A Robust Feature Set for Residential Air Conditioners

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CONTENTS

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EXECUTIVE SUMMARY

Robust central air conditioners and heat pumps are units with special features and specifications. They are designed to remain very efficient and require minimum maintenance over their lifetimes, and to automatically signal problems that require service. This report stresses the specifications required to identify the differentiated product, and the parameters that we varied in a simulation to study the benefits of the specification. We highlight several issues that define the robust air conditioner and will require discussion among stakeholders:

- **Base Efficiency Level.** The robust air conditioner meets the 2006 ENERGY STAR performance specifications, SEER 14 and EER 11.5.[1] Using the ENERGY STAR specification allows easy consideration of the additive energy-saving benefits beyond the ENERGY STAR specification, and suggests that robust and ENERGY STAR complement each other in achieving savings for customers.
- **Variable Cooling Capacity.** Modulating or two-speed compressors can improve air conditioner performance, particularly by mitigating the most severe aspects of oversizing. Capacity modulation is not a prerequisite for the robust specifications.
- **Adaptive SHR.** SHR, or Sensible Heat Ratio, measures the ability of the air conditioner to remove moisture. All air conditioners remove more moisture when the incoming air is more humid. The question is whether to require extended capabilities in this area. We did not make adaptive SHR a prerequisite, feeling that this area would receive further consideration elsewhere.
- **Alarms.** The robust system thermostat will be required to display alarms corresponding to excessive pressure drop across the air filter, and when service is needed, as noted in the technical specification.

In our simulations, we considered the effects of the following:

- **Improper Refrigerant Charge.** Because the 2006 SEER 13 minimum standard seems to have led to widespread adoption of refrigerant metering devices that incorporate feedback (such as thermostatic expansion valves, or TXVs), we simulated the effects but did not include these savings relative to orifice or other fixed metering devices.
- **Adaptive Airflow.** This is the potential savings with a BPM (brushless permanent magnet), ECM (electronically commutated motor), or other advanced air handler motor. Our simulations suggest that it would save 6–11% of total electricity use, depending on climate.
- **Slow Refrigerant Leak.** We simulated the effect of a leak slow enough that the unit would quit working when the refrigerant charge dropped to 80% of the proper level. Relative to an orifice-equipped unit, the robust air conditioner would save 6–8% of electricity use over its lifetime by sending an alarm to the thermostat (or service firm) when charge level dropped to 90%. Savings relative to a TXV-equipped unit would be smaller.

[1] The ENERGY STAR specification contains no installation components. Requirements are slightly lower for packaged units (SEER 14 and EER 11). SEER stands for Seasonal Energy Efficiency Ratio and EER stands for Energy Efficiency Rating.

- **Reduced Air Handler Unit (AHU) Leakage.** This would save 1–3% of total electricity use, primarily because the air handler has the largest pressure differential with the ambient, both negative (air handler end) and positive (supply plenum). Of course, for attic-mounted equipment the ambient temperature at the air handler/evaporator is generally much higher than stipulated in the test method, so drawing in "ambient" attic air gives a mix that is hotter than the 80ºF return expected from the rating method.

After initial discussions with manufacturers and others, we removed several possible features of the robust air conditioner. These included (with reasons):

- **Oversizing.** This is largely outside the control of the manufacturer. The OEM can influence this to some extent by supplying units with modulating compressors, but this is a feature we did not want to include, as noted below.
- **Modulating Cooling Capacity.** This is an important premium feature and presumed to be a key profitability driver for these products. Our goal was to develop a specification for a unit that would be differentiated from "commodity" offerings or today's ENERGY STAR specification, but not one that simply recognizes the most expensive models.
- **Adaptive Sensible Heat Ratio.** To some extent, conventional units are already adaptive: the higher the wet bulb temperature, the more humidity will be removed at a given evaporator surface temperature. To go beyond this requires advanced features, such as modulating compressors interacting with variable speed motors to tune the ratio of airflow to refrigerant flow and thus control evaporator temperature. This is currently available in some premium products that are controlled by both temperature and humidity level. It can also be addressed by alternative designs, such as the Cromer cycle, that increase the potential for dehumidification. These are not yet on the market, and have some intellectual property issues that may prevent multiple manufacturers from adopting these features.
- **Condenser Fouling and Blockage.** A clogged condenser loses efficiency. Some coil and fin designs may be more resistant to this degradation than others, but there is little data on this aspect of long-term efficiency losses. In any event, for a given manufacturer, the most likely design feature to ameliorate this decay would be to increase the condenser heat exchange area. This "brute force" approach did not look like a smart inclusion for the robust criteria.

We conclude that the following features define a Robust Central Air Conditioner that has the potential to improve customer satisfaction and provide a platform for marketing premium, differentiated products. It will save energy, but all of the savings are from characteristics that are not reflected in the federal test procedure.

- **SEER 14; EER 11.5.** We aligned these performance parameters with the 2006 ENERGY STAR specification, to allow easy comparison of simulated energy use. We recommend that the robust air conditioner SEER and EER be set at ENERGY STAR levels, or CEE Tier 2 (currently SEER 15/EER 12) as a value-added feature of efficiency programs.

- **Resistant to refrigerant charge errors.** Maintains 90% capacity and efficiency with +/-20% mischarge. In effect, this virtually mandates an adaptive expansion valve, such as a TXV.
- **Resistant to refrigerant leaks and other degradation**. Shuts down or calls for service if efficiency or capacity drops below 90% of nominal.
- **Adaptive air handler**. Air handler adjusts airflow in response to ductwork external static pressure, to maintain specified air flow. The specification requires maintaining airflow within 15%, with ESP from 0.1 to 0.9 inches of water. We would also require that the motor have current limiting or temperature rise control to prevent significant life reduction when operated in high external static pressure systems. This feature is required since so many houses have very high external static pressures. This can be caused by high-MERV air filters, or duct design and installation problems.
- **Air handler integrity**. We propose that the robust unit will have a tightly-sealing filter rack included as part of the package, and that the cabinet leakage be *de minimus.* We characterize this as "<1%," pending a suitable test method.
- **Alarms**. We specify that the unit should include an "alarm" (thermostat notification) if the air filter is loaded to the point that airflow is significantly reduced, or that the unit needs service (such as results from loss of refrigerant). The alarm should call for service or alert the owner when performance changes by 10% or more. This would include, for example, an airflow increase/ESP drop due to disconnect of a register boot.
- **Quality**. The robust air conditioner shall have a single source and be sold as an integrated package of the condensing unit, evaporator, and air handler (with or without furnace section) and thermostat. This is necessary to assure proper sizing and component match, and to avoid finger-pointing among vendors in case of problems.

From simulations of the features selected for inclusion, we conservatively estimate that the robust unit would save on the order of 10%, which is more than the change from 14 SEER to 15 SEER. That is, a robust air conditioner should give better performance than a conventional unit rated one SEER point higher. However, there remain many uncertainties, particularly with respect to the fraction of units with slow refrigerant leaks, and the benefits of an alarm at 90% v. 80% for these units.

GOAL

Residential air conditioners and heat pump systems as installed in the field generally do not achieve the efficiency implied by their SEER ratings. There are several reasons for this performance shortfall. One cause is the inability of a national performance standard (SEER) to adequately predict performance in hot-dry or hot-humid climates. In addition, equipment oversizing and installation errors such as improper refrigerant charge substantially reduce efficiency, too. Finally, duct leakage easily increases energy use by 20% or more. Clearly, different problems require different solutions, and not all of these issues can be controlled by the equipment manufacturer.

However, the cumulative effect is huge: Neme, Proctor, and Nadel (1999) concluded that improved practices could save 24% of heating, ventilation, and air conditioning (HVAC) energy in existing buildings and 35% in new construction. Customers are not getting the comfort and economy they contract to buy, dealers are getting too many call-backs, and manufacturers are blamed for inefficient and poorly operating systems, when they only control the equipment, which is one element of the system.

Within this set of problems, the focus of this report is the set of attributes that can be influenced by design decisions and factory quality control. We start with the question: "Within the range of reasonable costs, how close can a premium central air conditioner come to running efficiently, without maintenance except for air filter changes, for fifteen years?" Of course, the basic features should include high efficiency (SEER) and good high temperature performance (EER). However, the key features would also include the capacity to adapt automatically to different (duct system) external static pressure (ESP), excellent ability to compensate for refrigerant charge anomalies (and slow leaks), and auto-diagnostics for major failure modes. We call air conditioners with this package of features "robust."

Our goals are to develop and evaluate an appropriate set of criteria, and then work with manufacturers and market transformation organizations to label and encourage robust equipment. We hope that this document will stimulate industry discussion and eventual formulation of an attractive package that will benefit industry and consumers.

BACKGROUND

The robust air conditioner specification addresses issues over which manufacturers have full or partial control. These include the following:

- *Minimum energy efficiency* (SEER ≥14 and EER ≥11.5. These are the 2006 ENERGY STAR specification for split systems.)
- *Adaptive response* to incorrect charge
- *Airflow management*
- *Integrated controls:* condensing unit, evaporating unit, and controls from the same source

The concept of the robust air conditioner has been informally discussed in the efficiency community, and with some manufacturers, since 2001.[2] Over the period of its gestation, much has changed. In particular, a new minimum efficiency standard, SEER 13, took effect January 23, 2006. This improves minimum efficiency as defined by this parameter by 23%.[3] Further increases will garner smaller energy savings, and may face substantially higher costs at very high SEER levels. In 2006, the U.S. Environmental Protection Agency (EPA) adopted a revised ENERGY STAR specification, at 14 SEER, and added a requirement of minimum 11.5 EER for split systems (11.0 for packaged units). The EPA specification does include installation quality requirements, such as those developed by ACCA (2006).

Neither approach exhausts the potential for products that deliver consumer value and assist manufacturers with differentiating such products. However, the robust specification could become the basis for an industry or other program, analogous to the "Good Housekeeping" seal, to recognize products engineered to run as well as possible for as long as feasible, regardless of site-specific conditions. Alternatively, the robust air conditioner specification could be recognized as a later (2009?) feature for ENERGY STAR, helping to assure that high performance is achieved and maintained for the life of the equipment.

The robust concept is designed to give manufacturers some independent validation for their efforts to market products differentiated by energy efficiency and reliability. Among other characteristics, robust requires a single source for the system (condenser/compressor, evaporator, air handler/motor, and thermostat/controls). This will result in premium pricing, but will avoid many common forms of finger-pointing among manufacturers and installers. It will facilitate getting the required alarm functions into the thermostat, too. In turn, by showing what can be achieved in manufacturing a robust product, and what cannot (control of sizing, distribution, and shell), our work will support efforts to improve technician training, and to tighten standards for construction.

The robust air conditioner is not inherently more efficient under test conditions than a conventional product, but is expected to maintain its performance better than conventional equipment, thereby offering long-term efficiency gains. By maintaining performance and adapting to the house as well as possible, the robust air conditioner should have demonstrable consumer comfort benefits, as well.

However, the robust specification is not a panacea. Not every conforming product will meet expectations, largely because the robust specification cannot control installation issues. These include choosing the correct capacity: oversized (single-stage) equipment loses efficiency (and environmental control) when it does not run long enough to reach steady state. The robust equipment will do the best job possible with a leaky duct system or one with anomalously high external static pressure, but it cannot fix the underlying problems.

[2] The concept was outlined and circulated to some stakeholders by H. Sachs in the summer of 2001, and given focus through informal conversations with industry representatives involved in the Air-Conditioning and Refrigeration Technology Institute, Inc.'s (ARTI) "21CR" research program. See http://www.arti-research.org/21cr_summary.php.

[3] Energy consumption is proportional to the inverse of SEER: energy use goes *down* as SEER goes *up*. Thus, the *decrease* in energy use is the difference in the reciprocals of the SEER values: 23% instead of the 30% change from SEER 10 to SEER 13.

Thus, the robust specification is a complement to other programs, such as an ENERGY STAR specification. It also complements ongoing field verification and duct performance programs by utilities and others. In combination with them, it can assure that residential air conditioning systems actually achieve persistent savings, and that the systems will notify occupants when they cannot maintain control and efficiency.

How We Worked

This work included several discrete components:

1. ACEEE led the development of a Trial Feature Set, with participation by CDH Energy, FSEC, ECW, and WECC. The final set (see Appendix A) was the basis for simulations by CDH Energy.
2. Based on the accepted Trial Feature Set, CDH Energy simulated the energy efficiency benefits of equipment incorporating the specification, by house type and climate.
3. Our work will be circulated to manufacturers and other experts so they can incorporate the robust feature set into market transformation efforts. It will also be circulated to the EPA and CEE as a concept for inclusion in their respective market transformation programs, ENERGY STAR, and coordinated incentive tiers.

