Procedures**

1. Measure airflow (if not previously measured) using the method described in PD Step 9, and follow the airflow diagnostic described in Figure 16 if necessary to identify problem areas. Refer to Attachment C for detailed measurement methods. For proper refrigerant diagnostics, airflow should be no less than 300 cfm/ton, and can be as high as 500 cfm/ton in dry climates. It may be necessary to remove the filter to achieve the required airflow during testing.
2. Measure temperature split (if not previously measured) using the method described in PD step 11.

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<sup>6</sup> Adapted from the method described by the California Energy Commission (CEC 2008) and modified with the assistance of Jim Phillips of The Energy Savers.

3. If the indoor temperature is less than 65°F, use the heating system or open windows as needed to insure that the return air dry bulb temperature will be not less than 70°F when measurements are taken. Ideally, the indoor temperature should be no greater than 80°F.
4. Apply a clamp-on amp meter to one leg of the power to the compressor.
5. Connect the refrigerant gauges to the service ports, taking proper precautions to avoid introducing air into the system.
6. Turn the cooling system on and let it run for 15 minutes to stabilize temperatures and pressures before taking any measurements. While the system is stabilizing, proceed with attachment of the temperature sensors.
7. Securely attach one clamp-on temperature sensor to the suction line near the suction line service valve and attach another clamp-on temperature sensor to the liquid line near the liquid line service valve. Attachment points on lines should be thoroughly cleaned to insure good thermal contact between the thermocouple and the line. If the refrigerant line set is long and runs through a hot attic, consider measuring the suction line temperature at the indoor coil instead of at the condensing unit.
8. Provide a temperature sensor to measure the condenser entering air dry-bulb temperature. Position the sensor so that it records the average condenser air entering temperature and is shaded from direct sun.

### **Airflow Too Low?**

*First, go to the next higher blower motor tap setting or dip switch setting and retest.*

*Next, remove the filter and test again. If airflow is still less than 300 cfm/ton, then repairs will be needed before refrigerant charge can be tested.*

*If removing the filter provides sufficient airflow for tests to proceed, be sure to include the addition of a second return or larger filter grille to the repair list. The velocity at the filter(s) should be no more than 300 fpm.*

### **Measurements (Enter all values into Table A-2)**

9. Measure the return air dry bulb ( $T_{\text{return, db}}$ ), return air wet bulb ( $T_{\text{return, wb}}$ ) and the supply air dry bulb temperature ( $T_{\text{supply, db}}$ ).
10. Measure the evaporator saturation temperature ( $T_{\text{evap, sat}}$ ) from the low side gauge.
11. Measure the condenser saturation temperature ( $T_{\text{cond, sat}}$ ) from the high side gauge.
12. Measure the suction line temperature ( $T_{\text{suction}}$ ) from the clamp-on thermocouple.
13. Measure the liquid line temperature ( $T_{\text{liquid}}$ ) from the clamp-on thermocouple.
14. Measure the condenser entering air dry-bulb temperature ( $T_{\text{cond, db}}$ ) by averaging the coil entering air temperatures at the bottom, center, and top of coil.
15. Measure compressor amps.

### **Cold and Hot Weather Testing**

The ideal conditions at which to evaluate refrigerant charge is at AHRI test conditions of 80°F indoor dry bulb, 67°F indoor wet bulb, and 95°F outdoor dry bulb, but it is rare that these conditions will exist when testing is being performed. For non-TXV systems, lower indoor temperatures will generally produce a high temperature split and low subcooling. Lower outdoor temperatures will result in higher superheat (that can disguise an overcharge) and higher outdoor

temperatures may reduce subcooling to zero (suggestive of an undercharge). These factors must be considered when evaluating test results.

Refrigerant charge tests should not be attempted at outdoor temperatures below 55°F and indoor temperatures below 70°F. Operation below 55°F outdoor temperature can damage the compressor due to flooding and inadequate lubrication. For non-TXV systems, target superheat values below 5°F do not exist, and for TXV systems it may be difficult to achieve manufacturers' subcooling targets. It is recommended that tests be delayed at least until weather conditions result in indoor and outdoor temperatures over 70°F.

If the outdoor temperature is below 80°F and superheat is too high (because of too much refrigerant in the condenser), the discharge from the condensing unit may be partially blocked to simulate an outdoor temperature of 95°F<sup>7</sup>. For R-22 systems, raising the high side pressure to 278 psi using this method would correspond to a condensing temperature of 126°F, which would be reasonable for 95°F outdoor air temperature. However, COA must be measured and recorded first because the higher condensing temperature will produce a false COA reading. This method should never be used at outdoor temperatures below 55°F or if not recommended by the manufacturer.

Low indoor dry bulb and wet bulb temperatures can be increased by running the heating system and/or by using plug-in electric resistance heaters. The indoor temperature must be maintained above 70°F for at least 15 minutes to allow useful charge readings to be taken.

### 3.3 Calculations

#### ***Temperature Split***

Calculate the temperature split using:

$$\text{Temperature Split} = T_{\text{return, db}} - T_{\text{supply, db}}$$

Find the target temperature split from Table 2 using  $T_{\text{return, db}}$  and  $T_{\text{return, wb}}$ . The measured temperature split should not deviate from the target value by more than 3°F.

#### ***Condensing Temperature Over Ambient (COA)***

Calculate the difference between the condensing saturation temperature ( $T_{\text{cond, sat}}$ ) and the temperature of the air entering the condenser ( $T_{\text{cond, db}}$ ).

$$\text{COA} = T_{\text{cond, sat}} - T_{\text{cond, db}}$$

A high COA (for example 30°F or higher) can indicate an overcharge, liquid line restriction, or non-condensables in the refrigerant. These defects can be distinguished by evaluating subcooling and superheat results.

#### ***Superheat***

Calculate Actual Superheat as the suction line temperature minus the evaporator saturation temperature:

$$\text{Actual Superheat} = T_{\text{suction}} - T_{\text{evap, sat}}$$

<sup>7</sup> This method is documented in CEC 2011



**For fixed orifice expansion devices:** Find the Target Superheat in Table 4 that corresponds to the measured return air wet-bulb temperature ( $T_{\text{return, wb}}$ ) and condenser air dry-bulb temperature ( $T_{\text{cond, db}}$ ). If a dash mark is read from Table 4, the target superheat is less than 5°F and it is difficult to perform a refrigerant charge verification test under these conditions.

Calculate the difference between actual superheat and target superheat (Actual Superheat – Target Superheat). The difference should not vary by more than  $\pm 5^\circ\text{F}$ . Refrigerant is normally removed if actual superheat is more than 5°F below target (negative difference) and added if actual superheat is more than 5°F above target (positive difference).

**For TXV's:** If available, determine the acceptable superheat range specified by the manufacturer. If not available, superheat values between 4°F and 25°F may be within an acceptable range. Since the purpose of a TXV is to maintain constant superheat, measuring superheat can be used to determine if the TXV is working properly as indicated in Section 3.4.

### **Subcooling**

Calculate Actual Subcooling as the condenser saturation temperature minus the liquid line temperature:

$$\text{Actual Subcooling} = T_{\text{cond, sat}} - T_{\text{liquid}}$$

Refer to manufacturer tables typically provided with the condensing unit to obtain Target Subcooling for systems with TXVs. Tests completed for the California Energy Commission suggest that subcooling as low as 2°F, and as high as 8°F over the manufacturer's target value, will not reduce normalized sensible EER by more than 5% (CEC 2011).

Systems with fixed orifice expansion devices do not typically provide subcooling targets, but values of  $10^\circ\text{F} \pm 5^\circ$  are typical. Subcooling decreases with increasing outdoor temperature and may fall to zero above 105°F. Conversely, at an outdoor temperature of 70°F, 25° of subcooling may be expected.

Calculate the difference between actual subcooling and target subcooling (Actual Subcooling – Target Subcooling). For systems with TXVs, the difference should not vary by more than  $\pm 3^\circ\text{F}$ , but greater variations are seen with fixed orifice systems. For TXV systems, refrigerant is normally removed if actual subcooling is more than 3°F above target (positive difference) and added if actual subcooling is more than 3°F below target (negative difference).

## **3.4 Diagnosis**

### ***Evaluating the Measurements***

To determine the nature of the defect requires a full diagnosis using COA, EST, superheat, and subcooling measurements. It is critical to first determine whether the system uses a TXV or fixed orifice metering device because different diagnostic procedures apply. All of the measurements listed below must be obtained before attempting to complete a diagnosis. Once a defect is identified and corrected, measurements should be repeated. For example, if the system is overcharged, the resulting high COA can make it difficult to detect other problems such as non-condensables or a liquid line restriction that would only be identified through retesting.

**Temperature Split.** Assuming that adequate airflow has been verified using direct flow measurements, a low temperature split (relative to the target value) indicates low capacity, which can be caused by one or more defects. Ideally, proper temperature split (within +/-3°F) should be further validated by checking refrigerant charge since proper temperature split can be found in systems that are overcharged or that contain non-condensables. Relatively low indoor wet bulb/dry bulb temperatures tend to elevate the temperature split and can conceal other problems, so measurements should ideally be made when temperatures are close to the AHRI rating points of 80°F dry bulb and 67°F wet bulb.

**Condensing Temperature Over Ambient (COA).** A high COA (over 30°F for 13 SEER and over 40°F for 10 SEER and lower) can indicate an overcharge condition, non-condensables in the refrigerant, or a liquid line restriction (typically a plugged filter-dryer). A high COA caused by non-condensables and/or a liquid line restriction will be accompanied by high subcooling. If subcooling is not too high, a drop in temperature may be observed across the restriction due to the drop in pressure. A low COA suggests an undercharge, or an expansion device overfeed (check for an improperly installed TXV bulb).

**Evaporator Saturation Temperature (EST or T<sub>evap, sat</sub>).** A low EST (less than 28°F) indicates either an undercharge or liquid line restriction or both. A low EST may also indicate a fouled evaporator coil. Excessive fouling would be detected during airflow tests, but a low EST may also be seen with minimal air blockage if dirt on the coil causes non-laminar airflow through the coil. A high EST (>45) will be seen at high outdoor temperatures (>100), with an overcharged system, and could also indicate an expansion device overfeed.

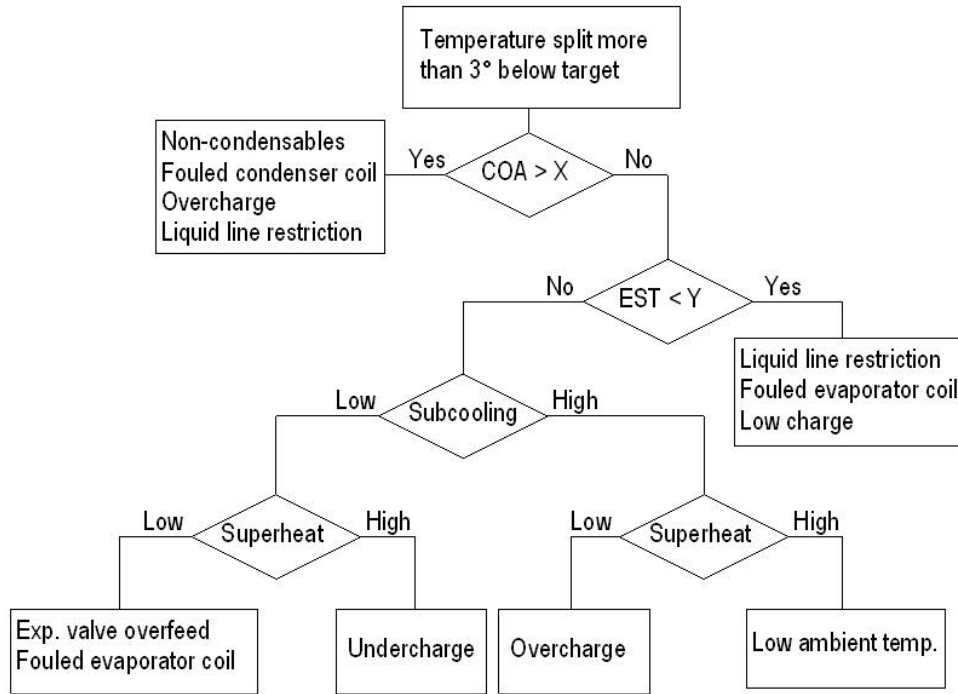
**Superheat and Subcooling.** Typically, low superheat and high subcooling indicate an overcharge, and refrigerant should be removed. Conversely, high superheat and low subcooling indicate an undercharge, and refrigerant should be added. High superheat coincident with high subcooling, and low superheat coincident with low subcooling are indicators of other problems.

**Using Diagnostic Flow Charts**

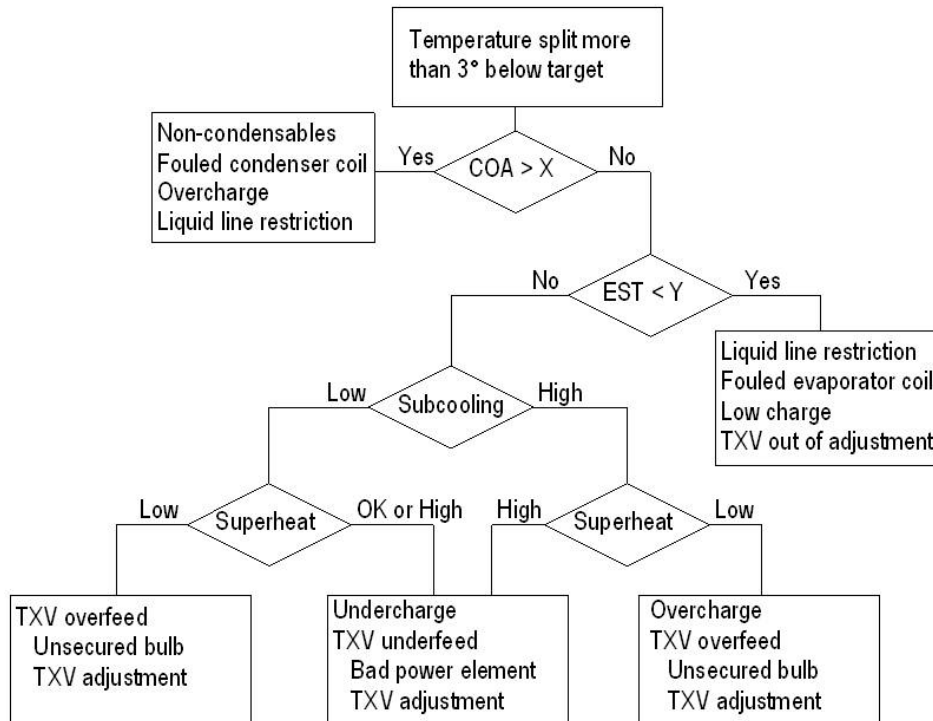
Figures 17 and 18 and Table 3 can be used to isolate which possible system defects are responsible for test results that deviate from targets. Use the chart that corresponds to the expansion device type and the appropriate values for X and Y from Table 3. Both the individual measurements and the charts should be used to guide the diagnostic process.

**Table 3. Recommended COA and EST Values to be Used in Diagnostics.**

System SEER	Maximum COA (X)	Indoor Wet Bulb Temp.	Minimum EST (Y)
>12	25	62-63	35
10-12	30	64-65	40
<10	35	66-67	45



**Figure 17. Refrigerant diagnostic flowchart: systems with fixed orifice expansion devices.**



**Figure 18. Refrigerant diagnostic flowchart: systems with thermostatic expansion devices.**

Correct diagnosis of refrigerant systems can require creative thinking. Since values of COA and EST vary with system efficiency and indoor wet bulb temperature, the values in Table 3 should be used as starting points for the quantities of X and Y in the flow chart, but it may be necessary

to modify these values. For example, an oversized evaporator coil will increase the load on the compressor and will tend to increase COA. A low outdoor temperature will lower COA, and a low indoor temperature will lower the EST.

Several of the defects listed in the flow diagrams can be checked by simple inspection, for example the TXV bulb attachment and coil fouling. If, after making these corrections and adjusting charge, the tested values continue to disagree with target values, defects including bad or out-of-adjustment TXV, non-condensables, and a liquid line restriction must be investigated. Diagnosis may require multiple passes through the flow chart and multiple corrections. Measurements and flow chart checks should always be repeated after correcting any faults or adjusting refrigerant charge until the measured values approach target values.

If a site glass is installed, this can be used as another diagnostic tool. Bubbles in the refrigerant are an indication of an undercharge, expansion device overfeed, or very high outdoor air temperature.

Liquid line restrictions are usually the result of a dirty filter-dryer or a plugged capillary tube, but the liquid line should also be inspected for kinks if this is suspected. Solid contaminants or moisture can cause a restriction in the filter-dryer.

A further test can be used to detect non-condensables. With the system running in a steady state condition, measure the temperature at a return bend about halfway up the condenser coil and convert this temperature to a pressure using the appropriate refrigerant table. If the measured high side pressure on the manifold gauge is significantly greater than the pressure corresponding to the condensing temperature, it is likely the system contains non-condensables and will need to be evacuated and recharged.

### **Removing and Replacing Refrigerant**

If non-condensables are detected or it is necessary to open the system to correct a defect such as a plugged filter-dryer, it is extremely important to use proper procedures for evacuation and recharging to avoid venting refrigerant into the atmosphere and to prevent the introduction of non-condensables and other contaminants into the refrigerant system. Detailed procedures for evacuation and recharging are provided in Attachment C. General procedures are as follows:

- Recover old refrigerant.
- Purge the system with nitrogen to remove moisture and check for leaks by holding the system at 150 psi for 10 minutes. If the system will not hold 150 psig, then locate and repair leaks.

#### **Vacuum Pump Maintenance is Important!**

*If vacuum pump oil isn't frequently changed, as after each use, it will lose the ability to reach the levels of vacuum that are necessary to remove non-condensable gases from the system.*

#### **Don't Mix Lubricants!**

*Lubricants used for R-22 systems and R-410A systems are not compatible. When upgrading systems to R-410A be careful to purge systems thoroughly. Using manifold gauges that are dedicated for R-22 and R-410A use will help avoid contamination and will have pressure ranges that are appropriate to the refrigerant type.*

- Evacuate the system to 240 microns. If pressure does not rise above 500 microns after 10 minutes, then the evacuation has been completed and the system is ready for charging with new refrigerant.
- Recharge using the weigh-in method. Follow up with superheat, subcooling, and final temperature split tests.

**Compressor Amps and Condenser Fan Amps (or Watts).** Measured amps (or watts) that are significantly higher than nameplate can be used to detect a failing compressor or condenser fan motor (motor winding or bearing failure). As a more detailed check, measure the current draw from the compressor only and compare to the manufacturer's compressor performance data to obtain the target amperage that corresponds to the conditions under which the current draw is measured. Excessive current draw can indicate an overcharge or non-condensables. When measured amps are lower than the manufacturer's listed value, causes could be an undercharge, liquid line restriction, or expansion valve failure.

### ***Equipment Replacement***

If the compressor or other components are found to be defective or ready for replacement by virtue of age, low efficiency or capacity, oversizing, or other reasons, manufacturers' installation instructions should be strictly followed for refrigerant charging and other installation procedures. Most likely, the indoor coil will require replacement because of the higher pressure ratings needed for R-410A.