Field Performance Deficiencies Considered

In this section, we consider performance and reliability issues that can be addressed through improved equipment capabilities. These concerns include:

- Refrigerant Charge Errors
- Refrigerant Leakage over Time
- Air Handler Impact of Oversizing (Indirect)
- Ductwork and Accessories External Static Pressure
- Cabinet Air Leakage
- Air Filter Rack and Tight Door
- Call for Service (Other)

Refrigerant Charge Errors

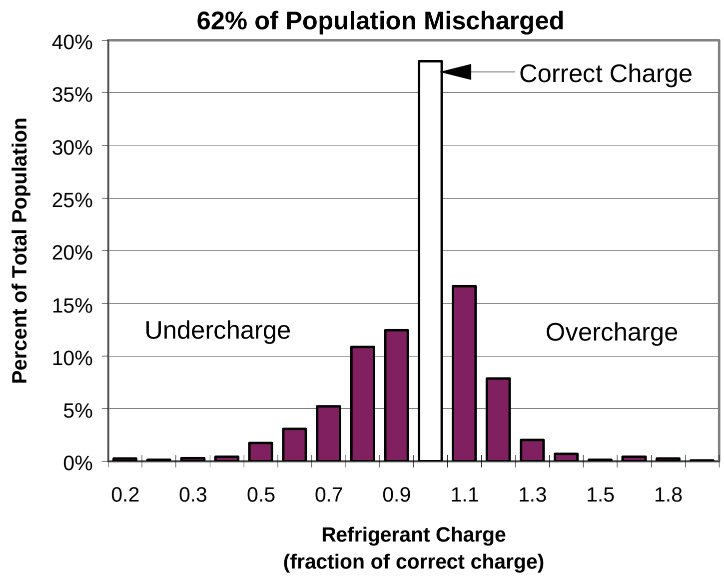
The performance of residential central air conditioners in the field falls short of expectations implied by the federal SEER values. In part, this reflects differences between the test and field conditions. In the test facility, units always have the proper refrigerant charge. Also, the external static pressure in the test procedure is no more than 0.2 inches of water gauge, or IWG (ARI 2003,[4] p. 9, table 6), while the *average* for installed air conditioners is about 0.5 IWG (Mowris, Blankenship, and Jones 2004; Neme, Proctor, and Nadel 1999; Proctor and

[4] Superseded by AHRI 210-240-2008: Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment. For federal efficiency standards purposes, the relevant document is [DOE]. Department of Energy, 2005. 10 CFR Part 430. "Energy Conservation Program for Consumer Products: Test Procedure for Residential Central Air Conditioners and Heat Pumps; Final Rule." *Federal Register* / Vol. 70, No. 195 / Tuesday, October 11, 2005, 59121 – 59180.

Parker 2000). This results in lower airflow in the field, possibly with greater ability to dehumidify.[5] This is exacerbated by installation technicians who do not select the right power tap (PSC) or properly set the airflow specification (ECM).

In terms of the features of the robust air conditioner, refrigerant charge of pre-2006 equipment is generally far enough out of specification that performance is impacted. Proctor Engineering Group, Ltd. (2000) showed that almost two out of three units studied (62% of 4,000) are mischarged by 10% or more (see Figure 1).[6] These data have been largely verified in other studies—for example, see Mowris, Blankenship, and Jones (2004).

Figure 1. Frequency Distribution of Refrigerant Charge

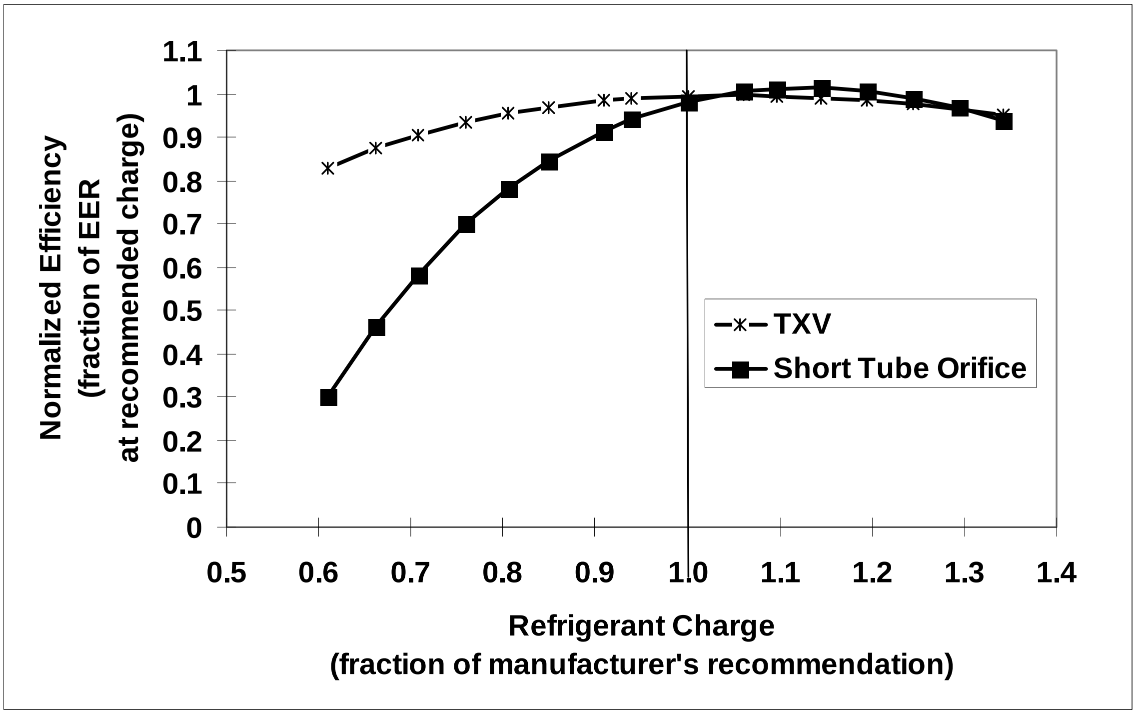


Source: Proctor Engineering Group, Ltd. (2001)

Improper charge is a critical performance issue, particularly for systems that do not have TXVs or other active feedback controlling liquid flow to the evaporator. Figure 2 shows the efficiency drop-off with relative refrigerant charge for TXV- and short-tube-orifice (no active feedback) systems. For the latter group, a 20% charge deficiency (roughly 20% of all systems in Figure 1, above) yields more than a 20% drop in EER relative to test conditions, while TXV equipment is relatively resistant to performance degradation from inadequate charge.

[5] This is true for conventional systems with PSC motors; systems with adaptive motors, such as proposed for the robust specification, will respond instead by increasing their power consumption to maintain specified airflow.
[6] This is almost uniquely addressed in California with the “CheckMe,” “RCA,” and “HVAC Assistant” equipment and programs.

Figure 2. Normalized Efficiency as a Function of Refrigerant Charge



Source: Proctor Engineering Group, Ltd. (2001)

Slow Refrigerant Leak

Proper charge at time of installation and prevention of leaks during the operational life of the unit are keys to efficiency — and robustness. Unfortunately, system degradation over time from very slow leaks has not been adequately studied. The robust criteria propose to address this issue with an alarm function to alert the owner of a problem, via the thermostat. This specification implicitly encourages a TXV or better feedback-controlled device to meter liquid refrigerant to the evaporator.

Air Handler Adaptive Capabilities

Neal (1998) included the effects of airflow problems as well as refrigerant charge anomalies in considering air conditioning system performance. He showed that errors in charge and airflow alone can easily degrade performance by 20–25% (Neal's Table 2). There are three key airflow issues: high external static pressure, duct leakage, and duct insulation. All are related to distribution system design or installation. Particularly in regions where slab-on-grade construction predominates, air conditioners and their ducts have typically been installed in attics, where the ambient air is very hot in summer and very cold in winter. Duct leakage in such "outside the thermal envelope" places requires make-up air, typically drawn from the attic. This is very expensive to temper to desired indoor conditions. Typical duct leakage is 20%: often it is more. In addition, typical attic and crawl space ducts are not well insulated, exacerbating energy losses. Although these sources of inefficiency are important, they are outside the control of the manufacturer, and thus not considered in this specification.

However, equipment can be designed to respond appropriately to the third issue, high external static pressure. Over three million units have been sold with ECM air handler fan motors from a single manufacturer[7] (ACHR 2003). These are DC permanent magnet motors that have two important characteristics: (1) They are substantially more efficient than conventional (permanent split capacitor induction) motors, particularly at lower speeds; and (2) The motor controller adjusts motor speed to maintain the airflow rate selected by the installing technician.[8] Within the fan's power limitations, this assists air distribution (particularly to remote rooms). More importantly, maintaining a specific airflow rate can be critical for humidity control. Depending on the system, in a hot-dry climate the installer might be able to increase the flow rate (e.g., from 350 cfm/ton to about 400 cfm/ton) to increase the sensible heat ratio, and reduce compressor run time. Conversely, in a humid climate, the installer could reduce airflow to enhance dehumidification.

These fan motors are sold as part of premium packages. The robust air conditioner specification requires the ability to perform with ESP from 0.1 to 0.9 IWG to give the best possible assurance that the equipment can meet customer needs. One solution, the ECM, is likely to remain a premium product feature, but the performance specification may allow other technologies to enter the market as well. The ECM allows the installer to set airflow. Other approaches might use alternative feedback systems, or even require manual adjustment.

Cabinet Air Leakage, Air Filter Rack and Door

Residential air handlers are not well-sealed. As a consequence, they draw air into the air handler cabinet and push it out through the air conditioning heat exchanger sections (Cummings et al. 2002). Our draft specification requires that air leakage of the factory-specified air handler and evaporator assembly shall be less than 1% of rated airflow, compared with typical values averaging 4.5% (Cummings 2005). This is particularly important for units installed outside the thermal envelope (attics and attached garages, for example). In addition, the unit shall be shipped with or include a filter holder with a tightly-sealing access door.[9] We are not certain how 1% leakage could be reliably measured — or achieved, but intend to simulate the effect of leakage reduction for units installed in unconditioned spaces.[10]

Air Handler Filter Loading or Blockage (Alarm)

Failure to change air filters when necessary is common, at least anecdotally. The robust unit requires an alarm signal at the thermostat when filter blockage is significant, to indicate it's time to change the filter (or call service). We simulated the absence of the alarm, and

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[7] The context suggests that these have predominantly been installed in furnaces and air handlers.
[8] This feature is enabled with DIP switches on the unit's control board in the fan compartment, and thus is not considered user-selectable.
[9] Missing air filter pocket covers may allow leakage of 15% of total airflow, and this will often be from outside the thermal envelope (Cummings 2005).
[10] One reviewer has suggested that units designed for installation in unconditioned spaces (e.g., horizontal units designed for attics) should not have filters or filter racks. Instead, these should be included as return air grille filters.

consequent operation with partially blocked filter, by estimating the percentage degradation and the fraction of the time the unit operates in this condition. We suggest change in ESP of 0.2 IWG as the performance degradation criterion that marks a functionally blocked filter.[11] Alternatively, we could use 10% and 15% flow reduction (360 and 340 cfm/ton, respectively), and are considering one additional low-flow level (320 cfm/ton), which would correspond to the airflow associated with increasing likelihood of coil freezing. In either case, the alarm must be correlated to display text on the thermostat that instructs the user to service the air filter, and to call for service if the alarm persists (indicating a likely duct problem).[12]

Call for Service, Other (Alarm)

This alarm would respond to changes that are unlikely to be resolvable by the homeowner. Such situations include:

- Relatively rapid change in ESP, corresponding to failure of a ductwork connection (duct tape falls away, flex duct crushed by serviceperson, etc.), as opposed to slow loading of air filter.
- Significant change in refrigeration system, whether as a result of a slow leak (discussed above), or other failures.

For this set of alarm conditions, we simulate conditions as follows:

- The robust unit would signal to call for service when performance changes by 10%. This would correspond to a change in ESP from one or at most two failures of register boot to flex duct connections, or crushing one or at most two flex duct final runs. It would correspond to a significant slow leak, or to reduced condenser performance.
- The baseline unit would be allowed 20% performance degradation before it presumably quits working.

To ensure that the alarm functions could be enabled easily by field personnel, we propose to require that the condensing unit, air handler, and thermostat/controller all be provided by the same OEM, who will take responsibility for certification and integration of the system.

Although this is a long list of features, there were others that we elected *not* to include in the present study. These are discussed more fully in Appendix B, but include:

[11] One commentator suggested that a timer would be more appropriate for electronic air cleaners. We would be amenable to this approach if the air filter were integral to the equipment or installation, but this may be difficult in practice.

[12] Electronic air cleaners will require a different approach. ESP does not tend to increase appreciably (from already low levels) as these are loaded, but efficacy drops. It may be that an inexpensive sensor system could "hear" and count the sparks that become more frequent with dust build-up.

- **Oversizing**. This is determined by the installer.
- **Modulating Cooling Capacity**. This is currently a high-priced premium feature for central air conditioners. Modulating equipment can compensate for oversizing, but we determined that this lies outside the thrust of our project.
- **Adaptive Sensible Heat Ratio**. This would address a chronic concern in humid regions such as the Southeast, where maximum dehumidification needs are at much lower ambient temperatures than maximum sensible heat needs. We did not include this because fully addressing it will require specialized equipment not yet commonly available, and adaptive SHR is not a core element of reliability and long-term efficiency.
- **Condenser Fouling and Blockage**. This is considered to be a significant problem, but the only practical way we could find to build in fouling prevention was through oversizing the condensing unit. This would affect costs and performance.

SIMULATIONS

A building simulation model developed in TRNSYS was used to assess the annual energy impacts of the robust air conditioner features. The TRNSYS model was previously developed for the Task 4[13] simulations. A detailed model was developed for a single-story, slab-on-grade 2,000 sq. ft. reference or baseline house as defined for the Home Energy Rating System (HERS).[14] Appendix C provides the details of the hourly simulations and analysis procedures. The details of the reference house are given in the Task 4 report.

The baseline house was maintained at the cooling set point of 75°F and evaluated in ten different climate locations. Insulation levels and window performance were consistent with the HERS reference requirements for each city (based on the IECC zones). The reference house has the HERS-specified amount of natural infiltration with no explicit mechanical ventilation. Duct leakage and thermal losses to the attic were consistent with an 80% distribution efficiency (as required for the HERS reference house).

The baseline air conditioner was assumed to be a 14 SEER unit. Fan power was increased (above the nominal fan power used to determine SEER) in order to reflect the higher static pressures that are typically observed based on field measurements (e.g., 0.5 inches). The baseline supply fan used a permanent split capacitor (PSC) motor that used 0.42 Watts/cfm at 400 cfm/ton. For each city, the air conditioner unit was sized to match the cooling load at the cooling set point temperature of 75°F to within 0.1 tons.