Condensing units are typically provided with sufficient charge to accommodate 15–25 ft of refrigerant tubing. The line sets and indoor coil must be purged with nitrogen and evacuated to 240 microns of mercury and held for 10 minutes prior to opening service valves. Be sure to replace vacuum pump oil after each evacuation to eliminate contamination, which will damage the pump and prevent it from achieving the required 240 micron vacuum. When not in use, cover the vacuum pump exhaust port.

**Table 4. Target Superheat.**

		Return Air Wet-Bulb Temperature (°F)																										
		(T <sub>return,wb</sub> )																										
		50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76
Condenser Air Dry-Bulb Temperature (°F) (T <sub>condenser,db</sub> )	55	8.8	10.1	11.5	12.8	14.2	15.6	17.1	18.5	20.0	21.5	23.1	24.6	26.2	27.8	29.4	31.0	32.4	33.8	35.1	36.4	37.7	39.0	40.2	41.5	42.7	43.9	45.0
	56	8.6	9.9	11.2	12.6	14.0	15.4	16.8	18.2	19.7	21.2	22.7	24.2	25.7	27.3	28.9	30.5	31.8	33.2	34.6	35.9	37.2	38.5	39.7	41.0	42.2	43.4	44.6
	57	8.3	9.6	11.0	12.3	13.7	15.1	16.5	17.9	19.4	20.8	22.3	23.8	25.3	26.8	28.3	29.9	31.3	32.6	34.0	35.3	36.7	38.0	39.2	40.5	41.7	43.0	44.2
	58	7.9	9.3	10.6	12.0	13.4	14.8	16.2	17.6	19.0	20.4	21.9	23.3	24.8	26.3	27.8	29.3	30.7	32.1	33.5	34.8	36.1	37.5	38.7	40.0	41.3	42.5	43.7
	59	7.5	8.9	10.2	11.6	13.0	14.4	15.8	17.2	18.6	20.0	21.4	22.9	24.3	25.7	27.2	28.7	30.1	31.5	32.9	34.3	35.6	36.9	38.3	39.5	40.8	42.1	43.3
	60	7.0	8.4	9.8	11.2	12.6	14.0	15.4	16.8	18.2	19.6	21.0	22.4	23.8	25.2	26.6	28.1	29.6	31.0	32.4	33.7	35.1	36.4	37.8	39.1	40.4	41.6	42.9
	61	6.5	7.9	9.3	10.7	12.1	13.5	14.9	16.3	17.7	19.1	20.5	21.9	23.3	24.7	26.1	27.5	29.0	30.4	31.8	33.2	34.6	35.9	37.3	38.6	39.9	41.2	42.4
	62	6.0	7.4	8.8	10.2	11.7	13.1	14.5	15.9	17.3	18.7	20.1	21.4	22.8	24.2	25.5	27.0	28.4	29.9	31.3	32.7	34.1	35.4	36.8	38.1	39.4	40.7	42.0
	63	5.3	6.8	8.3	9.7	11.1	12.6	14.0	15.4	16.8	18.2	19.6	20.9	22.3	23.6	25.0	26.4	27.8	29.3	30.7	32.2	33.6	34.9	36.3	37.7	39.0	40.3	41.6
	64	-	6.1	7.6	9.1	10.6	12.0	13.5	14.9	16.3	17.7	19.0	20.4	21.7	23.1	24.4	25.8	27.3	28.7	30.2	31.6	33.0	34.4	35.8	37.2	38.5	39.9	41.2
	65	-	5.4	7.0	8.5	10.0	11.5	12.9	14.3	15.8	17.1	18.5	19.9	21.2	22.5	23.8	25.2	26.7	28.2	29.7	31.1	32.5	33.9	35.3	36.7	38.1	39.4	40.8
	66	-	-	6.3	7.8	9.3	10.8	12.3	13.8	15.2	16.6	18.0	19.3	20.7	22.0	23.2	24.6	26.1	27.6	29.1	30.6	32.0	33.4	34.9	36.3	37.6	39.0	40.4
	67	-	-	5.5	7.1	8.7	10.2	11.7	13.2	14.6	16.0	17.4	18.8	20.1	21.4	22.7	24.1	25.6	27.1	28.6	30.1	31.5	33.0	34.4	35.8	37.2	38.6	39.9
	68	-	-	-	6.3	8.0	9.5	11.1	12.6	14.0	15.5	16.8	18.2	19.5	20.8	22.1	23.5	25.0	26.5	28.0	29.5	31.0	32.5	33.9	35.3	36.8	38.1	39.5
	69	-	-	-	5.5	7.2	8.8	10.4	11.9	13.4	14.8	16.3	17.6	19.0	20.3	21.5	22.9	24.4	26.0	27.5	29.0	30.5	32.0	33.4	34.9	36.3	37.7	39.1
	70	-	-	-	-	6.4	8.1	9.7	11.2	12.7	14.2	15.7	17.0	18.4	19.7	20.9	22.3	23.9	25.4	27.0	28.5	30.0	31.5	33.0	34.4	35.9	37.3	38.7
	71	-	-	-	-	5.8	7.3	8.9	10.5	12.1	13.6	15.0	16.4	17.8	19.1	20.3	21.7	23.3	24.9	26.4	28.0	29.5	31.0	32.5	34.0	35.4	36.9	38.3
	72	-	-	-	-	-	6.4	8.1	9.8	11.4	12.9	14.4	15.8	17.2	18.5	19.7	21.2	22.8	24.3	25.9	27.4	29.0	30.5	32.0	33.5	35.0	36.5	37.9
	73	-	-	-	-	-	5.6	7.3	9.0	10.7	12.2	13.7	15.2	16.6	17.9	19.2	20.6	22.2	23.8	25.4	26.9	28.5	30.0	31.5	33.1	34.6	36.0	37.5
	74	-	-	-	-	-	-	6.5	8.2	9.9	11.5	13.1	14.5	15.9	17.3	18.6	20.0	21.6	23.2	24.8	26.4	28.0	29.5	31.1	32.6	34.1	35.6	37.1
	75	-	-	-	-	-	-	5.6	7.4	9.2	10.8	12.4	13.9	15.3	16.7	18.0	19.4	21.1	22.7	24.3	25.9	27.5	29.1	30.6	32.2	33.7	35.2	36.7
	76	-	-	-	-	-	-	-	6.6	8.4	10.1	11.7	13.2	14.7	16.1	17.4	18.9	20.5	22.1	23.8	25.4	27.0	28.6	30.1	31.7	33.3	34.8	36.3
	77	-	-	-	-	-	-	-	5.7	7.5	9.3	11.0	12.5	14.0	15.4	16.8	18.3	20.0	21.6	23.2	24.9	26.5	28.1	29.7	31.3	32.8	34.4	36.0
	78	-	-	-	-	-	-	-	-	6.7	8.5	10.2	11.8	13.4	14.8	16.2	17.7	19.4	21.1	22.7	24.4	26.0	27.6	29.2	30.8	32.4	34.0	35.6
	79	-	-	-	-	-	-	-	-	5.9	7.7	9.5	11.1	12.7	14.2	15.6	17.1	18.8	20.5	22.2	23.8	25.5	27.1	28.8	30.4	32.0	33.6	35.2
	80	-	-	-	-	-	-	-	-	-	6.9	8.7	10.4	12.0	13.5	15.0	16.6	18.3	20.0	21.7	23.3	25.0	26.7	28.3	29.9	31.6	33.2	34.8
	81	-	-	-	-	-	-	-	-	-	6.0	7.9	9.7	11.3	12.9	14.3	16.0	17.7	19.4	21.1	22.8	24.5	26.2	27.9	29.5	31.2	32.8	34.4
	82	-	-	-	-	-	-	-	-	-	5.2	7.1	8.9	10.6	12.2	13.7	15.4	17.2	18.9	20.6	22.3	24.0	25.7	27.4	29.1	30.7	32.4	34.0
	83	-	-	-	-	-	-	-	-	-	-	6.3	8.2	9.9	11.6	13.1	14.9	16.6	18.4	20.1	21.8	23.5	25.2	26.9	28.6	30.3	32.0	33.7
84	-	-	-	-	-	-	-	-	-	-	5.5	7.4	9.2	10.9	12.5	14.3	16.1	17.8	19.6	21.3	23.0	24.8	26.5	28.2	29.9	31.6	33.3	
85	-	-	-	-	-	-	-	-	-	-	6.6	8.5	10.3	11.9	13.7	15.5	17.3	19.0	20.8	22.6	24.3	26.0	27.8	29.5	31.2	32.9		
86	-	-	-	-	-	-	-	-	-	-	5.8	7.8	9.6	11.3	13.2	15.0	16.7	18.5	20.3	22.1	23.8	25.6	27.3	29.1	30.8	32.6		
87	-	-	-	-	-	-	-	-	-	-	-	5.0	7.0	8.9	10.6	12.6	14.4	16.2	18.0	19.8	21.6	23.4	25.1	26.9	28.7	30.4	32.2	
88	-	-	-	-	-	-	-	-	-	-	-	-	6.3	8.2	10.0	12.0	13.9	15.7	17.5	19.3	21.1	22.9	24.7	26.5	28.3	30.1	31.8	
89	-	-	-	-	-	-	-	-	-	-	-	-	5.5	7.5	9.4	11.5	13.3	15.1	17.0	18.8	20.6	22.4	24.3	26.1	27.9	29.7	31.5	

Shaded area requires return plenum temperature of 70°F or higher.

Table 4 (continued). Target Superheat.