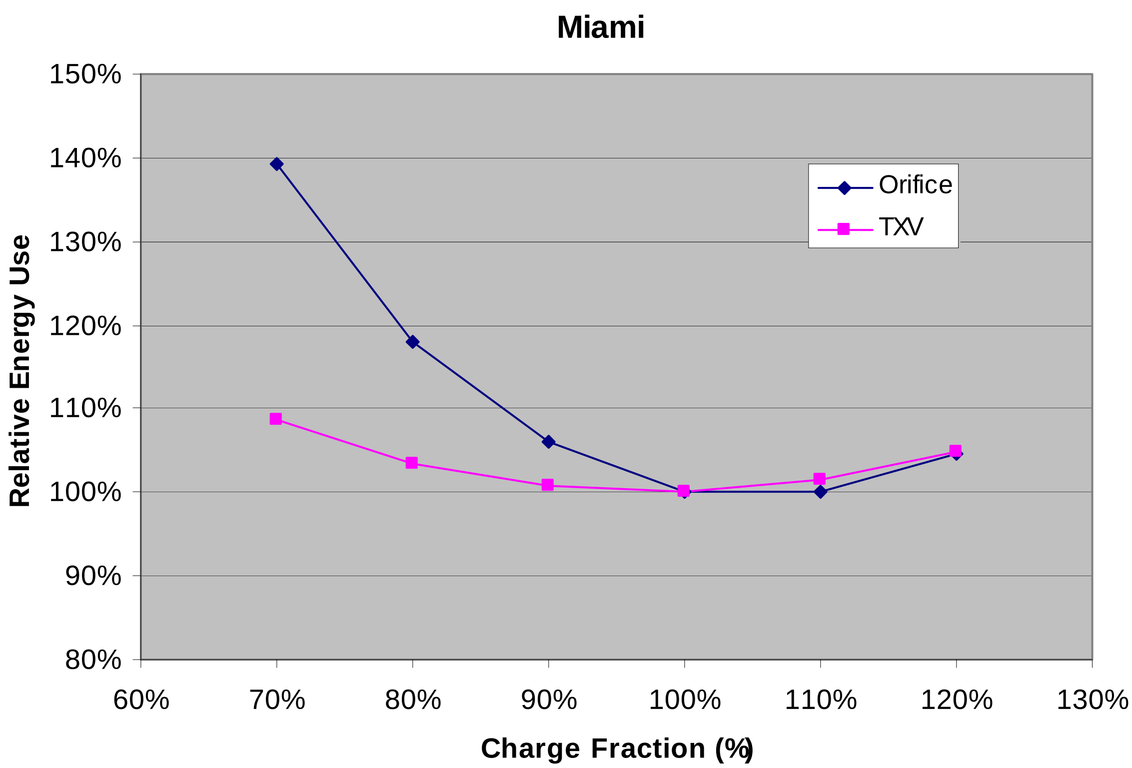
[13] See http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-1716-07.pdf.
[14] The HERS reference house is defined by the RESNET standards. See www.natresnet.org.

The Impact of Improper Refrigerant Charge

Laboratory testing by Pacific Gas & Electric (Davis and D'Albora (2001a, 2001b) quantified the impact of improper refrigerant charge over a wide range of operating conditions for a unit with a TXV and a short orifice. We distilled their test results into a set of simple correlations that could be incorporated into the simulation model. These results were combined with data from the CheckMe! Program (circa 2001) on the distribution of refrigerant charge measured for a large population of air conditioners (see Figure 1).

A key feature of the robust air conditioner would be a TXV. This type of expansion device makes an AC unit much more tolerant of improper refrigerant charge. The simulation results show that a unit with an orifice expansion device at 70% of full charge uses 31–41% more energy (compared to a properly charged unit) on an annual basis, depending on the climate location. For the robust unit with a TXV, the energy penalty at 70% charge becomes only 7–9%. Figure 3 shows the simulation results for Miami. When the results are weighted to reflect the range of refrigerant charge levels in the air conditioner population (see Figure 1), the robust unit with a TXV saves 3–4% of annual energy use, on average, compared to the baseline orifice unit.

Figure 3. Simulated Impact of Refrigerant Charge Fraction on Relative Energy Use for Miami



Slow Refrigerant Leaks

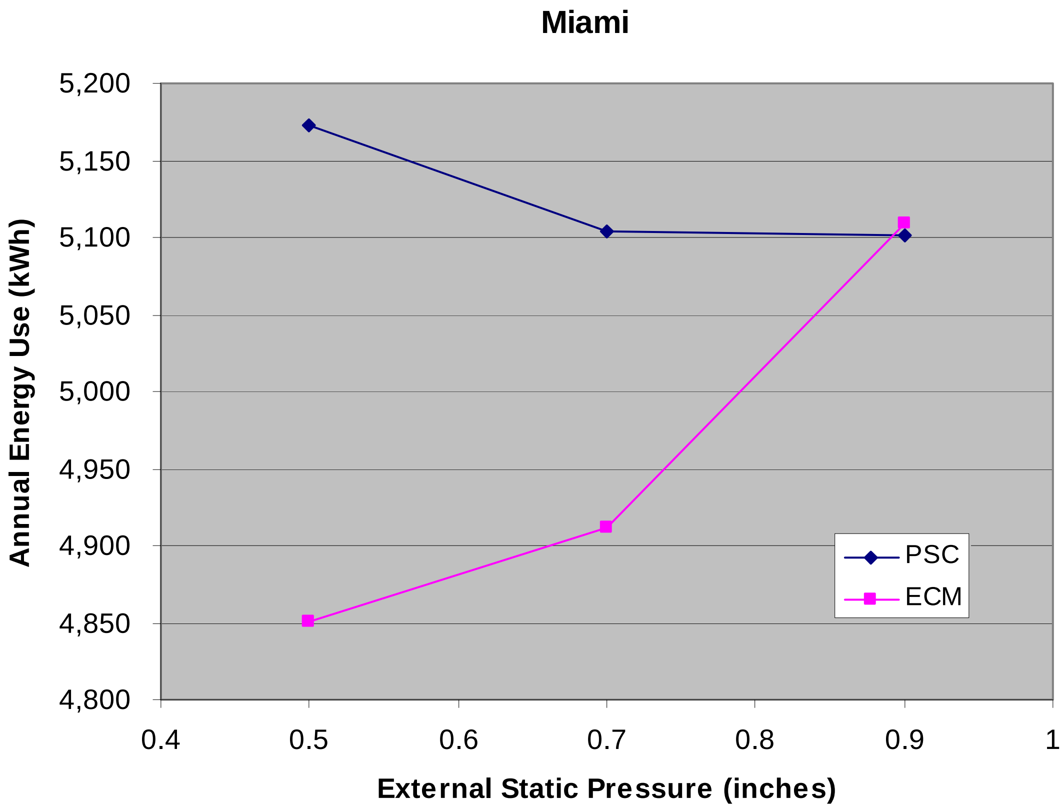
Another feature of a robust AC unit would be the ability to detect when refrigerant charge is low. We estimate that current air conditioner units can lose charge at a rate as fast as 10% per year. As a baseline, we assume that homeowners perceive a change in air conditioner performance when refrigerant charge drops to 80%. A robust unit would detect when refrigerant charge reaches 90% of full charge, and provide an alarm to let the homeowner know that a service technician should be called to recharge the unit. As a baseline, we took an average of the simulation results for the cases with 100%, 90%, and 80% of full charge with an orifice unit. For the robust unit, we took the average of simulation results with 100% and 90% of full charge. With these assumptions, the robust unit (with a TXV) would use 6–8% less energy over its lifecycle. The alarm feature of the robust unit would ultimately improve customer satisfaction and reduce maintenance costs by allowing the unit to be serviced only when a low-charge condition was detected. This would eliminate the need for calendar-based service calls.

Adaptive Airflow

The robust air conditioner would use an ECM on the supply fan instead of a PSC motor. An ECM motor is not only more efficient, but also offers the ability to maintain a more constant airflow as the external static pressure increases. This adaptive feature can help ensure that the desired airflow is maintained even as filters become clogged or other pressure restrictions are added to the system. The challenge is that this adaptive system also uses more energy because the fan motor must speed up to maintain the desired flow rate.

The simulations show that the ECM fan motor reduces total annual energy use by 6–11% compared to the PSC motor at an ESP of 0.5 inches. As more pressure drop is added to the system (for example, by an air filter becoming loaded with dust), the ECM fan motor is better able to maintain the desired airflow, but its energy use increases. When the external static reaches 0.9 inches with the ECM motor, it uses nearly the same amount of energy as the PSC motor. Figure 4 illustrates this concept using data from Miami. However, while the ECM uses more energy, it does maintain the airflow closer to the desired level. Reduced airflow can lead to comfort and thermal distribution concerns since some zones in the home may no longer receive the necessary amount of cooling (or heating).

Figure 4. Results for Miami with PSC and ECM Motors at Different Static Pressures



Reduced Air Handler Unit (AHU) Leakage

Field testing of residential air handlers by FSEC and others has revealed that a significant amount of air leakage occurs at the air handler. Most of this leakage is on the return or negative pressure side of the system. Filter access doors and other sheet metal components on the depressurized side of the cabinet have small openings that provide a return-side leakage path. If the AHU is located in the garage or attic, then the air entering the unit has a much higher enthalpy than the return air from the house and significantly impacts energy use. Even if the AHU is located in a closet within the home, leakage paths in the closet walls can still pull significant amounts of air from outdoors or the attic. As a baseline, we assumed that the return ductwork and AHU have combined leakage equivalent to 4% of the nominal flow rate.

The robust air conditioner has a tighter AHU cabinet that only allows return leakage equivalent to 1% of the nominal flow. For the simulations, we assumed all the air leakage into the AHU was from the unconditioned attic zone. Reducing the return air leakage fraction by 3% (from 4% to 1%) decreases energy use by 1.5 to 3%. Humid climates such as Miami and Houston show the biggest impact. Reducing the return air leakage rate by 4% (from 4% to 0%) has a proportionally larger impact.

RECOMMENDATIONS

This project and its simulations suggests that the energy savings potential of a robust air conditioner criteria are significant, probably at least 10%. A robust air conditioner should give better performance than a conventional unit rated one SEER point higher. However, there remain many uncertainties, particularly with respect to the fraction of units with slow refrigerant leaks, and the benefits of an alarm at 90% v. 80% for these units. These savings are exclusive of the shift from fixed orifice to adaptive refrigerant metering devices (TXVs and equivalent), and are the results of using ECM or equivalent motors, notifying users of service needs when refrigerant charge falls below 90% of the proper value, and reduced cabinet air leakage.

The next step is wider publication; we are preparing a manuscript on the concept for the *ASHRAE Journal*. We intend for this article to treat some additional topics, such as advanced diagnostics. This discussion will more fully introduce the topic to the industry and to groups promoting efficiency programs.

DISCUSSION

In January, 2006, the legal minimum cooling efficiency level for central air conditioners and heat pumps moved from SEER 10 to SEER 13. This should decrease unit electricity consumption for air conditioning by 23%, all other things being equal, and will have significant impacts. Equivalent gains will not be accomplished easily by raising SEER further. First, the savings increments per SEER point are smaller, as a percentage and as energy saved or bill reductions achieved. Second, the technology steps involved are likely to be more expensive per kWh saved than those in the SEER 10 to SEER 13 step. These steps are likely to require larger heat exchangers, multi-speed or continuously modulating compressors (and probably, condenser fans), advanced indoor air handlers with improved aerodynamics, and improved controls to effectively use the full area of the heat exchangers without adverse consequences. This is likely to make much higher SEERs less cost-effective in regions with relatively cool and/or short summers.

These changes mean that voluntary programs that promote better-than-standard air-conditioners face significant challenges. Two program types can be highlighted. Utility incentives are the first program type. In many areas, utilities offer rebates or incentives for better equipment, if the incremental costs are less than the cost of energy or new demand. The Consortium for Energy Efficiency[15] promotes program coordination through common efficiency levels and other characteristics. Coordinated programs provide manufacturers with consistent signals that promote energy efficiency. Further, the common programs offer utilities increased assurance that their programs will be deemed prudent where that is important for utility regulation.

The other program model is the national ENERGY STAR program, which attempts to recognize the best 20–25% of models in the market, when such models offer significant

[15] See www.cee1.org.

savings at reasonable cost. This robust specification builds on the 2006 ENERGY STAR specification by focusing on features that manufacturers can provide to maximize the likelihood that a well-installed system will have a long and happy life. It could be a suitable refinement of the ENERGY STAR program in 2009 or later. Alternatively, it could be offered as a free-standing program. We have aligned the efficiency requirements for robust units with the 2006 ENERGY STAR levels, but remain open to the option of base robust requirements at legal minimum performance.[16]

Implications

Criteria for robust air conditioners could help premium products. For manufacturers, differentiating products from commodities is a key to profitability. Energy efficiency, customer satisfaction, and features (such as humidistats) are routes to differentiation. Manufacturers assert that higher product efficiency floors (such as SEER=13) leave them less room to differentiate a higher efficiency product: relative to a SEER=13, the economic savings from a SEER=15 or 16 are small in most regions. Further, they assert that manufacturing costs rise quickly at higher SEER levels. For example, they believe that variable capacity is required for SEER=15 and above, while SEER=13 can be done with conventional, fixed-output equipment. The robust criteria program seeks to offer a way to validate and support sales of products that work better, independent of SEER rating.

Of course, a robust air conditioner program will not fix all of the issues raised in the course of this STAC project. In particular, this criteria does not address shortcomings in the rating method.[17]

To summarize, we believe that a package of robust features could help differentiate a product class that will appeal to customers seeking reliability and performance, as well as savings. Such a program could be offered by the Consortium for Energy Efficiency, designated by ENERGY STAR, or even offered through the trade association, AHRI (Air-conditioning, Heating, and Refrigeration Institute).

The next steps should be presentations to ENERGY STAR and CEE, together with encouraging both groups to discuss the robust concept with manufacturers.

We are confident that the manufacturers can build units with the features listed here. The issue from their point of view is whether such products would be profitable. The question for efficiency advocates is whether estimates of the lifecycle gains in efficiency warrant attention to these features, as an alternative to or complement to increased SEER and EER stringency. Finally, there are the market issues: how would conforming products be identified for the consumer and the contractor, and how would the value of the features be established for them?

[16] In practice, the simulation results can be transformed from one base case to the other, to a very good approximation.
[17] This was the subject of our Task 1 work.

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APPENDIX A: FINAL TRIAL FEATURE SET

Sustained High Performance from Residential Central Air Conditioners

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Introduction and Background

"Air conditioner compressors don't die, they are killed by bad installations." — Anon.

The efficiency of residential central air conditioners in the field is often lower than the SEER rating would suggest, because units are installed or operated at less than ideal conditions. The robust[18] CAC defined by this project will maintain its efficiency over a wider range of operating conditions and be more tolerant of installation and servicing deficiencies. This is true even for new SEER 13 minimum seasonal efficiency rating takes effect, because the laboratory tests and protocol have substantial limitations in predicting performance of units in the field. In this work, we define a "robust" air conditioner as one designed to assure savings persistence for the life of the equipment, by responding to slow refrigerant leaks, clogged filters or heat exchangers, or other operating anomalies.

This appendix outlines the important features of the robust air conditioner in the form of equipment specifications and field deviations from these specifications (see Table A-1).

[18] "Robust" is a provisional term for use while the concept is evaluated. Under laboratory test conditions using the DOE-approved methods, the robust air conditioner is not necessarily more efficient than a conventional unit. It is differentiated by its resistance to degradation over time under abusive conditions In the field.

Table A-1. Simulation Elements. This is the set of modeled characteristics, and the amount by which each was changed in the simulations.

Input Parameters		**Robust**	**Baseline**	**Comments**
SEER		14	14	Note A
EER				
	single-speed compressor	11.5	11.5	Note A
	multi-speed compressor	11	11.5	Note A
Changes to Simulate		**Acceptable Performance Re: Specification**	**Simulated Defect in Parameter**	
Refrigerant Charge Errors		90%	+/- 20%	Note B
Slow Refrigerant Leak		95%	3 yr. cycle	Note C
Air Handler Adaptive Capabilities Re: Ducts		95%	+/- 15%	Note D
Cabinet Air Leakage		99%	+/- 4.5%	Note E
Air Filter Rack and Tight Door		100%	10%	Note F
Filter Clogging Airflow Reduction		+/- 0.15"	+/- 0.3"	Note G
Call for Service, Other		90%	20%	Note H

Notes:

A	ENERGY STAR Tier 1. See http://energystar.gov/ia/partners/prod_development/revisions/downloads/ac_ashp/Final_CAC_ASHP_Eligibility_Criteria.pdf.
B	Considered maximum feasible performance level.
C	Model system recharge every 36 months. Lose charge at the rate of 10% per year. For example, 0% lost charge in year 1, 10% charge reduction in year 2, 20% reduction in year 3, 0% charge reduction in year 4 (required recharge), etc. 20% charge reduction for a standard unit would yield roughly a 20% loss in total capacity, which would probably cause a service call and refrigerant recharge.
D	Ignore this note; issue resolved.
E	Considered feasible with good manufacture and good installation.
F	15% leakage if door missing (Cummings 2005)
G	We are expressing this in terms of static pressure range (e.g., 0.1 – 0.9 IWG) over which airflow has to remain within some percentage.
H	If efficiency or capacity drops to 90%, then the AC calls for service. Baseline unit shuts down after falling to 80%.

Deleted:

Condenser fouling and blockage	95%	+/- 5%	Will 5% airflow reduction show up in fan watts, compressor watts, or both?

These specifications will be used as inputs to computer simulations to study the benefits of robust units. The simulations will also allow us — and the broader community, including AC equipment manufacturers — to choose and prioritize the features that will make the greatest difference in performance and in value perceived by the customer.

Discussion of Features

Performance Ratings. The baseline unit for comparison to the robust unit is the ENERGY STAR Tier 1 specification (SEER 14; EER 11.5). The robust unit meets the specification..[19] This allows market actors to see robust as a potential added feature for the ENERGY STAR specification.

Our discussion is confined to air conditioners and heat pumps in cooling mode. We do not consider heating performance of heat pumps. Because matching of the heat exchangers becomes more critical with heat pumps, we highly recommend this as a future extension of this project.

Refrigerant Charge Errors. The average air conditioner in the field shows significant deviations from the manufacturer's refrigerant charge specification, with over-charging about as common as undercharging (Mowris, Blankenship, and Jones 2004; Neme, Proctor, and Nadel 1999). The robust unit will require the use of feedback-controlled liquid metering devices[20] that enable the unit to maintain at least 90% of its capacity and efficiency at peak conditions, with refrigerant charge at 80%? and 120% of the manufacturer's specification.[21] We expect this to be a "weak" requirement for 2006, because we expect that virtually all SEER 13 and higher units for 2006 and beyond will incorporate TXVs or better, as an inexpensive route to higher performance.

Slow Refrigerant Leak. Some residential central air conditioners have very slow refrigerant leaks. We will model the performance degradation that ensues as a time series: full charge down to failure (less than 80% of refrigerant remaining) in 36 months, then re-charge and leak again, iterating for fifteen years.[22] In contrast, the robust unit is modeled as having a reduction of this degradation from its abilities to compensate for modest leaks (see Refrigerant Charge Errors, above). The robust unit will alarm when the charge loss is great enough that the unit can no longer maintain performance.[23]

Air Handler Adaptive Capabilities. Adaptive variable speed motor that maintains 400 cfm/ton (or other number specified by the manufacturer), to satisfy the design Sensible Heat Ratio (SHR) requirement across an external static pressure range of 0.1–0.9 inches water pressure. Motor shall have current limiting or temperature rise control to prevent significant

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[19] See http://energystar.gov/ia/partners/prod_development/revisions/downloads/ac_ashp/Final_CAC_ASHP_Eligibility_Criteria.pdf.

[20] Thermostatic expansion valves, electronic expansion valves, etc., or an equivalent that meets the performance specification.

[21] This language is adapted from CEC (2001). One effect of these parameters is to further encourage the use of TXV or equivalent expansion controls. Some form of active or "feedback" expansion device is required so the unit can adapt to variations in refrigerant charge, air flow, and even outdoor temperature.

[22] Our proposed leak and recharge sequence is requires recharge every 36 months (3 years). The unit would lose charge at the rate of 10% per year. For example, 0% lost charge in year 1, 10% charge reduction in year 2, 20% reduction in year 3, 0% charge reduction in year 4 (required recharge), etc. 20% charge reduction for a standard unit would yield roughly a 20% loss in total capacity, which would probably cause a service call and refrigerant recharge.

[23] This specification will be self-certified by performance with reduced charge; we do not anticipate field measurements.

life reduction when operated in high external static pressure system. This feature is required since so many houses have very high external static pressures. This can be caused by high-MERV air filters, or duct design and installation problems. A variable speed motor is inherently more efficient when moving the same amount of air as a conventional (permanent split capacitor) motor. When system pressures require more work, the adaptive variable speed motor can maintain flow across the coil. Where the duct system works, the variable speed motor may save several hundred kWh/yr. If the duct system is poor, the variable motor can still deliver comfort, although with increased energy consumption.

Air Handler Filter Blockage. Failure to change air filters when necessary is common, at least anecdotally. The results for conventional (PSC fan) units are decreased airflow, and (probable) lower SHR (greater dehumidification). For ECM fan motor air handlers, feedback mechanisms result in airflow being maintained (up to the capacity of the motor), but power consumption increases. The robust unit requires an alarm signal at the thermostat when filter blockage is significant, to indicate its time to change the filter (or call service). We will simulate the absence of the alarm, and consequent operation with partially blocked filter, by estimating the percentage degradation and the fraction of the time the unit operates in this condition. We will use 10% and 15% flow reduction (360 and 340 cfm/ton, respectively), and are considering one additional low-flow level (320 cfm/ton), which would correspond to the airflow associated with increasing likelihood of coil freezing.

Cabinet Air Leakage. Air leakage of the factory-specified air handler and evaporator assembly shall be less than 1% of rated airflow, compared with typical values averaging 4.5% average.[24] This is particularly important for units installed outside the thermal envelope (attics and attached garages, for example). In addition, the unit shall be shipped with or include a filter holder with tightly-sealing access door.[25]

*Alarms***.** To signal trouble such as filter change required, the unit is required to have alarm displays on the thermostat.[26] It must display different messages for at least the following problems:

- *Replace or service air filter*. This could be sensed by too much pressure drop across the filter, by downstream particle scattering, by run hours=800 (or other value), or by some other method chosen by the manufacturer.
- *Call for Service*. The goal is an alarm that tells the occupant to get help when the unit cannot keep itself within its proper operating band. It may, but need not, display a specific telephone number of the manufacturer, dealer, or other entity. The alarm requirement is not waived if the equipment automatically notifies service personnel. Conditions under which the alarm must be indicated include:
    - Static pressure drop has increased or decreased more than 15%, from initial setting. This could indicate need for filter change, dirt on evaporator coil, or

[24] Average value from Cummings (2005).
[25] According to Cummings (2005), missing air filter pocket covers may allow leakage of 15% of total air flow, and this will often be from outside the thermal envelope.
[26] This acknowledges that specific thermostat models may be required for the robust system from a given manufacturer.

pressure drop due to disconnect of one or more ducts. Must also alarm if the fan can't maintain specified cfm/ton air supply because ESP is high enough that temperature rise in the fan motor will significantly reduce its expected lifetime.
  - System needs service (other).
  - Low refrigerant charge (feedback from existing low refrigerant pressure sensor? or add an intermediate (non-fatal) low pressure signal that provides a warning that the low pressure cutout is being approached)?

These factors are simulated by estimating frequency in the population and severity of impact for the unit.

*Maintenance***.** The goal is to establish expectation of exemplary performance characterized by:

- No scheduled maintenance except filter replacements for 15 years.[27]
- Unless unit is installed within 10 miles of saltwater coast,[28] it is feasible to offer a full warranty for 15 years, contingent on filter replacement and response to alarms.[29]

The simulation of other parameters discussed in this document shows how performance degradations can be avoided by this specification, so this reliability and performance level can be achieved.

Discussion

These features should make the units robust, in the sense that the manufacturer will have provided all features that can be reasonably designed into the product to assure that it will operate efficiently over its life time. We believe that these robust requirements are the minimum set for an air conditioner to actually deliver high efficiency for many years, across a wide range of outdoor temperatures, given commonly encountered refrigerant charging errors and the most common deviations from airflow at rated conditions. If a unit meets these requirements, a failure to efficiently meet comfort requirements strongly implies major installation errors: a grossly defective shell or ductwork, or a serious miscalculation of the unit capacity required to meet the building's load.

[27] This will be pitched as an effort to get refrigerator-type reliability from a split system. At first, manufacturers will take an actuarial (insurance) approach to the 15 year life issue. At least some manufacturers purchase insurance as a way to handle out-year warranty claims. However, the best will quickly decide to design and manufacture for better quality, and thus reduce warranty and insurance costs.
[28] The near-coastal exemption is required: current construction methods do poorly in salt air.
[29] This will be a challenge for OEMs, but represents the most visible value added to the consumer. Discussions continue in this area.

APPENDIX B. TOPICS NOT INCLUDED IN THE ROBUST SPECIFICATION

The robust concept does not address some performance issues that can only be fixed by the contractor. These include proper equipment sizing, selection and setup, and installing tight ductwork that has low external static pressure and delivers (and returns) adequate air to (and from) each space in the building. The robust specification addresses some of these issues tangentially, such as the inclusion of requirements that the unit adapt to a wide range of ESPs.

Oversizing

Neal and O'Neal (1992) carried out a lab study of refrigerant charging and capacity errors (oversizing). They took an optimistic perspective, assuming no cycling losses when the equipment is running only part of the time. They studied a single line of (now obsolete) capillary-tube equipment, but the results should be fairly representative for short-orifice metering systems, too. Table B-1 summarizes their results

Table B-1. Variation in Utility (Coincident) Load with Oversizing and Refrigerant Charge Variations

Unit Capacity tons	Size, re actual load	Demand	Charge, re mfg. spec. 80%	100%	120%
3	100%	Utility kW	3.86	4.20	4.33
		% of correct	92%		103%
3.5	124%	Utility kW	4.83	4.25	5.03
		% of correct	114%		118%
5	175%	Utility kW	5.17	4.33	5.09
		% of correct	119%		118%

Source: Neal and O'Neal (1992)[30]

This robust specification indirectly addresses oversizing through its requirement that the air handler be capable of delivering design airflow across a range of external static pressures, 0.1–0.9 IWG. Otherwise, we expect that almost all equipment manufactured in 2006 and later will use TXVs (thermostatic expansion devices) or better feedback-controlled liquid refrigerant metering devices that are less susceptible to these problems than fixed orifices.