		Return Air Wet-Bulb Temperature (°F)																										
		(T <sub>return, wb</sub> )																										
		50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76
Condenser Air Dry-Bulb Temperature (°F) (T <sub>condensat, db</sub> )	90	-	-	-	-	-	-	-	-	-	-	-	-	-	6.8	8.8	10.9	12.8	14.6	16.5	18.3	20.1	22.0	23.8	25.6	27.5	29.3	31.1
	91	-	-	-	-	-	-	-	-	-	-	-	-	-	6.1	8.1	10.3	12.2	14.1	15.9	17.8	19.7	21.5	23.4	25.2	27.1	28.9	30.8
	92	-	-	-	-	-	-	-	-	-	-	-	-	-	5.4	7.5	9.8	11.7	13.5	15.4	17.3	19.2	21.1	22.9	24.8	26.7	28.5	30.4
	93	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.8	9.2	11.1	13.0	14.9	16.8	18.7	20.6	22.5	24.4	26.3	28.2	30.1
	94	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.2	8.7	10.6	12.5	14.4	16.3	18.2	20.2	22.1	24.0	25.9	27.8	29.7
	95	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.6	8.1	10.0	12.0	13.9	15.8	17.8	19.7	21.6	23.6	25.5	27.4	29.4
	96	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	7.5	9.5	11.4	13.4	15.3	17.3	19.2	21.2	23.2	25.1	27.1	29.0
	97	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	7.0	8.9	10.9	12.9	14.9	16.8	18.8	20.8	22.7	24.7	26.7	28.7
	98	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.4	8.4	10.4	12.4	14.4	16.4	18.3	20.3	22.3	24.3	26.3	28.3
	99	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.8	7.9	9.9	11.9	13.9	15.9	17.9	19.9	21.9	24.0	26.0	28.0
	100	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.3	7.3	9.3	11.4	13.4	15.4	17.5	19.5	21.5	23.6	25.6	27.7
	101	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.8	8.8	10.9	12.9	15.0	17.0	19.1	21.1	23.2	25.3	27.3
	102	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.2	8.3	10.4	12.4	14.5	16.6	18.6	20.7	22.8	24.9	27.0
	103	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.7	7.8	9.9	11.9	14.0	16.1	18.2	20.3	22.4	24.5	26.7
	104	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.2	7.2	9.3	11.5	13.6	15.7	17.8	19.9	22.1	24.2	26.3
	105	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.7	8.8	11.0	13.1	15.2	17.4	19.5	21.7	23.8	26.0
	106	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.2	8.3	10.5	12.6	14.8	17.0	19.1	21.3	23.5	25.7
	107	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.7	7.9	10.0	12.2	14.4	16.6	18.7	21.0	23.2	25.4
	108	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.2	7.4	9.5	11.7	13.9	16.1	18.4	20.6	22.8	25.1
	109	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.9	9.1	11.3	13.5	15.7	18.0	20.2	22.5	24.7
110	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.4	8.6	10.8	13.1	15.3	17.6	19.9	22.1	24.4	
111	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.9	8.1	10.4	12.6	14.9	17.2	19.5	21.8	24.1	
112	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	5.4	7.6	9.9	12.2	14.5	16.8	19.1	21.5	23.8	
113	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	7.2	9.5	11.8	14.1	16.4	18.8	21.1	23.5	
114	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.7	9.0	11.4	13.7	16.1	18.4	20.8	23.2	
115	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	6.2	8.6	10.9	13.3	15.7	18.1	20.5	22.9	

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## Appendix A: Test Forms and Checklists

Table A-1. Preliminary Diagnostics Form.

Site Address _____		Checked by _____	Date _____
TEST/OBSERVATION	CRITERIA	RESULTS/NOTES	
Maintenance Problems	Yes or No, describe if yes		
Comfort Problems	Yes or No, describe if yes		
System Age / Condition	Years / good, fair, poor		
Condensing Unit Make/Model	From nameplate		
Condenser Size	kBtuh/12, from nameplate	Tons	
Condenser Coil Condition	Good, or Fouled and Cleaned		
Number of Returns	1 or _____		
Number of Zones	1 or _____		
Bypass Damper Installed	If system is zoned, yes or no		
Filter Location	In furnace / at return grille / other		
Filter Condition	New / Fair / Clogged / Missing		
Suction Line Insulation/Location	Condition / Attic, Garage, Other		
TXV bulb secured and insulated	No, Yes, or Repaired		
Minimum Required Airflow	= 300 x Tons	CFM	
Filter Size	= L _____" x W _____" /144	ft <sup>2</sup>	
Filter Velocity	= Minimum Airflow / Filter Size	FPM (<300)	
Normal System Op. Pres. (NSOP)	<input type="checkbox"/> with filter <input type="checkbox"/> without filter	Pa	
Airflow at All Registers?	Yes, or No and specify		
Measured Duct Leakage	If DTS performed (optional)	CFM	
Measured Airflow	Method:	CFM	
Supply air dry bulb temperature	Measured at supply plenum (S)	Deg. F	
Return air dry bulb temperature	Measured at return plenum (R)	Deg. F	
Return air wet bulb temperature	Measured at return air intake	Deg. F	
Temperature Split.	= R – S	Deg. F	
Target Temperature Split	From Table 2	Deg. F	
Measured Airflow with Filter	Must be greater than Min. Required	CFM	
Measured Airflow without Filter	If first measurement <300 CFM	CFM	
Test Pressures. Refer to PD Step 10 for instructions.	Supply – Return with filter (<190 Pa)	_____ Pa	
	Supply – Return w/o filter (<140 Pa)	_____ Pa	
	Supply – Ambient (< 40 Pa)	_____ Pa	
	Supply – Ambient (< 40 Pa)	_____ Pa	
<b>Results and Actions</b> <input type="checkbox"/> Temperature split within 3°F of target <input type="checkbox"/> Problem(s) identified: (describe) <input type="checkbox"/> Airflow adequate with filter in place <input type="checkbox"/> Referred to HVAC tech <input type="checkbox"/> Airflow adequate only with filter removed <input type="checkbox"/> Repairs made: (describe)			

**Table A-2. Comprehensive Diagnostics Form (supplement to Table A-1).**

Site Address	Technician	Date
<b>TEST/OBSERVATION</b>	<b>CRITERIA</b>	<b>RESULTS/NOTES</b>
Return air dry bulb temperature	T <sub>return, db</sub>	°F
Return air web bulb temperature	T <sub>return, wb</sub>	°F
Supply air dry bulb temperature	T <sub>supply, db</sub>	°F
Evaporator saturation temperature	T <sub>evap, sat</sub>	°F
Condenser saturation temperature	T <sub>cond, sat</sub>	°F
Suction line temperature	T <sub>suction</sub>	°F
Liquid line temperature	T <sub>liquid</sub>	°F
Condenser entering air dry bulb	T <sub>cond, db (measure average)</sub>	°F
Compressor amps, measured		Amps
Compressor amps, rated	From condensing unit nameplate	Amps
Temperature Split	From Table A-1	°F
Condensing over Ambient	= T <sub>cond, sat</sub> – T <sub>cond, db</sub>	
Superheat Measured	T <sub>sh, meas</sub> = T <sub>suction</sub> – T <sub>evap, sat</sub>	
Superheat Target	T <sub>sh, target</sub> (from Table 4)	
Superheat Deviation	= T <sub>sh, meas</sub> – T <sub>sh, target</sub>	
Subcooling Measured	T <sub>sc, meas</sub> = T <sub>evap, sat</sub> – T <sub>liquid</sub>	
Subcooling Target	T <sub>sc, target</sub> (from manufacturer tables, or use 10°F ±5°)	
Subcooling Deviation	= T <sub>sc, meas</sub> – T <sub>sc, target</sub>	
Diagnosis and Actions	<input type="checkbox"/> Undercharge <input type="checkbox"/> Overcharge <input type="checkbox"/> Liquid line restriction <input type="checkbox"/> Non-condensables <input type="checkbox"/> Faulty TXV or bulb installation <input type="checkbox"/> Compressor amps excessive <input type="checkbox"/> Repairs made: (describe)	

## Appendix B: Supporting Data and Calculations

### B-1: Energy Savings for Replacement with High Efficiency Air Conditioners

This section describes modeling and calculations that were completed to estimate the energy savings and incremental costs for replacement of existing air conditioners with systems that have performance ratings higher than the current standard of SEER 13. BEopt was used to model existing homes in five U.S. climate zones. The foundation was assumed to be slab-on-grade for all zones. Building geometry and configuration were determined based on research and review of the 2010 DOE Buildings Energy Data Book (BEDB) (DOE, 2011) and the 2009 California Residential Appliance Saturation Study (RASS) (KEMA, 2010) and include the following:

- Single story 1,800 square ft home with three bedrooms. This is consistent with RASS and BEDB averages. RASS average for single family homes is 1,882 ft<sup>2</sup> for CA building stock. BEDB estimates characteristics of a typical single-family home from the mid 1970s with a heated average floor space of 1,934 ft<sup>2</sup> and a cooled average floor space of 1,495 ft<sup>2</sup>. Average conditioned floor area of homes constructed between 1980 and 1989 is 1,806 ft<sup>2</sup>.
- 16% glass as a percentage of conditioned floor area, based on RASS averages. The BEDB reports window areas for a typical 1970s-era home of between 11% and 15% of conditioned floor area.
- Glass is equally distributed on all four sides to eliminate orientation effects
- Wall area and slab perimeter based on an aspect ratio of 2.0. This is consistent with a sampling of actual homes. To maintain equal wall and glazing areas distribution on all orientations, an L-shape building model was used.

The home and its existing air conditioner were assumed to be from the 1980’s era. Building characteristic assumptions are based on the Building America House Simulation Protocols (HSP) as well as California’s Title 24 vintage assumptions, where appropriate. U-factor and solar heat gain coefficient (SHGC) values for the single metal pane windows were taken from the default values in Tables 116-A and 116-B in California’s 2008 Building Energy Efficiency Standards (CEC 2010).