Modulating cooling capacity. As one possible response to oversizing, we considered requiring modulating cooling capacity in the robust specification, but decided that this premium feature should first be analyzed elsewhere in this project. Pigg (2003) has shown that 2-stage condensing furnaces largely remove the performance penalties associated with oversized furnaces, but running on low-fire almost all the time. With appropriate airflow

[30] Values are for a 34,000 Btuh actual building load. In Column 2, 124% corresponds to ACCA Manual J sizing.[30] 175% is typical oversizing. In Column 3, Demand is expressed in coincident utility kW (not site load), and in percentage of utility load for under- and over-charged units. Data are for 95ºF. The authors also give data for 100ºF, which show higher utility impacts.

controls, this may also be true for air conditioning, where latent heat control presents a challenge that heating equipment does not face.

Adaptive sensible heat ratio. We considered requiring that robust air conditioners have the ability to change their ability to remove moisture more than the ability inherent in conventional machinery. The value of such a feature in the chronically humid parts of the US is unquestioned, because it can help prevent mold problems and improve comfort. Nonetheless, we decided not to include this as a core feature. Where dehumidification is a pervasive need, specialized equipment may be a better approach.[31] Where enhanced humidity control is a short-term, acute problem, there is much lower risk that mold will damage the equipment or conditioned spaces.

Condenser fouling and blockage. In the field, grass clippings, fallen leaves, and other debris can partially block condenser airflow. Blockage, fouling, or obstructions are expected to be most severe in the lowest parts of the condenser. We considered requiring that the robust specification require the unit to maintain at least 95% of its capacity and efficiency at peak conditions, with the bottom 10% of the condenser face blocked. This is purely a performance specification, which could be met with several alternative solutions. (1) Some firms may choose to rate equipment with "oversized" condenser coils and good controls. (2) A variable speed condenser fan could increase airflow in response to system performance indicators. (3) Variable fin spacing, with wider spacing in the bottom where more debris is expected, might help. Testing by Henderson (2001) suggests that fouling of air-cooled condenser coils degrades performance by about 8% (+/- 3%), so the effect is worth controlling.[32]. Alternatively, a prescriptive route could be offered, giving credit for shipping and installing the unit with a substantially elevated condenser. In the end, we decided to delete this requirement.

[31] Our STAC team is addressing this issue in Task 4, the development of an air conditioner for hot-humid climates.
[32] This value is from a study of DX chillers, but Henderson regarded the value as relatively solid.

APPENDIX C: HOURLY SIMULATIONS TO QUANTIFY THE PERFORMANCE BENEFITS OF ROBUST AC

The TRNSYS model developed for a Home Energy Rating System (HERS) Reference house, which is described in the Task 4 report, was also used for this analysis of the robust AC. The TRNSYS model is used to evaluate the annual average energy impacts of the various robust AC improvements.

House Details

This analysis used the HERS reference house from the Task 4 simulations. The 2000 sq ft slab-on-grade house includes natural infiltration (ELA= 138.24), duct leakage and thermal losses to the attic space (4% return, 3% supply, R-6 duct insulation). Simulations were completed for the cities listed below with a 75°F cooling set point. All cities with include house characteristics for the proper IECC zone except Chicago (which used IECC Zone 4 building characteristics instead of Zone 5)

Table C-1. Weather Cities and IECC Construction Details

	Weatherfile	IECC Zone	SinZone_bno
1	Miami-FL	1	7
2	Atlanta-GA	3	9
3	Sterling-VA	4	10
4	Houston-TX	2	8
5	Fort_Worth-TX	3	9
6	Chicago-IL	5	10
7	Phoenix-AZ	2	8
8	New_York_City-NY	4	10
9	San_Francisco-CA	3	9
10	Fresno-CA	3	9

Note: SinZone_bno is keyword corresponding the IECC construction details in each IECC zone.

Air Conditioner Details

The air conditioner used in this analysis was assumed to have a "gross" EER of 16.1 Btu/Wh at the nominal rating point[33] based on gross coil capacity and without considering the supply fan power. The SHR of the cooling coil at the nominal rating point is 0.77. By default, the unit has performance characteristics typical of a system that uses a TXV refrigerant expansion device and reciprocating compressor. The degradation coefficient is assumed to be 0.12. Using a supply fan power of 0.350 Watts per cfm — which corresponds to a conventional fan motor with an external static pressure of approximately 0.10-0.15 inches — the SEER for this unit is 14.0 Btu/Wh. Single-stage operation is assumed for all cases. The characteristics for the AC unit are summarized in the table below.

[33] Nominal rating point (or, the condition where a 1-ton unit provides 12,000 Btu/h of gross cooling capacity) is 95°F outdoors, 80°F/67°F entering coil and 450 cfm per ton supply air flow.

Table C-2. Characteristics of Air Conditioners Used in Analysis

	Rated Gross[1] EER (Btu/Wh)	Supply Fan Power (W/cfm)	SEER (Btu/Wh)	Rated EER (Btu/Wh)	Rated SHR (-)
Conventional AC	16.1	0.350	14.0	12.8	0.752

In reality, external static pressures are much higher than 0.15 inches. Therefore, these runs we used a fan power of 0.42 Watts per cfm, for the default case of 400 cfm per ton with a PSC motor. This corresponds to an external static pressure (ESP) of 0.5 inches according to laboratory measurements taken by Wilcox et al. (2006).[3]

Impact of Refrigerant Charge

Davis and D'Albora (2001a, 2001b) collected detailed laboratory data for Pacific Gas Electric on the impact of various levels of refrigerant charge. This series of laboratory tests looked at the impact of refrigerant charge on capacity and efficiency over a range of ambient conditions as well as at wet-coil and dry coil conditions. Tests were completed for R410a and R22 units. When the data for the R22 units were normalized, there was very little difference between wet and dry coil performance. Similarly, the relative performance variations were not strongly a function of ambient temperature. The performance variations were reduced to the form shown in the figures below for the normalized EER and cooling capacity. A similar curve was also developed for sensible heat ratio (SHR) for the orifice unit (the SHR for the TXV units was not a strong function of charge). This analysis is given in Appendix D.

Figure C-1. Comparing Quadratic Curve Fit to Normalized EER Data for Wet Coil Conditions

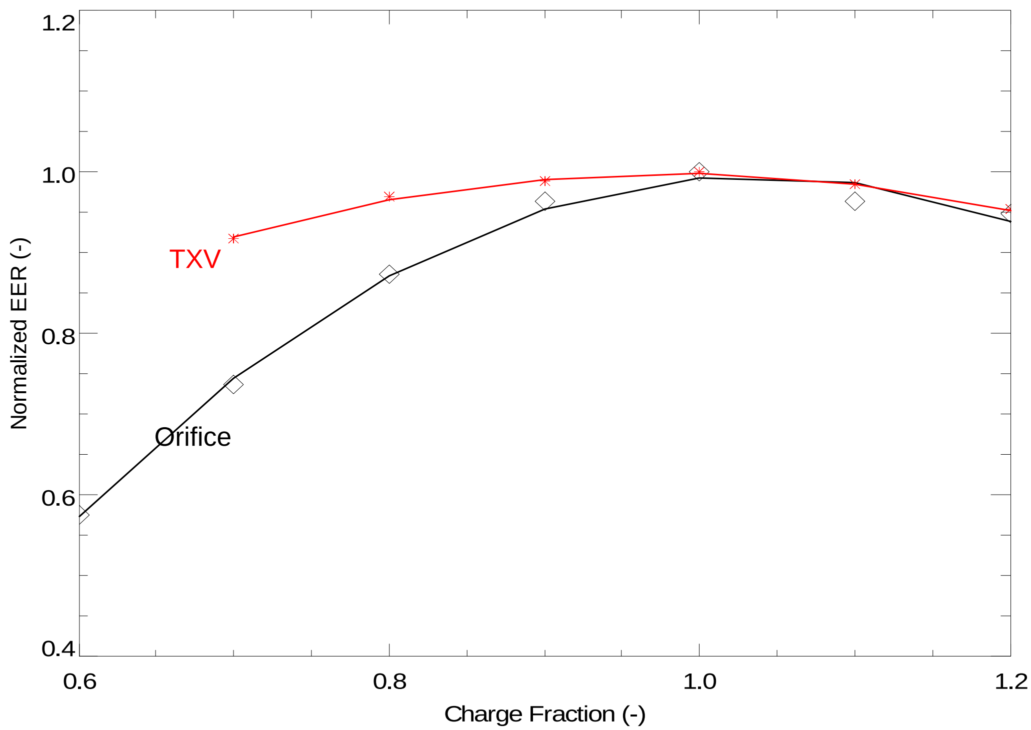
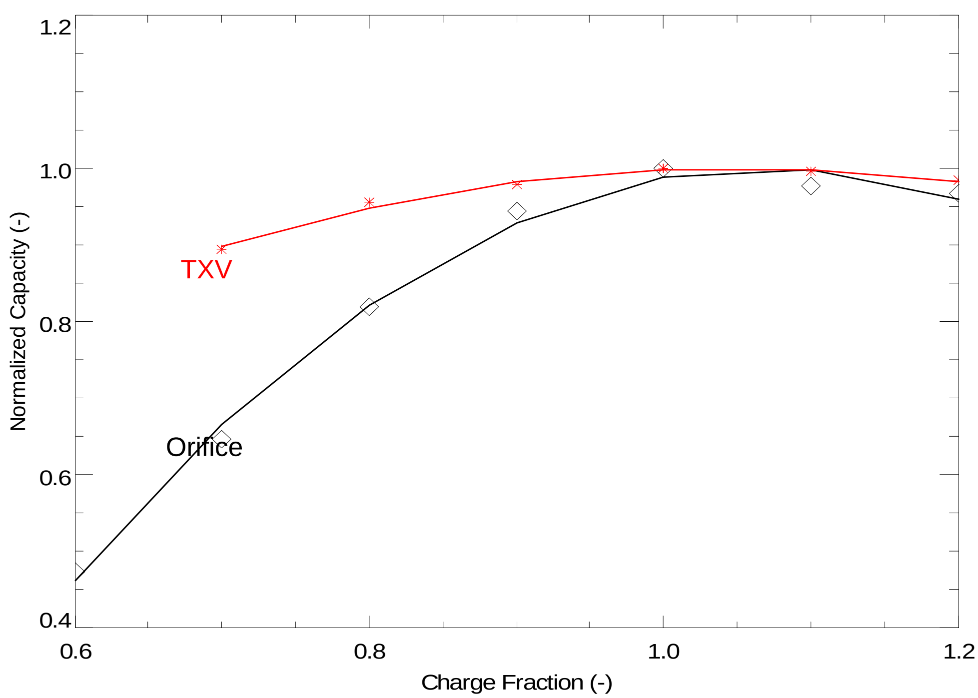


Figure C-2. Comparing Quadratic Curve Fit to Normalized Capacity Data for Wet Coil Conditions



The AC model is based on curve-fits of results from Heat Pump Design Model (HPDM) available at ORNL's web site.[34] HPDM is a detailed, air conditioner model that considers refrigerant-side and air-side performance details. The model was run in the parametric mode with different compressor and expansion device specified. The parametric runs varied ambient temperature, airflow and entering wet bulb over a range of conditions. Regression models were developed to predict total capacity and EER as a function of these parameters. The SHR was determined using the ADP/BF approach developed by Henderson, Rengarajan, and Shirey (1992). The development of this model is discussed in detail in Appendix D.

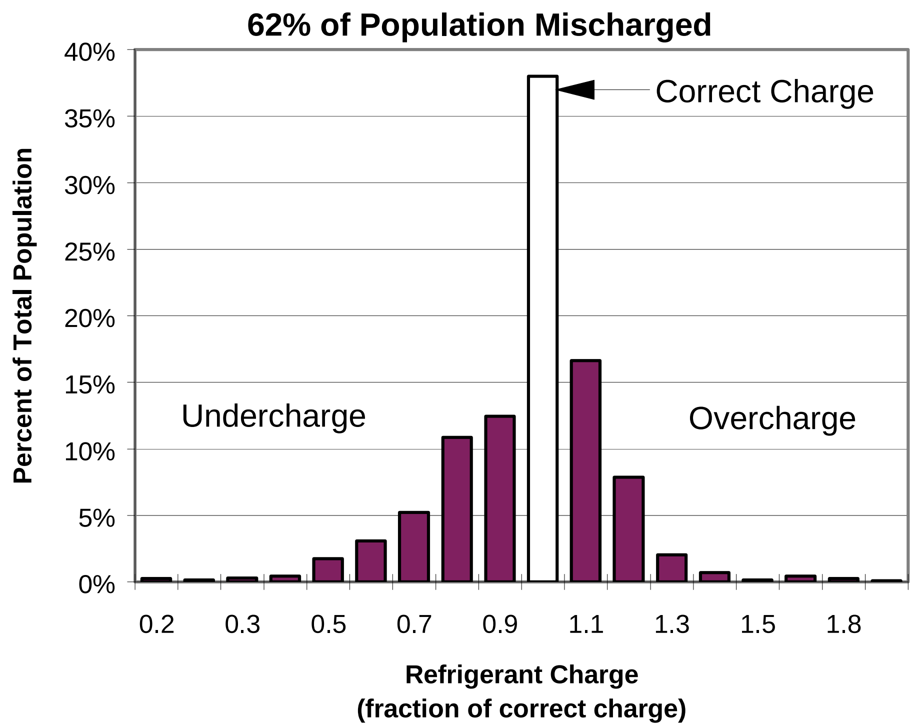
Table C-3 shows the range of operating conditions and system configurations considered in the analysis. The weighting factors in Figure C-3 were applied to the simulation results to predict the overall impact of mischarged units on overall energy use for the total population.

Table C-3. Range of Air Conditioner Operating Conditions/Configurations Considered in Simulation Analysis

	Charge Ratio (chrg_ratio)	AC Type (ac_coef)	Expansion Device (exp_type)
TXV, Reciprocating Compressor	0.7-1.2	4	2
TXV, Scroll Compressor	0.7-1.2	2	2
Orifice, Reciprocating Compressor	0.7-1.2	5	1

[34] HPDM at available at http://www.ornl.gov/sci/btc/apps/hpdm.html

Figure C-3. Distribution of Mischarged Units from the CheckMe! Database



Charge Ratio	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
Population Fraction	0.02	0.03	0.05	0.11	0.12	0.37	0.17	0.08	0.02	0.01

Source: Proctor Engineering Group, Ltd. (2000)

Simulations were run for the scenarios listed above for the ten different cities. The results of the simulations for each city are listed in Table C-4. The results for Miami are shown graphically in Figure C-4. The energy use is lowest when the air conditioner is at 100% of the full refrigerant charge. Both over-charging and under-charging has an energy penalty. This penalty is mitigated when a TXV is used. The orifice unit shows a greater energy impact of being mischarged. The orifice unit in Miami uses 39% more energy when the charge fraction is 70% and used 5% more energy at a chare fraction of 120%. In contrast, the corresponding increases in energy use for the TXV unit are 9% and 5% at charge fractions of 70% and 120%.

Figure C-5 compares the energy use increase for the Orifice unit at 70% charge for the ten different locations. The increase in energy use at 70% charge ranges from 31 to 41% in Chicago and Phoenix, respectively. The variation occurs because the supply fan becomes a proportionally larger part of the annual electric load in cities where significant fan operation happens during the heating season (i.e., the total electric use includes fan operation in the heating mode. Heating is provided by another fuel.). Indeed, this is a bit of a drawback in the current test regime: neither the furnace nor the air-conditioner test fully accounts for this energy use.

Table C-4. Simulation Results Showing Impact of Refrigerant Charge on Annual Energy Use

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	754	2,336.2	24.7	2,360.9	4,516	1,111	5,627	109%
TXV, 80% Charge	737	2,201.7	24.7	2,226.4	4,307	1,047	5,355	104%
TXV, 90% Charge	725	2,122.6	24.7	2,147.2	4,206	1,010	5,216	101%
TXV, 100% Charge	722	2,082.5	24.6	2,107.2	4,182	991	5,173	100%
TXV, 110% Charge	722	2,083.4	24.6	2,108.0	4,257	992	5,249	101%
TXV, 120% Charge	728	2,120.7	24.7	2,145.3	4,419	1,009	5,428	105%
Orifice, 70% Charge	475	3,361.8	24.7	3,386.5	5,597	1,593	7,190	139%
Orifice, 80% Charge	551	2,654.8	24.7	2,679.5	4,835	1,260	6,096	118%
Orifice, 90% Charge	646	2,280.0	24.7	2,304.7	4,392	1,084	5,476	106%
Orifice, 100% Charge	743	2,079.8	24.7	2,104.5	4,178	990	5,168	100%
Orifice, 110% Charge	856	2,054.2	24.7	2,078.9	4,197	978	5,175	100%
Orifice, 120% Charge	967	2,119.9	24.7	2,144.6	4,393	1,009	5,401	105%

Atlanta	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	192	871.5	456.6	1,328.1	1,680	625	2,304	107%
TXV, 80% Charge	192	823.9	456.7	1,280.6	1,606	602	2,209	103%
TXV, 90% Charge	192	794.9	456.9	1,251.8	1,569	589	2,158	101%
TXV, 100% Charge	193	781.1	457.1	1,238.2	1,562	582	2,145	100%
TXV, 110% Charge	193	781.6	456.8	1,238.4	1,591	583	2,173	101%
TXV, 120% Charge	193	794.6	456.8	1,251.4	1,650	589	2,238	104%
Orifice, 70% Charge	132	1,267.3	457.0	1,724.2	2,089	811	2,900	136%
Orifice, 80% Charge	150	988.9	457.1	1,446.0	1,792	680	2,472	116%
Orifice, 90% Charge	174	852.0	457.0	1,309.0	1,633	616	2,249	105%
Orifice, 100% Charge	196	778.3	457.1	1,235.4	1,555	581	2,136	100%
Orifice, 110% Charge	207	767.8	456.8	1,224.7	1,560	576	2,136	100%
Orifice, 120% Charge	222	789.7	456.7	1,246.4	1,628	586	2,214	104%

Sterling	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	49	742.8	515.1	1,257.9	1,575	655	2,230	107%
TXV, 80% Charge	47	701.1	514.9	1,216.0	1,504	633	2,137	103%
TXV, 90% Charge	46	676.1	515.1	1,191.2	1,469	620	2,089	101%
TXV, 100% Charge	47	664.1	515.0	1,179.1	1,462	614	2,076	100%
TXV, 110% Charge	45	664.3	515.0	1,179.4	1,488	614	2,102	101%
TXV, 120% Charge	47	675.9	515.1	1,191.0	1,544	620	2,164	104%
Orifice, 70% Charge	14	1,080.1	514.9	1,595.0	1,961	831	2,792	135%
Orifice, 80% Charge	23	842.6	514.9	1,357.4	1,680	707	2,387	115%
Orifice, 90% Charge	36	725.0	514.8	1,239.9	1,529	646	2,175	105%
Orifice, 100% Charge	47	662.0	515.0	1,177.1	1,455	613	2,068	100%
Orifice, 110% Charge	69	653.3	515.1	1,168.4	1,461	608	2,069	100%
Orifice, 120% Charge	90	672.8	514.8	1,187.7	1,527	618	2,145	104%

Note: All units use reciprocating compressors.

Table C-4. Simulation Results Showing Impact of Refrigerant Charge on Annual Energy Use (cont.)

Houston	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	1,031	1,523.7	202.3	1,725.9	3,194	870	4,064	109%
TXV, 80% Charge	1,022	1,436.5	202.3	1,638.8	3,047	826	3,873	103%
TXV, 90% Charge	1,017	1,384.8	202.4	1,587.1	2,975	800	3,775	101%
TXV, 100% Charge	1,015	1,359.0	202.3	1,561.3	2,958	787	3,745	100%
TXV, 110% Charge	1,014	1,358.7	202.3	1,561.0	3,009	787	3,796	101%
TXV, 120% Charge	1,019	1,384.1	202.3	1,586.4	3,126	800	3,925	105%
Orifice, 70% Charge	790	2,198.4	202.2	2,400.6	3,958	1,210	5,168	138%
Orifice, 80% Charge	880	1,729.4	202.2	1,931.6	3,412	974	4,386	117%
Orifice, 90% Charge	942	1,485.9	202.1	1,688.0	3,100	851	3,951	106%
Orifice, 100% Charge	1,023	1,355.4	202.2	1,557.6	2,949	785	3,734	100%
Orifice, 110% Charge	1,103	1,338.9	202.1	1,541.0	2,962	777	3,739	100%
Orifice, 120% Charge	1,182	1,381.0	202.3	1,583.3	3,099	798	3,897	104%

Fort Worth	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	132	1,406.9	290.4	1,697.4	3,049	884	3,933	108%
TXV, 80% Charge	131	1,327.2	290.3	1,617.5	2,911	842	3,753	103%
TXV, 90% Charge	131	1,279.0	290.5	1,569.5	2,842	817	3,659	101%
TXV, 100% Charge	131	1,255.7	290.3	1,546.1	2,827	805	3,632	100%
TXV, 110% Charge	131	1,256.4	290.4	1,546.8	2,878	806	3,684	101%
TXV, 120% Charge	131	1,278.6	290.3	1,568.9	2,986	817	3,803	105%
Orifice, 70% Charge	78	2,001.2	290.3	2,291.5	3,714	1,193	4,907	136%
Orifice, 80% Charge	95	1,592.6	290.5	1,883.0	3,236	981	4,217	117%
Orifice, 90% Charge	115	1,371.2	290.5	1,661.7	2,948	865	3,813	106%
Orifice, 100% Charge	131	1,249.8	290.3	1,540.1	2,804	802	3,606	100%
Orifice, 110% Charge	159	1,233.7	290.2	1,523.9	2,816	794	3,609	100%
Orifice, 120% Charge	184	1,270.6	290.5	1,561.1	2,941	813	3,754	104%

Chicago	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	76	665.8	783.5	1,449.3	1,256	682	1,938	107%
TXV, 80% Charge	76	629.4	783.7	1,413.1	1,201	665	1,866	103%
TXV, 90% Charge	76	607.1	783.8	1,390.9	1,173	654	1,828	101%
TXV, 100% Charge	76	596.0	783.8	1,379.8	1,167	649	1,816	100%
TXV, 110% Charge	76	596.5	783.6	1,380.1	1,189	649	1,838	101%
TXV, 120% Charge	76	607.0	783.6	1,390.7	1,234	654	1,888	104%
Orifice, 70% Charge	65	961.7	783.4	1,745.1	1,553	821	2,374	131%
Orifice, 80% Charge	69	756.4	783.6	1,540.0	1,341	724	2,065	114%
Orifice, 90% Charge	74	651.2	783.8	1,435.0	1,221	675	1,896	105%
Orifice, 100% Charge	78	594.3	783.6	1,377.9	1,162	648	1,810	100%
Orifice, 110% Charge	82	585.9	783.6	1,369.5	1,166	644	1,810	100%
Orifice, 120% Charge	89	602.6	783.6	1,386.1	1,216	652	1,868	103%

Note: All units use reciprocating compressors.

Table C-4. Simulation Results Showing Impact of Refrigerant Charge on Annual Energy Use (cont.)

Phoenix	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	-	1,725.7	117.5	1,843.1	5,013	1,239	6,252	108%
TXV, 80% Charge	-	1,629.6	117.5	1,747.1	4,790	1,174	5,964	103%
TXV, 90% Charge	-	1,573.0	117.4	1,690.4	4,683	1,136	5,819	101%
TXV, 100% Charge	-	1,544.8	117.5	1,662.3	4,659	1,117	5,777	100%
TXV, 110% Charge	-	1,545.2	117.5	1,662.7	4,742	1,117	5,859	101%
TXV, 120% Charge	-	1,571.8	117.5	1,689.3	4,917	1,135	6,052	105%
Orifice, 70% Charge	-	2,527.4	117.6	2,644.9	6,239	1,777	8,017	141%
Orifice, 80% Charge	-	1,989.3	117.5	2,106.8	5,364	1,416	6,779	119%
Orifice, 90% Charge	-	1,697.5	117.5	1,815.0	4,834	1,220	6,054	106%
Orifice, 100% Charge	-	1,541.4	117.4	1,658.8	4,575	1,115	5,690	100%
Orifice, 110% Charge	-	1,534.4	117.4	1,651.8	4,624	1,110	5,734	101%
Orifice, 120% Charge	-	1,590.0	117.4	1,707.5	4,852	1,147	6,000	105%

New_York_City	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	30	773.3	589.1	1,362.4	1,451	641	2,092	107%
TXV, 80% Charge	30	730.7	589.3	1,320.0	1,387	621	2,008	103%
TXV, 90% Charge	30	705.2	589.1	1,294.3	1,355	609	1,964	101%
TXV, 100% Charge	29	692.7	589.1	1,281.8	1,349	603	1,952	100%
TXV, 110% Charge	30	693.1	589.2	1,282.3	1,374	603	1,977	101%
TXV, 120% Charge	30	704.9	589.4	1,294.3	1,425	609	2,034	104%
Orifice, 70% Charge	15	1,123.6	589.1	1,712.7	1,808	806	2,614	134%
Orifice, 80% Charge	19	877.4	589.1	1,466.6	1,550	690	2,239	115%
Orifice, 90% Charge	23	755.6	589.1	1,344.7	1,411	633	2,043	105%
Orifice, 100% Charge	30	690.4	589.3	1,279.7	1,344	602	1,946	100%
Orifice, 110% Charge	34	681.1	589.2	1,270.4	1,348	598	1,946	100%
Orifice, 120% Charge	40	701.0	589.2	1,290.2	1,408	607	2,015	104%

San_Francisco	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	-	1,193.2	327.4	1,520.6	2,587	792	3,379	108%
TXV, 80% Charge	-	1,126.1	327.3	1,453.4	2,471	757	3,228	103%
TXV, 90% Charge	-	1,084.9	327.3	1,412.2	2,412	736	3,148	101%
TXV, 100% Charge	-	1,065.3	327.3	1,392.6	2,400	725	3,126	100%
TXV, 110% Charge	-	1,065.8	327.3	1,393.0	2,443	726	3,169	101%
TXV, 120% Charge	-	1,084.8	327.4	1,412.3	2,535	736	3,271	105%
Orifice, 70% Charge	-	1,710.9	327.4	2,038.3	3,149	1,062	4,211	136%
Orifice, 80% Charge	-	1,359.2	327.3	1,686.5	2,739	878	3,617	117%
Orifice, 90% Charge	-	1,166.3	327.3	1,493.6	2,487	778	3,265	106%
Orifice, 100% Charge	-	1,060.9	327.3	1,388.2	2,362	723	3,085	100%
Orifice, 110% Charge	-	1,047.5	327.4	1,374.9	2,373	716	3,089	100%
Orifice, 120% Charge	-	1,081.0	327.3	1,408.4	2,483	734	3,217	104%

Note: All units use reciprocating compressors.

Table C-4. Simulation Results Showing Impact of Refrigerant Charge on Annual Energy Use (cont.)