**Table B-1. Existing home assumptions by building vintage**

	<b>1980-1989</b>
Wall Insulation	R-11
Window U-value/SHGC	Single metal 1.28/0.80
Slab Insulation R-Value and Depth	R-0
Ceiling Insulation	R-9

Title-24 default SEER and EER assumptions were used for the base case air conditioner for units installed between 1984 and 1991; SEER 8.9, EER 7.8. Fan watt draw was estimated at 0.58 W/cfm. Ductwork was assumed to be located in the attic with R-2.1 insulation and 30% leakage. Results of simulations for each climate zone/city are listed in Table B-2. These values were used

to calculate energy savings. A flat rate of \$0.12 per kWh was used to calculate utility bill savings.

**Table B-2. Annual Energy Use Determined by BEopt.**

Climate Zone	Sample City	Annual Cooling Energy Use (kWh)			
		Base Case	SEER 13, EER 11.1	SEER 15, EER 12.7	SEER 18, EER 13.4
Hot-Dry	Fresno, CA	7,435	5,304	4,619	4,866
Hot-Humid	Houston, TX	8,356	6,247	5,456	5,833
Mixed-Humid	Atlanta, GA	5,343	3,823	3,332	3,547
Marine	Santa Rosa, CA	1,009	684	593	645
Cold	Denver, CO	2,677	1,885	1,640	1,850

Air conditioner sizes for the existing and replacement systems that were used for cost estimating are listed in Table B-3. Average installed costs of \$130, \$140, and \$160 per kBtuh for the 13, 15, and 18 SEER systems, respectively, were obtained from the National Residential Energy Efficiency Measures Database at: <http://www.nrel.gov/ap/retrofits/index.cfm>. These were used to calculate the incremental costs for 15 and 18 SEER replacements that are listed in Table 1.

The higher energy use for the SEER 18 system reported in Table B-2 is a result of the performance curve utilized by BEopt for two-speed condensing units. Some high SEER products perform better on average, but when operated only at high speed, perform only marginally better than SEER 13 systems.

**Table B-3. Assumed Sizes of Replacement Air Conditioners.**

Climate Zone	A/C Size
Hot-Dry	3.0
Hot-Humid	3.0
Mixed-Humid	2.5
Marine	1.5
Cold	2.0

**Table B-4. Replacement Cost vs. SEER Rating.**

Replacement Cost	13	15	18
Hot-Dry	\$4,680	\$5,040	\$5,760
Hot-Humid	\$4,680	\$5,040	\$5,760
Mixed-Humid	\$3,900	\$4,200	\$4,800
Marine	\$2,340	\$2,520	\$2,880
Cold	\$3,120	\$3,360	\$3,840

It should be noted that the energy use value of 7,435 kWh/year determined using BEopt for the Fresno climate is more than six times higher than California Appliance Saturation Survey data

(RASS 2009), which cites a unit energy consumption value of 1,189 kWh/year. Further research is needed to determine the source of this discrepancy.

**B-2: Replacement Blower Motor Energy Savings Estimates**

To measure fan energy savings for an ECM motor replacement, a 1970’s vintage GE Model BLU100C942A1 furnace was tested first with its stock motor and subsequently with an Evergreen 1/3-1/2 HP replacement motor. To simulate different supply duct static pressure conditions, each motor was tested using three different “equivalent” supply restrictions (A, B, and C) that yielded corresponding plenum pressures of 100, 150, and 200 Pa at the ECM motor’s highest tap setting. A TrueFlow grid was used to measure airflow and served as a proxy for the filter and return plenum/grille airflow restriction. The airflow produced by this furnace is suitable for a 3-ton air conditioner if the ducts are of adequate size, but since typical supply plenum pressures are close to 0.5” (125 Pa), the furnace is more appropriate for a 2-1/2 ton system. Test results are shown in Table B-5. Of the three PSC motor tap settings, the Medium setting provided the closest correspondence to the ECM’s High setting, and the PSC’s Medium setting corresponded to the ECM’s Med-Low tap setting, so these pairs were used to estimate energy savings.

Curve fits of supply plenum pressure vs. power were used to develop equations that allow calculation of energy savings as a function of heating and cooling supply plenum pressures (representing a range of duct systems) and annual operating hours. The curve fits were used to develop a spreadsheet that can be used to test different operating assumptions.

**Table B-5. Results of Comparative Tests of a PSC and ECM Motor.**

	Target Pressure	“Duct” Setting	Measured Pressure, Pa	Airflow CFM	Power Watts
PSC, "Cooling Mode" High tap setting	100	A	100	1255	830
	150	B	150	1037	762
	200	C	200	819	705
PSC, "Cooling Mode" Medium tap setting		A	90	1182	683
		B	139	995	591
		C	194	788	524
PSC, "Heating Mode" Low tap setting		A	57	950	580
		B	106	865	524
		C	162	719	458
ECM, "Cooling Mode" Evergreen, "High" tap settings		A	87	1152	492
		B	143	991	483
		C	194	803	437
ECM, "Heating Mode" Evergreen, "Med-Low" tap setting		A	61	960	284
		B	107	852	305
		C	163	729	324

For the example provided in Table B-6, run times were obtained from an eQuest simulation of a typical house located in Fresno (hot-dry climate). At \$0.17 per kWh, annual savings were calculated to be about \$30. If financed under a 15-year loan at 5%, these savings would offset an expenditure of \$130, which would not cover the cost of the motor and labor for replacement unless the original motor was defective. Once a contractor has become familiar with the replacement process, the time to complete the work is one-half to one hour, depending on site conditions.

Motor operating hours are the most critical factor in determining savings, and are a function of the house heating and cooling loads and system capacities. When houses are tightened up and insulation is improved, run times will be reduced, diminishing savings from fan motor replacement.

**Table B-6. Annual Cost Savings Calculation Example.**

Measured Plenum Pressure, Heating	100	Pa
Annual Heating Hours	397	Assume 25% run time, Nov-Feb
Watt Savings-Heating	230.3	Watts
Annual Heating Savings	91.4	kWh
Measured Plenum Pressure, Cooling	150	Pa
Annual Cooling Hours	798	Assume 10% run time, June-Sept
Watt Savings-Cooling	104.1	Watts
Annual Cooling Savings	83.0	kWh
Total Savings	174.5	kWh
Mean Tariff	\$0.17	per kWh
Cost Savings	\$ 29.66	Annual Savings

Genteq provides an Energy Savings Calculator at: <http://www.genteqcomfort.com/ShowMeTheMoney/EnergySavingsCalculator.aspx>.

Inputs are location (city/state) and motor size. Entering Stockton, California, and ½ hp yielded savings of \$92 and much higher run times of 1158 hours (cooling) and 1834 hours (heating).

### **B-3: Verification of Temperature Split Method for Checking Refrigerant Charge**

The California Energy Commission’s Title 24 energy standards include an alternate method for verifying airflow using the temperature difference between the supply and return plenums. This “temperature split” method is used to confirm the validity of superheat or subcooling refrigerant charge tests, specifically to insure that low airflows are not producing erroneous charge measurements.

Most home performance contractors (HPCs) carry duct blasters and can use them to accurately measure airflow but are not equipped to measure or correct refrigerant charge. Consequently, the PD method described in this guideline provides for HPCs to verify airflow and use temperature split to detect significant refrigerant-side problems. Low refrigerant charge, liquid line restrictions, expansion valve defects, and the presence of excessive non-condensable gases can all affect cooling capacity, which in turn will proportionally reduce the temperature split.

The Target Temperature Split table (Table 2) in the Implementation section of this guideline is borrowed from Title 24 compliance documents (CEC 2008). To verify the values in the



temperature split table, Robert Mowris of Verified Inc. commissioned Intertek to test the method using one properly charged air conditioner operated under a variety of conditions. Table B-5 compares the actual measured temperature split (TS) with values in Table 2 (Required TS) using fixed airflow and return air dry bulb (RDB) temperatures, and varying return air wet bulb (RWB) and condenser entering air temperatures. The greatest difference between target and measured splits was 3.0°F (Run 418), which is the same deviation allowed under the Title 24 tests. These results suggest that a temperature split greater than 3°F provides sufficient evidence that the capacity may be compromised by a refrigerant side defect.

**Table B-5. Laboratory Test Results Comparing Measured and Target Temperature Splits.**

Intertek Run	Actual TS (°F)	Req'd TS (°F)	RDB (° F)	RWB (° F)	SDB (° F)	SWB (° F)	Return Wet Bulb and Return Dry Bulb Temperature (°F)	Condenser Entering Air Temperature (°F)	Evaporator Airflow CFM/ton	EER* <sub>A</sub>
412-0	19.0	18.3	80.0	66.9	61.0	58.5	80/67	95	400.0	9.99
412	21.3	21.3	79.9	62.9	58.6	55.3	80/63	115	400.1	6.92
414	21.9	21.3	80.1	63.1	58.2	54.8	80/63	100	399.8	8.74
416	21.9	21.3	80.0	63.1	58.1	54.2	80/63	85	399.9	10.57
418	21.6	24.6	80.0	57.0	58.4	48.4	80/57	115	400.7	6.96
420	23.4	24.6	80.0	56.9	56.6	47.4	80/57	95	400.4	9.26
422	23.8	24.6	80.0	57.1	56.2	47.5	80/57	75	400.4	11.33

## Appendix C: Detailed Methods and Procedures

### C-1: Air Conditioner Replacement Decision Tree

The following diagram can be used to guide decisions on when to consider replacing an older air conditioner (PNNL 2011). This diagram does not address changes in sizing caused by improvements to the building enclosure or air conditioners that were incorrectly sized when originally installed. Manual J and S calculations should be completed prior to selecting equipment for replacement.

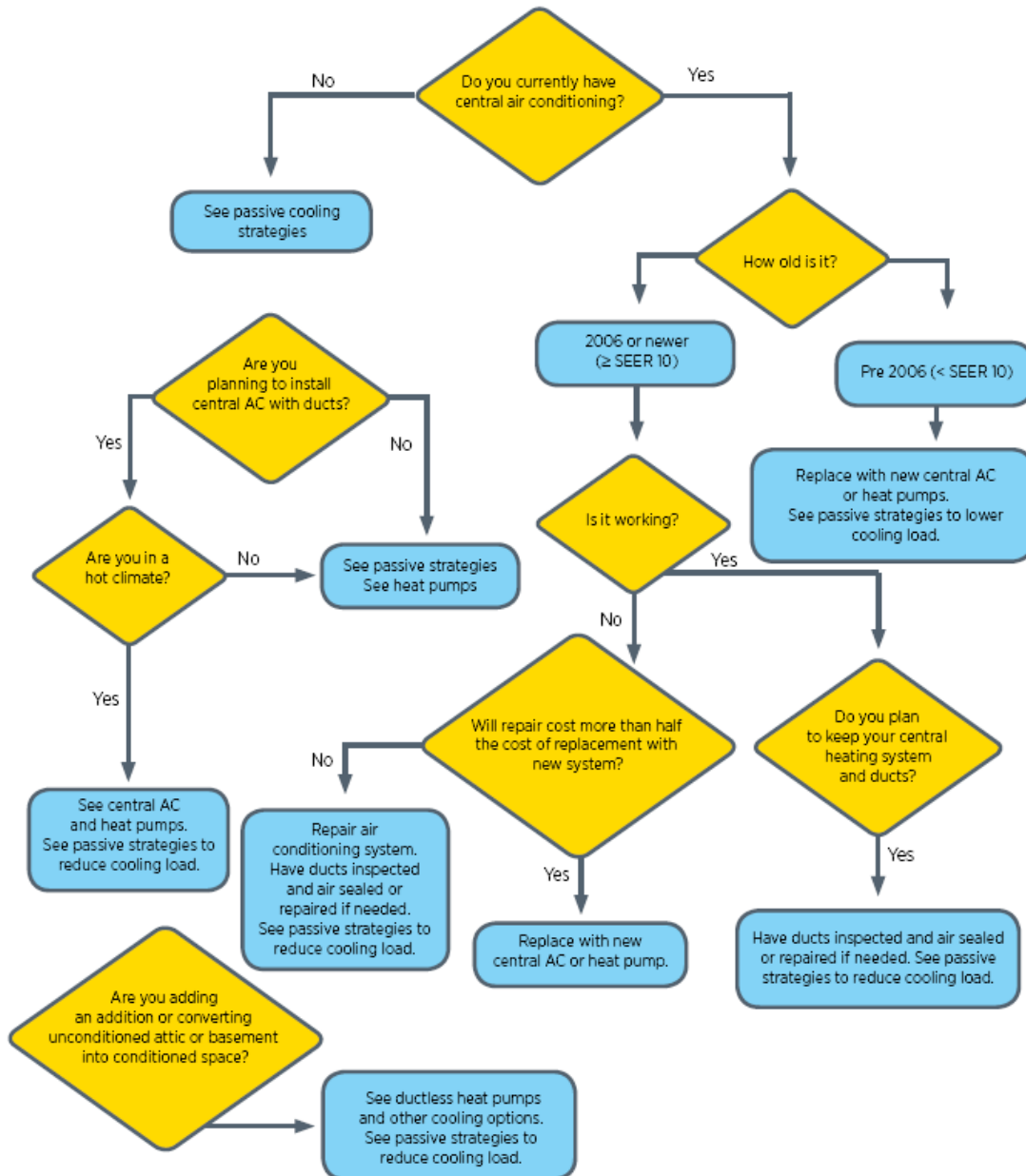


Figure C-1. Air conditioner replacement decision tree.

## C-2: Detailed Airflow Measurement Methods

The following methods were excerpted from Appendix RA-3 of the California Title 24 Residential Energy Standards (CEC 2008). Modifications were made to the method to allow measurement of airflow with and without the filter in place.

### General Instructions

Begin tests with the filter installed. If the measured airflow is less than 300 cfm per ton of AC, repeat the measurements with the filter removed. Use either Table A-1 or Table A-2 to record the results

Airflow tests must be conducted with the fan operating at the speed used for AC (cooling speed). The fan switch on newer systems provides an airflow lower than that used for cooling, in which case the system must be set to operate in cooling mode by lowering the thermostat setting. The condenser may be disabled by pulling the fuses if necessary to avoid over-cooling the space.

For multizone systems, all testing must be performed with all zone dampers open and bypass dampers locked in the closed position. In all cases, refer to manufacture instructions for the particular test equipment used for additional information.

### Powered Flow Hood (Fan Flowmeter)

This fan flow measurement shall be performed using the following procedures:

1. With the system fan on at the maximum speed (the cooling speed), measure the pressure difference (in Pa) between the supply plenum and the conditioned space (Psp). Psp is the target pressure to be maintained during the fan flow tests. Place the pressure probe in the Supply Pressure Measurement Location marked “S” in Figure B-1. Adjust the probe to achieve the highest pressure and then firmly attach the probe to ensure that it does not move during the fan flow test.

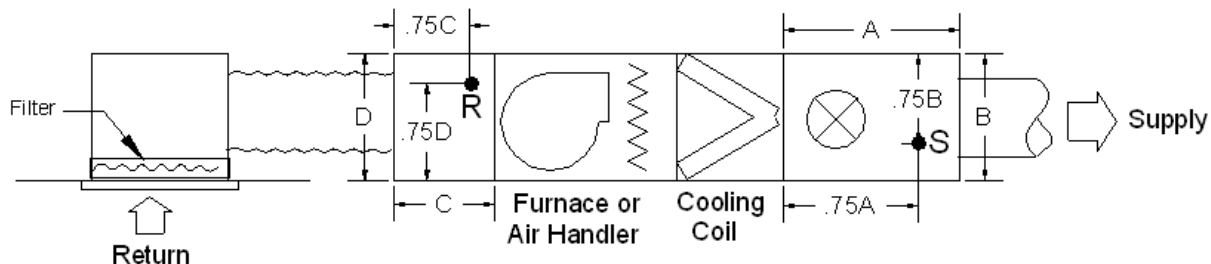


Figure C-2. Pressure measurement locations.

2. Attach the fan flowmeter to the duct system at the main (largest) return air grille. If there are two or more returns, seal off the other return grilles.
3. Turn on the system fan and the fan flowmeter, adjusting the fan flowmeter until the pressure between supply plenum and conditioned space matches Psp.
4. Record the flow through the flowmeter (Qah, cfm); this is the diagnostic fan flow. In some systems, system fan and fan flowmeter combinations may not be able to produce enough flow to reach Psp. In this case, record the flow (Qah, cfm) and pressure (Pmax)

between the supply plenum and the conditioned space. Use Pmax and Psp with the following equation to correct measured system flow (Qah<sub>corrected</sub>).

$$Qah_{corrected} = Qah \times (Psp/Pmax)^{0.5}$$

### **Flow Grid**

The fan flow measurement shall be performed using the following procedures:

1. With the system fan on at the maximum speed (the cooling speed), measure the pressure difference (in Pascal) between the supply plenum and the conditioned space (Psp). Place the pressure probe in the Supply Pressure Measurement Location marked “S” in Figure B-1. Adjust the probe to achieve the highest pressure and then firmly attach the probe to ensure that it does not move during the fan flow test.
2. Remove the filter and replace it with a flow grid that corresponds to the filter size.
3. Re-measure the system operating pressure with the flow grid in place.
4. Measure the airflow through the flow grid (Qah) and the test pressure (Ptest).
5. Use the following equation to correct flow through the flow grid and pressure (Qah and Ptest) to operating condition at operating pressure (Psp).

$$Qah_{corrected} = Qah \times (Psp/Ptest)^{0.5}$$

### **Flow Capture Hood**

If neither a powered flow hood or a flow grid are available, a calibrated flow capture hood (balometer) may be used, though accuracy will vary greatly depending on the type and orientation of the device used. With the system fan on at the maximum speed (the cooling speed), use the flow hood to measure airflow at each return grille. Sum the measured return airflows to determine the system fan flow (Qah, cfm). To improve accuracy of balometer measurements, the same correction may be applied as used with the flow grid.

### C-3: Detailed Condenser Coil Cleaning Instructions

Accumulated dust, lint, leaves, grass clippings, and other debris can be sucked into the fins of a condensing unit coil as air is drawn through them by the fan. These obstructions will lower the effectiveness of the coil, elevate condensing temperature, and reduce cooling efficiency. A visual inspection is generally adequate to determine whether the coil must be cleaned, though cleaning a relatively unobstructed coil will likely improve efficiency.

1. If a digital thermometer with a surface probe is available, measuring the temperature of the liquid line (the small uninsulated line) before and after the cleaning process will provide some indication of the effectiveness of the cleaning process.
2. Biodegradable coil cleaners can be obtained from HVAC supply houses and will reduce the cleaning time and improve the effectiveness of the cleaning process. A hose with a spray nozzle or a pressure washer are also required. If using a pressure washer, care must be exercised to insure the water stream is not so powerful as to bend or damage the fins of the coil. Follow the instructions listed below:
3. Shut off the cooling system at the thermostat and pull the fuse from the panel that powers the outdoor unit.
4. If the exterior of the coil is conspicuously covered with litter, use a shop vacuum to remove it, being careful not to bend the fins.
5. Carefully remove the top of the cabinet. As shown in Figure B-1, the fan motor is sometimes attached to the top. In this case, set the fan-motor assembly aside and prop it up as needed to avoid putting strain on the wires.
6. Spray the coil from the inside and outside first using water, then using the coil cleaner and allow it to sit for 5-10 minutes, or as directed by the cleaning solution instructions.
7. Use the spray nozzle or pressure washer to flush the coil until clean.
8. Replace the top and restore the fuse disconnect.