Fresno	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
TXV, 70% Charge	-	1,050.4	299.6	1,350.0	2,599	794	3,392	108%
TXV, 80% Charge	-	991.9	299.6	1,291.5	2,483	759	3,242	103%
TXV, 90% Charge	-	956.5	299.6	1,256.2	2,425	739	3,164	101%
TXV, 100% Charge	-	939.5	299.6	1,239.0	2,413	729	3,142	100%
TXV, 110% Charge	-	939.5	299.6	1,239.1	2,455	729	3,184	101%
TXV, 120% Charge	-	956.4	299.6	1,256.0	2,549	739	3,288	105%
Orifice, 70% Charge	-	1,534.1	299.7	1,833.7	3,219	1,078	4,298	139%
Orifice, 80% Charge	-	1,201.1	299.6	1,500.7	2,762	882	3,644	118%
Orifice, 90% Charge	-	1,027.3	299.7	1,326.9	2,499	780	3,280	106%
Orifice, 100% Charge	-	935.4	299.6	1,235.0	2,375	726	3,101	100%
Orifice, 110% Charge	-	924.1	299.7	1,223.8	2,386	720	3,106	100%
Orifice, 120% Charge	-	953.1	299.7	1,252.7	2,497	737	3,233	104%

Note: All units use reciprocating compressors.

Figure C-4. Impact of Refrigerant Charge Fraction on Relative Energy Use for Miami

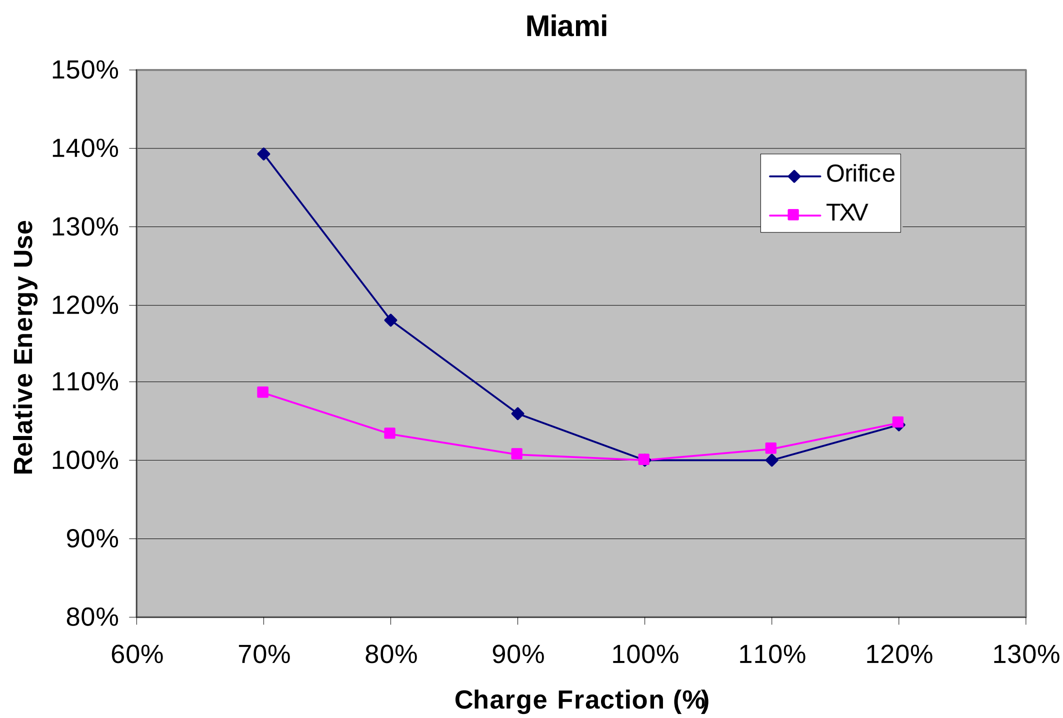
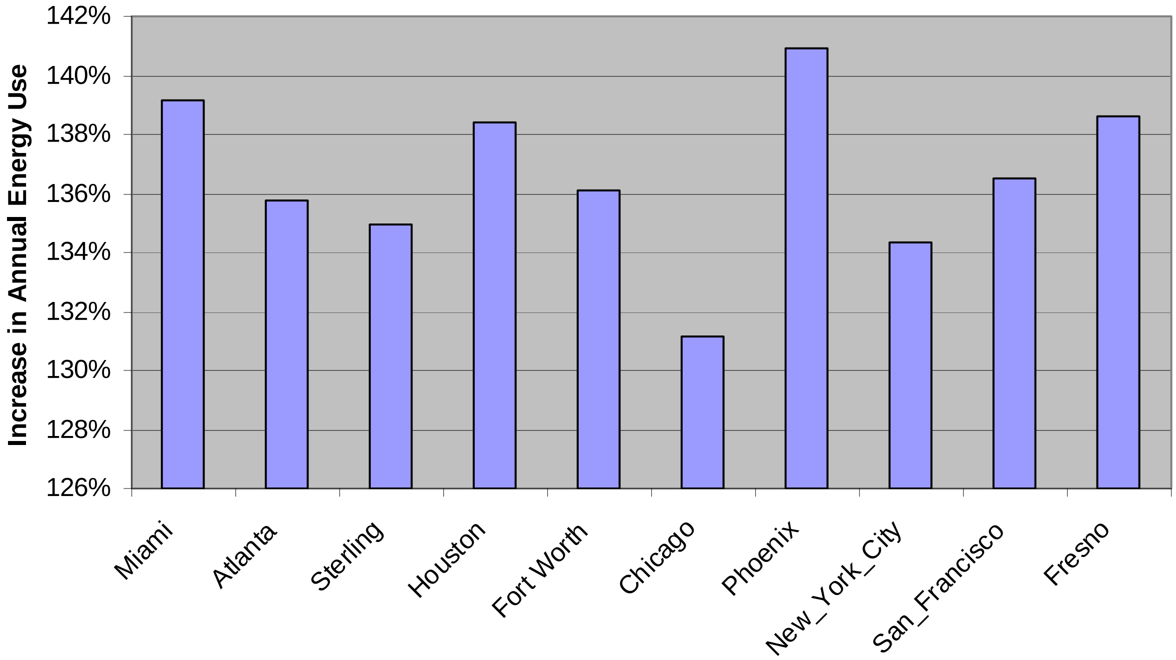


Figure C-5. Increased Energy Use at 70% Charge for Different Locations (Orifice, Reciprocating Compressor)



The data in Figure C-3 indicate what proportion of the population of air conditioners are at various levels of charge. This population data is used to "weight" the energy impacts corresponding to each level of charge in order to estimate the overall average energy impacts of having mischarged units in the field. Table C-5 shows the overall impact for each city for both the Orifice and TXV units. Combining the data from Figure C-3 with the data for Miami shown in Figure C-4 results in a weighted average increase of 5.6% in energy use for Orifice units compared to perfectly charged units. Similarly, the weighted average increase for the TXV units in Miami is 1.7%. If the entire population of units in Miami were to switch from an orifice to a TXV, the reduction in annual energy use would be 3.6%. Similar results were obtained for the other locations as well. The energy impact of converting from Orifice to TXV units ranges from a 4.2% reduction in Phoenix to a 2.9% reduction in Chicago.

Table C-5. Percentage Increase in Annual Energy Use for Population of Air Conditioners

	Weighted-Avg Orifice	TXV	Difference
Miami	5.6%	1.7%	-3.6%
Atlanta	4.9%	1.5%	-3.3%
Sterling	4.9%	1.5%	-3.2%
Houston	5.5%	1.7%	-3.6%
Fort Worth	5.2%	1.7%	-3.4%
Chicago	4.4%	1.4%	-2.9%
Phoenix	6.1%	1.6%	-4.2%
New York City	4.7%	1.4%	-3.1%
San Francisco	5.3%	1.6%	-3.5%
Fresno	5.5%	1.6%	-3.7%

Note: Using data at each charge fraction (e.g., from Figure C-4) with population weighting factors from Figure C-3.

$$\text{Difference} = \frac{(1 + \text{TXV\%})}{(1 + \text{Orifice\%})} - 1$$

A robust unit is assumed to provide an alarm signal once its refrigerant charge reaches 90%, so that a service technician would add refrigerant. In contrast a baseline AC unit is assumed to drop to 80% of full charge before a technician would add refrigerant. If we assume that an AC unit looses 10% of its charge each year, then the baseline unit would operate for 3 years without needing charge. The robust unit would be recharged every 2 years. The average energy use over the 3 year period for the baseline unit is the average of simulation runs corresponding to 100%, 90% and 80% charge fractions. The average energy use over the 2 year period for the robust unit is the average of simulation runs corresponding to 100% and 90% charge fractions.

Table C-6 shows the averaged energy use premium for these two scenarios. For the baseline unit with an orifice, the energy use premium (compared to a perfectly charged unit) is 6.3 to 8.5%, depending on the location. If the baseline unit has a TXV the energy use premium drops to 1.1-1.3%. In contrast, the robust unit with a TXV has much lower energy premium of 0.3-0.4%.

Table C-6. Percentage Increase in Annual Energy Use for Two Different Charging Scenarios

	Baseline (100-80%, 3 yrs)		Robust (100-90%, 2 yrs)
	Orifice	TXV	TXV
Miami	8.0%	1.4%	0.4%
Atlanta	7.0%	1.2%	0.3%
Sterling	6.9%	1.2%	0.3%
Houston	7.8%	1.4%	0.4%
Fort Worth	7.6%	1.4%	0.4%
Chicago	6.3%	1.1%	0.3%
Phoenix	8.5%	1.3%	0.4%
New York City	6.7%	1.2%	0.3%
San Francisco	7.7%	1.3%	0.4%
Fresno	7.8%	1.3%	0.3%

Adaptive Airflow

AHUs with ECM fans can typically vary fan speed in an attempt to maintain the desired airflow, regardless of static pressure. With an ECM motor, the fan power is typically lower so the SEER of unit with the ECM fan will be higher than 14 (assuming refrigeration system remains the same). We looked at two AC options:

1. Robust 14+ SEER unit with an ECM fan than maintains same airflow regardless of static pressure,

2. Baseline 14 SEER unit, with a PSC fan that rides the fan curve, or decreases flow as the external static increases.

The analysis was based on laboratory data on furnace fan performance collected by Proctor and Wilcox (2006) for the CEC (Wilcox 2006). The laboratory testing for one unit (Units 4 and 4A) was repeated with a Permanent Split Capacitor (PSC) motor as well as an Electronically Commutated Motor (ECM). The plots below compare the normalized airflow and power of the two motors as a function a various external static pressures. The nominal flow was assumed to correspond to an external static pressure of 0.5 inches.

The PSC motor behaves as a conventional system with the fan motor speed remaining constant (except for motor slip) as the static pressure changes. The ECM motor attempts to hold the airflow constant by increasing the fan speed as the static pressure increases. Figure C-6 and Figure C-7 show that the airflow changes less with a higher static pressure with the ECM motor (Figure C-6), however, the fan power increases (instead of decreasing) at higher static pressures (Figure C-7).

Figure C-6. Variation of Airflow with Static Pressure for PSC and ECM Motors

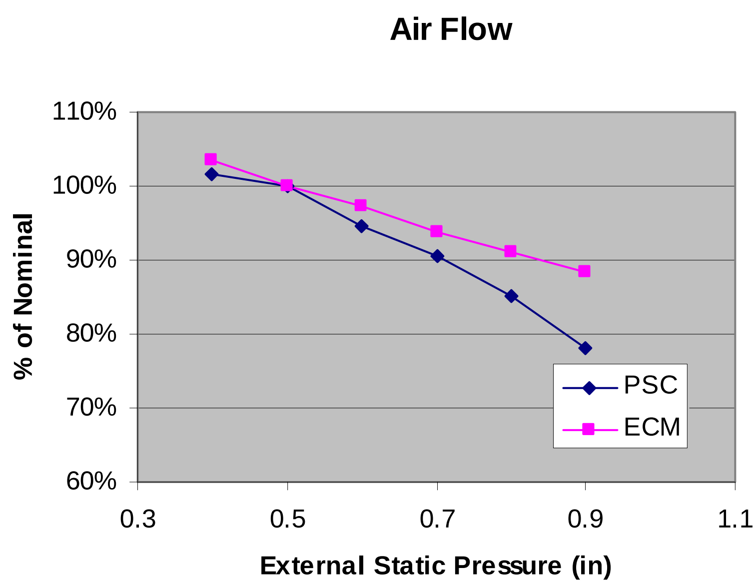


Figure C-7. Variation of Fan Power with Static Pressure for PSC and ECM Motors

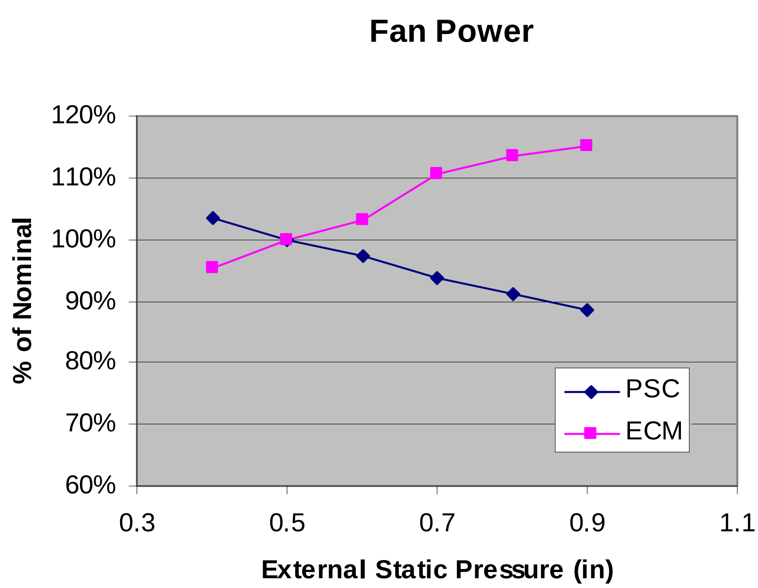


Table C-7 shows the variations in airflow and fan power at different external static pressures for the two fan motors. The ECM motor uses 30% less power at an external static pressure of 0.5 inches. At 0.7 inches, the savings decrease to 14% and at 0.9 inches the ECM actually uses 8% more power.