**Figure C-3. Removing of the top of the condensing unit for cleaning (Source: thefamilyhandyman.com, reprinted with permission).**

Note that some two-row condenser coils are split and curved in a “U” shape and the fins are separate and do not align, causing clogging between the two rows. In this case, it may be necessary to separate the coils for proper cleaning, a job that requires the skills of a qualified HVAC technician.

#### **C-4: Refrigerant Procedures to Prevent Non-Condensables and Restrictions**

To reduce the introduction of contaminants while lines are being brazed, it is recommended practice to leave one end of the line open and slowly introduce nitrogen into the line, allowing it to flow out the other open end. The nitrogen will prevent oxidation and contamination resulting from brazing by-products. Most manufacturers recommend installation of a filter drier in “all field-connected split-system air conditioners” (Carrier). The filter drier should be installed on the liquid line before the expansion device to remove moisture, acid, sludge, and particulate material from the refrigerant system and to avoid refrigerant restrictions. Filter driers should not be exposed to ambient air for more than a few minutes or else they will be contaminated and will lose their effectiveness. A deep vacuum will not remove moisture from a filter drier. Proper charging procedures (see Section C-8) must be used to clear lines of nitrogen and to avoid introducing air into the system. Air contains moisture, which can create oil sludge that reduces the lubricating properties of the oil, and plugs screens and oil passages in the compressor. Moisture can also damage the synthetic POE oil used in R410A and can freeze in the expansion device to cause restrictions. The heat of compression causes moisture to react with refrigerant and form hydrochloric and hydrofluoric acid. These acids erode machine surfaces and cause copper ions from the tubing to deposit on heated surfaces in the compressor causing compressor failure. These contaminants can cause the insulation on the motor windings to break down.

#### **C-5: Vacuum Pump Maintenance**

A common installation practice is to attach the vacuum pump to the system and allow it to run for a given time period, typically 4 hours. After that time, there is an assumption that a sufficient vacuum has been pulled to remove any non-condensables from the system. No vacuum gauge is installed, nor reference to potential vacuum level or vacuum change when the vacuum pump is turned off as specified in manufacturers' installation literature. While common, this is a problematic approach as vacuum pumps rarely receive recommended maintenance and cannot be guaranteed to produce the required system evacuation.

Manufacturers indicate that vacuum pump maintenance is required after each evacuation (JB Industries, 2007). Required maintenance includes draining contaminated oil while the vacuum pump crankcase is still warm from use, flushing the reservoir, refilling with new oil, and recycling old oil. New oil should not be exposed to the atmosphere for longer than two minutes because it will adsorb atmospheric moisture, causing loss of lubrication and sealing ability. New oil should be run in the pump for two or three minutes and then drained and refilled with fresh oil. This will remove contaminants from the pump and the oil. The vacuum pump must be stored with the exhaust port plugged or capped to prevent contamination due to air, water vapor, and dirt. A slight cut or dirt on the o-ring seals can cause leaks. The mating flare fitting faces must be wiped after each use and checked for damage before hookup. Vacuum pump oil creates a fine lubricant at these connections. Without this maintenance, the vacuum pump oil can react with refrigerants and produce hydrofluoric and hydrochloric acids. These acids can, over time, corrode and damage internal surfaces of the pump making it impossible to achieve the required vacuum for system installation.

Vacuum pump oil serves three functions: 1) coolant, 2) lubricant, and 3) sealant. If the oil is contaminated, then the vapor pressure of the oil will increase which causes the pump to not achieve proper evacuation. A new pump with clean oil can achieve a 15 to 50 micron vacuum.

Putting the vacuum gauge on the pump and capping the other fittings will tell whether or not the oil is contaminated or the pump is not working. If the pump cannot achieve 50 microns, then either the oil is contaminated or the pump is not working properly. It is difficult to visually determine if oil is contaminated with air or moisture, so manufacturers recommend draining, flushing and refilling with new oil after every use. Some contaminated systems or larger systems require from two to five oil changes during the evacuation process.

The vacuum pump is only as good as the oil that can maintain the correct vapor pressure. Vacuum pump oil will not introduce contamination to the system. However, contaminated oil will prevent the vacuum pump from achieving the 240 micron vacuum. The *2010 ASHRAE Handbook - Refrigerant* specifies a 240 micron vacuum for AC systems (some manufacturers recommend 240 microns for R410A and 500 microns for R22.). In order for the pump to pull a 240 to 500 micron vacuum, the oil must be clean and moisture-free throughout evacuation. At the beginning of the evacuation procedure, water vapor is removed quickly. If the system contains moisture, the oil will get contaminated quickly and this will make it difficult to achieve the correct vacuum. At the beginning of the evacuation, the gas ballast valve should be opened to remove moisture from the system and avoid contaminating the vacuum pump oil. After the vacuum pump quiets down and the micron gauge reads 1500 microns or less, the gas ballast valve should be closed.

A slight cut or dirt in the o-ring seals can cause leaks, therefore, the mating flare fittings faces should be wiped and checked for damage before hook up. It is important that evacuation always be performed from both the low and high sides of the system since this can save as much as 75% of the time to perform the evacuation compared to evacuating from only one side.

### **C-6: Evacuation and Recovery Procedures**

Proper recovery and evacuation procedures are required to avoid introducing non-condensables or restrictions to the AC system. To speed the evacuation process, it is necessary to remove the Schrader valve cores to eliminate the restriction they cause. However, the valve core depressors in the gauge hose connectors also cause a high restriction. Either remove the valve core depressors from the hose connectors, or use gauge hoses without valve core depressors during the evacuation process and change to regular hoses with EPA low loss fittings after the Schrader valve cores are replaced. An added restriction is also imposed by three-valve manifolds with 1/4" hose connections to the vacuum pump. For rapid evacuation, it is highly recommended that the service tech include (among other tools and equipment) a four-valve gauge manifold with a 3/8 in. hose to the vacuum pump, a separate hose to the refrigerant cylinder, and hoses with either no valve core depressors. The four-valve manifold setup can be used exclusively for the evacuation process.

Never pull a vacuum through Schrader valves. This practice will increase the evacuation time due to the high restriction it will cause.

The following procedures are recommended to perform refrigerant recovery:

1. The EPA prohibition on venting requires that the following procedures be followed when connecting or disconnecting refrigerant hoses (USEPA 1970):

- Hoses must be equipped with EPA low-loss fittings.
  - Attach center hose from manifold to recovery unit
2. Install a filter-drier in the vapor line inlet of recovery unit to protect compressor and CRO valve. Connect another hose from recovery unit to recovery tank.
  3. Open both valves on manifold and the valve on the recovery tank with the hose connected to it. Turn on the recovery unit and move valves on the recovery unit to recover refrigerant. Wait until the low-side gauge on the manifold reads 0 psig (0 inches of Hg) or slightly less than ambient pressure. Then close all valves on manifold and recovery unit. Shut off recovery unit.

After a motor burn-out, the refrigerant should never be reused and all recovered refrigerant must comply with ARI 700.

If it is necessary to run the vacuum pump overnight, it is highly recommended to use a full-port solenoid valve in-line with the vacuum line to the pump to avoid losing vacuum during a power failure. If loss of power occurs during the evacuation, the oil in the pump and non condensables will be drawn back into the system. Never base evacuation on time. Always measure the quality of the vacuum with a calibrated micron gauge.

The triple evacuation or deep vacuum methods are both acceptable evacuation methods and are interchangeable. General evacuation procedures are listed below.

1. To insure that the vacuum pump is capable of pulling an adequate vacuum, first connect the micron gauge directly to the pump, cap off other pump ports, and start the pump. Let it run until the micron gauge reads 50 microns or lower. Do not attempt to use a vacuum pump that does not pass this test. Remove the gauge and cap off the port to which it was attached.
2. After refrigerant is recovered and with atmospheric pressure in the system, remove the Schrader valve cores from system access ports, connect the flare tee to the high side access port, connect the low and high side hoses (each with no valve core depressors) to their appropriate access valve ports. Cap off the open flare tee port.
3. If necessary to check for refrigerant leaks, pressurize the system with dry nitrogen to 150 psi. When leaks are fixed and the system is returned to atmospheric pressure, remove the cap on the flare tee and connect the micron gauge to that port.
4. Next, connect the 3/8 in. gauge hose from the manifold to the vacuum pump and the refrigerant hose to the refrigerant cylinder. Before connecting the cylinder, weigh it with a digital scale and record. Purge the refrigerant hose, close the refrigerant valve, start the vacuum pump, and open the pump and high and low side manifold valves. Open the pump's ballast valve and close it at 1,500 microns.
5. When the pressure is reduced to the recommended 240 microns, close the vacuum pump valve on the manifold and observe that the pressure does not exceed 500 microns after 5 minutes.



To use the Triple Evacuation Method:

1. Perform a preliminary leak test by closing valves and adding dry nitrogen to 150 psig and hold for 10 minutes to remove water vapor. If system pressure does not stay at 150 psig for 10 minutes, then locate and repair any leaks.
2. Attach hoses from the liquid line and suction line Schrader valves to the vacuum pump (pumping from high and low side simultaneously can reduce the evacuation time by up to 75%). Open valves on manifold and turn on vacuum pump for the first time. At the beginning of the evacuation, the gas ballast valve should be opened to remove moisture from the system and avoid contaminating the vacuum pump oil. After the vacuum pump quiets down and the micron gauge reads 1500 microns or less, the gas ballast valve should be closed. Evacuate to 1,000 microns as indicated on the micron gauge.
3. If the indicator shows a pressure rise above 1500 microns and the pressure continues to rise without leveling off, a leak exists in the AC system or connecting hoses. Locate and repair leaks. If no leaks are detected, then system is leak proof (as indicated above).
4. If the system pressure rises slowly to between 1,000 and 1500 microns, then there is moisture in the system. Close valves and add dry nitrogen to remove water vapor and increase system pressure above atmospheric pressure to 15 psig and hold for 10 minutes.
5. Open valves on manifold and turn on vacuum pump a second time and evacuate to 240 microns and no higher than 500 microns. Immediately break the vacuum by adding dry nitrogen to 15 psig and hold for 3 minutes.
6. Open valves on manifold and turn on vacuum pump a third time and evacuate to 240 microns and no higher than 500 microns, as indicated on the micron gauge and hold for 5 minutes.
7. If the pressure isn't rising above 500 microns after 5 minutes, then the triple evacuation has been completed and the system is ready for charging with new refrigerant.

To use the Deep Evacuation Method:

1. Perform a preliminary leak test by closing valves and adding dry nitrogen to 150 psig and hold for 10 minutes to remove water vapor. If system pressure does not stay at 150 psig for 10 minutes, then locate and repair any leaks.
2. Attach hoses from the liquid line and suction line Schrader valves to the vacuum pump. Open valves on manifold and turn on vacuum pump and evacuate to 240 to 500 microns as indicated on the micron gauge and hold for ten minutes.
3. If the indicator shows a pressure rise above 1500 microns and the pressure continues to rise without leveling off, a leak exists in the AC system or connecting hoses. Locate and repair leaks. If no leaks are detected, then system is leak proof.
4. If the system pressure rises slowly to between 1,000 and 1500 microns, then there is moisture in the system. Close valves and add dry nitrogen to remove water vapor and increase system pressure above atmospheric pressure to 15 psig and hold for 10 minutes. Then, open valves on manifold and turn on vacuum pump a second time and evacuate to 240 microns and no higher than 500 microns.

5. If the pressure isn't rising above 500 microns after 5 minutes, then the deep vacuum has been completed and the system is ready for charging with new refrigerant.

### **C-7: Charging Procedures**

The following procedures are recommended to properly charge an air conditioner with refrigerant. Only add refrigerant charge through the liquid line. Adding charge through the suction line can damage the compressor.

1. Perform proper evacuation recovery and deep vacuum procedure noted above.
2. With the evacuation complete, remove the evacuation hoses, install Schrader valve cores, and purge the conventional three-valve manifold and hoses with new refrigerant. This can be accomplished on the EPA fittings by connecting the fitting to a W male flare fitting, purge, remove the flare fitting, and connect EPA fitting to the appropriate Schrader access valve port on the system. Repeat on the other hose.
3. Determine the factory charge from manufacturer's specifications on the name plate of documentation on the manufacturer's website. For split systems, include any line-set length beyond or shorter than the standard line-set length per manufacturer's specifications.
4. Place a tank of new refrigerant upside down on digital scale to ensure liquid is charged to the system (especially important for R410A), attach center hose of manifold to new refrigerant tank, locate manifold so as not to affect refrigerant scale readings, and place low-side hose to the suction line Schrader valve of the air conditioner (all hoses are evacuated to vacuum). Tare the scale to indicate a zero reading.
5. Open the valve on the tank of new refrigerant to fill the air conditioner with new refrigerant to as close to the factory charge as possible. Reinstall Schrader valve cores when the system is under a slight positive pressure.
6. The air conditioner will gradually fill with new refrigerant close to the factory charge. As the air conditioner gets within 10% to 40% of the factory charge, turn ON the air conditioner and allow the compressor to draw in the correct factory refrigerant charge.
7. Check for non condensables and/or restrictions (per diagnostic thresholds of evaporator saturation and condenser saturation temperatures).
8. Check the final charge to achieve proper superheat, subcooling, non condensables, and restrictions for TXV- and non-TXV-equipped systems (per manufacturer's specifications).

### **C-8: Proper Refrigerant Procedures to Avoid Non-condensables and Restrictions**

Proper refrigeration procedures are required to avoid introducing non-condensables and restrictions to the AC system. The EPA prohibition on venting requires precautions that must be followed when connecting or disconnecting refrigerant hoses. Never use contaminated refrigerant for recharging a system especially after burnout of a compressor. Beyond these guidelines, the following procedures should be followed in the care and use of service equipment.

1. All hoses must be equipped with EPA low-loss fittings.

2. Follow procedures carefully to insure that the system is fully evacuated and that hoses are purged with new refrigerant before recharging.
3. After charging and with the air conditioner operational, open the high-side valve and low-side valves on the manifold. This will cause any refrigerant trapped in the high-side and middle hose to be drawn into the low-side of the system. When both manifold gauge pressures are equalized, so the high-side is the same pressure as the low-side, then close off both high-side and low-side valves on the manifold. Then, remove the low-side manifold hose from the suction line Schrader valve of the air conditioner.

### **C-9: ASHRAE Recommendations for Proper Evacuation to Remove Non-condensable Air and Water Vapor**

Removal of non-condensable gases—air, nitrogen and other gases that are left in the system and mixed with the refrigerant—can have severe impacts on the performance of the unit. The *2010 ASHRAE Handbook – Refrigeration* recommends that units be evacuated to 240 microns of Hg. In general, evacuating to 240 microns Hg should remove non-condensables, but might not eliminate all moisture.

Removal of moisture can be much more difficult. The *2010 ASHRAE Handbook – Refrigeration* states the following. “Excess moisture in refrigeration systems may lead to freeze-up of the capillary tube or expansion valve. Contaminants can cause valve breakage, motor burnout, and bearing and seal failure. Except for freeze-up, these effects are not normally detected by a standard factory test. Therefore, it is important to use a dehydration technique that yields a safe moisture level without adding foreign elements or solvents. In conjunction with dehydration, an accurate method of moisture measurement must be established. Many factors, such as the size of the unit, its application, and the type of refrigerant, determine the acceptable moisture content. Table 1 shows moisture limits recommended by various manufacturers for particular refrigeration system components.”

### **C-10: Method to Detect Blocked Distributor Tubes**

Although it is an infrequent problem, unequal refrigerant flow can be caused by kinked or blocked distributor tubes. One way to detect this is, with pressure gauges installed, remove the coil cover and cover the opening with cardboard. While monitoring the suction pressure gauge, block the return grille enough to reduce the evaporator temperature to about 20°F–22°F. Let the system run for awhile so frost will form on the coil return bends. Remove the cardboard every few minutes to look for frost. When frost becomes evident, look for return bends with less or no frost as a clue to less or no refrigerant flow to an offending circuit. If all return bends have equal frost formation, it is a sign that refrigerant flow is uniform. Depending on where the low or no frost area is, the unequal frost formation could also be caused by non-laminar air flow, causing the TXV (if present) to throttle back, thereby increasing superheat and preventing frost from forming in those areas of the coil that have higher airflow.

## Appendix D: Test Equipment Specifications and Calibration

### D-1: Recommended Accuracy of Test Equipment

*Temperature:* Accuracy:  $\pm(0.1\% \text{ of reading} + 1.3^\circ\text{F})$   
Resolution:  $0.2^\circ\text{F}$   
Response Time: 15 seconds to reach temperature that is  $40^\circ\text{F}$  warmer or cooler than the ambient temperature

*Duct Pressure:* Accuracy:  $\pm 0.2 \text{ Pa}$  (use Dwyer A303 static pressure probes or equal)

*Airflow:* Accuracy:  $\pm 7\%$  of reading or  $\pm 5 \text{ cfm}$ , whichever is greater

*Refrigerant Pressure:* Accuracy:  $\pm 3\%$  of reading

### D-2: Calibration Method for Digital Thermometer Sensors

The following procedure should be used to check thermometer/temperature sensor calibration:

1. Fill an insulated cup (foam) with crushed ice. The ice must completely fill the cup. Add water to the top of the cup.
2. Insert two sensors into the center of the ice bath and attach them to the digital thermometer.
3. Let the temperatures stabilize. The temperatures must be  $32^\circ\text{F}$  (plus or minus  $1^\circ\text{F}$ ). If the temperature is off by more than  $1^\circ\text{F}$ , make corrections according to the manufacturer's instructions. Any sensors that are off by more than  $2^\circ\text{F}$  must be replaced.
4. Switch the sensors and ensure that the temperatures read on both channels are still within plus or minus  $1^\circ\text{F}$  of  $32^\circ\text{F}$ .
5. Affix sticker with calibration check date onto sensor.

### D-3: Refrigerant Gauge Calibration Procedure

Refrigerant gauges should be checked monthly to ensure that they are reading the correct pressures and corresponding temperatures. The following procedure should be used to check gauge calibration:

1. Place a refrigerant cylinder in a stable environment and let it sit for 4 hours minimum to stabilize to the ambient conditions.
2. Attach a calibrated sensor to the refrigerant cylinder using tape so that there is good contact between the cylinder and the sensor.
3. Insulate over the sensor connection to the cylinder.
4. Zero the low side and high side refrigerant gauges with all ports open to atmospheric pressure (no hoses attached).

5. Re-install the hose, attach the high side gauge to the refrigerant cylinder, and open the valves to measure the pressure in the refrigerant cylinder.
6. Read the temperature of the sensor on the refrigerant cylinder.
7. Using a pressure/temperature chart for the refrigerant, look up the pressure that corresponds to the temperature measured.
8. If gauge does not read the correct pressure corresponding to the temperature, the gauge is out of calibration and needs to be replaced or returned to the manufacturer for calibration.
9. Close the valve to the refrigerant cylinder, and bleed off a small amount of refrigerant to lower the high side pressure to give a corresponding temperature to between 45°F and 55°F.
10. Open the valves between the high side gauge and low side gauge.
11. If the two gauge's corresponding refrigerant temperatures do not read within 1°F of each other, the low side gauge is out of calibration and needs to be replaced or returned to the manufacturer for calibration
12. Affix sticker with calibration check date onto refrigerant gauge.

#### **D-4: Other Devices**

Airflow measurement devices and manometers should be returned to the manufacturer for calibration when the period of the manufacturer's guaranteed accuracy expires.

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