Table C-7. Variation of Flow and Fan Power at Different Air Pressures

	Ext Static (in)	Airflow (cfm/ton)	Fan Power (W/cfm)	Fan Power (W/ton)	
PSC	0.5	400	0.42	168	
	0.7	364	0.39	144	
	0.9	312	0.37	115	
ECM	0.5	400	0.30	118	70%
	0.7	376	0.33	123	86%
	0.9	331	0.38	125	108%

For these simulation cases, we assume that the duct leakage remains the same (in absolute terms) as the static pressure increases from 0.5 inches to the higher pressures.

Table C-8 shows the simulation results for both PSC and ECM motors using different static pressures in each city. Figure C-8 graphically shows the results for Miami. The PSC fan motor provides lower airflow and consumes less power as the external static pressure increases. The net result is that the annual energy use is essentially constant as the static pressure increases (since the compressor efficiency decreases at the lower airflow rate). At 0.5 inches of external static, the ECM motor uses 6% less total energy in Miami. However as the ECM motor speeds up to meet the higher static pressure, the energy savings from the ECM motor are reduced. At 0.9 inches external static, the energy use of the ECM motor increases by 6% so that it is equivalent to the PSC motor. Of course, one implication of the reduced airflow at higher ESP associated with the PSC is that it delivers less air to the room registers. That is, it may not meet the service needs, failing to meet a design specification.

Table C-8. Simulation Results Showing Impact of External Static Pressure on Annual Energy Use

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	722	2,082.5	24.6	2,107.2	4,182	991	5,173	100%
PSC, 0.7 inches, 0.39 W/cfm	653	2,145.9	27.1	2,173.1	4,267	864	5,131	99%
PSC, 0.9 inches, 0.37 W/cfm	535	2,271.8	31.7	2,303.5	4,442	745	5,186	100%
ECM, 0.5 inches, 0.30 W/cfm	739	2,064.3	24.7	2,088.9	4,149	702	4,851	100%
ECM, 0.7 inches, 0.33 W/cfm	681	2,111.7	26.3	2,138.0	4,215	743	4,958	102%
ECM. 0.9 inches, 0.38 W/cfm	572	2,220.6	29.9	2,250.6	4,370	793	5,163	106%

Atlanta	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	193	781.1	457.1	1,238.2	1,562	582	2,145	100%
PSC, 0.7 inches, 0.39 W/cfm	180	808.4	502.1	1,310.5	1,601	521	2,122	99%
PSC, 0.9 inches, 0.37 W/cfm	163	861.6	585.4	1,447.0	1,678	468	2,146	100%
ECM, 0.5 inches, 0.30 W/cfm	194	774.8	457.2	1,231.9	1,551	414	1,965	100%
ECM, 0.7 inches, 0.33 W/cfm	183	795.1	486.1	1,281.2	1,581	445	2,026	103%
ECM. 0.9 inches, 0.38 W/cfm	166	840.6	552.1	1,392.7	1,648	491	2,139	109%

Sterling	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	47	664.1	515.0	1,179.1	1,462	614	2,076	100%
PSC, 0.7 inches, 0.39 W/cfm	35	686.5	566.3	1,252.8	1,496	551	2,048	99%
PSC, 0.9 inches, 0.37 W/cfm	21	730.9	660.7	1,391.6	1,566	498	2,064	99%
ECM, 0.5 inches, 0.30 W/cfm	47	658.7	515.2	1,174.0	1,451	437	1,888	100%
ECM, 0.7 inches, 0.33 W/cfm	37	675.7	548.1	1,223.8	1,478	471	1,949	103%
ECM. 0.9 inches, 0.38 W/cfm	25	713.3	622.8	1,336.0	1,539	521	2,059	109%

Houston	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	1,015	1,359.0	202.3	1,561.3	2,958	787	3,745	100%
PSC, 0.7 inches, 0.39 W/cfm	944	1,401.1	222.2	1,623.3	3,020	691	3,711	99%
PSC, 0.9 inches, 0.37 W/cfm	852	1,484.4	259.4	1,743.8	3,145	604	3,749	100%
ECM, 0.5 inches, 0.30 W/cfm	1,019	1,347.1	202.4	1,549.5	2,935	558	3,493	100%
ECM, 0.7 inches, 0.33 W/cfm	974	1,378.8	215.3	1,594.0	2,983	593	3,577	102%
ECM. 0.9 inches, 0.38 W/cfm	893	1,450.7	244.6	1,695.3	3,094	640	3,734	107%

Table C-8. Simulation Results Showing Impact of External Static Pressure on Annual Energy Use (cont.)

Fort Worth	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	131	1,255.7	290.3	1,546.1	2,827	805	3,632	100%
PSC, 0.7 inches, 0.39 W/cfm	115	1,297.2	319.2	1,616.4	2,892	711	3,603	99%
PSC, 0.9 inches, 0.37 W/cfm	99	1,379.7	372.8	1,752.6	3,022	627	3,649	100%
ECM, 0.5 inches, 0.30 W/cfm	132	1,244.6	290.7	1,535.3	2,805	571	3,376	100%
ECM, 0.7 inches, 0.33 W/cfm	122	1,275.3	309.1	1,584.5	2,854	610	3,464	103%
ECM. 0.9 inches, 0.38 W/cfm	103	1,346.1	351.1	1,697.2	2,969	662	3,631	108%

Chicago	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	76	596.0	783.8	1,379.8	1,167	649	1,816	100%
PSC, 0.7 inches, 0.39 W/cfm	74	617.8	861.3	1,479.1	1,198	588	1,786	98%
PSC, 0.9 inches, 0.37 W/cfm	69	659.8	1,003.3	1,663.2	1,258	537	1,796	99%
ECM, 0.5 inches, 0.30 W/cfm	78	591.3	783.6	1,374.9	1,159	462	1,621	100%
ECM, 0.7 inches, 0.33 W/cfm	76	607.1	834.2	1,441.3	1,182	501	1,683	104%
ECM. 0.9 inches, 0.38 W/cfm	72	642.9	946.4	1,589.3	1,234	560	1,794	111%

Phoenix	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	-	1,544.8	117.5	1,662.3	4,659	1,117	5,777	100%
PSC, 0.7 inches, 0.39 W/cfm	-	1,593.0	129.0	1,722.0	4,758	978	5,736	99%
PSC, 0.9 inches, 0.37 W/cfm	-	1,692.6	149.8	1,842.4	4,959	851	5,810	101%
ECM, 0.5 inches, 0.30 W/cfm	-	1,531.6	117.5	1,649.1	4,623	792	5,415	100%
ECM, 0.7 inches, 0.33 W/cfm	-	1,567.4	124.9	1,692.3	4,700	840	5,540	102%
ECM. 0.9 inches, 0.38 W/cfm	-	1,652.2	141.2	1,793.4	4,878	902	5,781	107%

New_York_City	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	29	692.7	589.1	1,281.8	1,349	603	1,952	100%
PSC, 0.7 inches, 0.39 W/cfm	22	717.0	647.9	1,364.9	1,383	543	1,925	99%
PSC, 0.9 inches, 0.37 W/cfm	18	765.2	755.4	1,520.6	1,451	492	1,942	100%
ECM, 0.5 inches, 0.30 W/cfm	29	687.2	589.1	1,276.3	1,339	429	1,768	100%
ECM, 0.7 inches, 0.33 W/cfm	25	705.2	627.5	1,332.7	1,365	463	1,828	103%
ECM. 0.9 inches, 0.38 W/cfm	19	746.2	712.1	1,458.4	1,424	514	1,938	110%

Table C-8. Simulation Results Showing Impact of External Static Pressure on Annual Energy Use (cont.)

San_Francisco	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	-	1,065.3	327.3	1,392.6	2,400	725	3,126	100%
PSC, 0.7 inches, 0.39 W/cfm	-	1,101.3	359.7	1,461.0	2,457	643	3,100	99%
PSC, 0.9 inches, 0.37 W/cfm	-	1,174.7	420.6	1,595.3	2,571	571	3,142	101%
ECM, 0.5 inches, 0.30 W/cfm	-	1,055.2	327.8	1,383.0	2,380	515	2,894	100%
ECM, 0.7 inches, 0.33 W/cfm	-	1,082.7	348.4	1,431.2	2,425	551	2,975	103%
ECM. 0.9 inches, 0.38 W/cfm	-	1,145.2	396.2	1,541.4	2,526	601	3,127	108%

Fresno	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
PSC, 0.5 inches, 0.42 W/cfm	-	939.5	299.6	1,239.0	2,413	729	3,142	100%
PSC, 0.7 inches, 0.39 W/cfm	-	972.1	327.9	1,300.1	2,473	646	3,119	99%
PSC, 0.9 inches, 0.37 W/cfm	-	1,037.8	382.6	1,420.4	2,591	574	3,165	101%
ECM, 0.5 inches, 0.30 W/cfm	-	931.7	299.8	1,231.5	2,395	517	2,912	100%
ECM, 0.7 inches, 0.33 W/cfm	-	955.8	318.1	1,273.9	2,440	553	2,994	103%
ECM. 0.9 inches, 0.38 W/cfm	-	1,011.2	360.6	1,371.8	2,544	604	3,147	108%

Figure C-8. Results for Miami with PSC and ECM Motors at Different Static Pressures

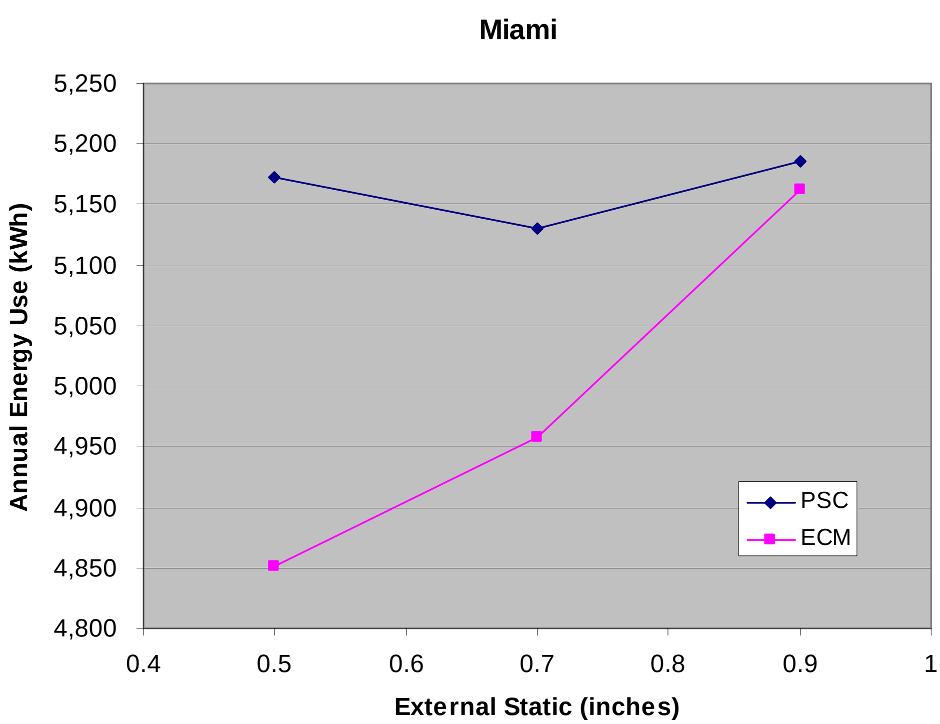
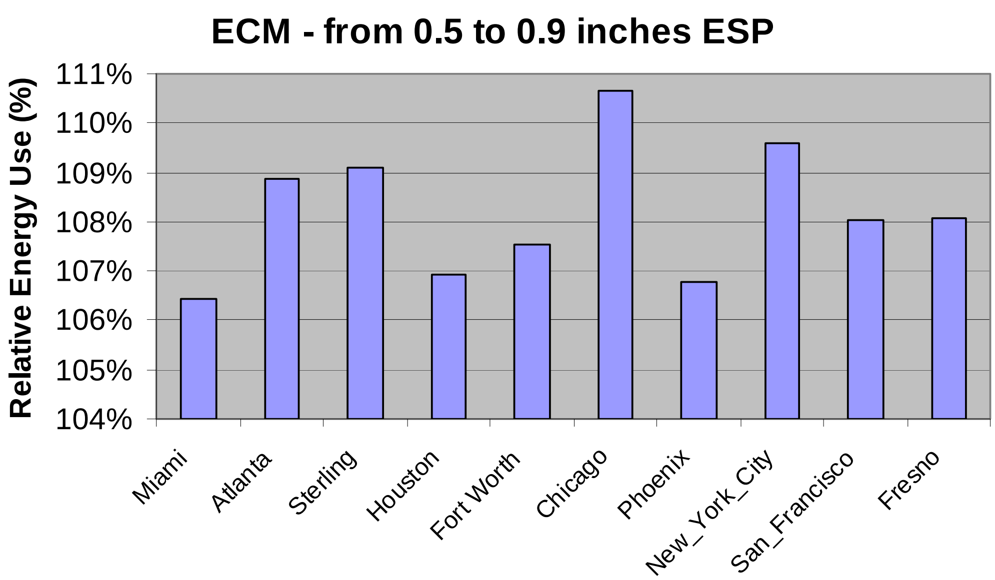


Figure C-9 shows the relative energy use of the ECM motor for the ten different locations. The increase in energy use when the ESP increases from 0.5 to 0.9 inches with the ECM motor ranges from 6 to 10%.

Figure C-9. Relative Energy Use of ECM Motors at 0.9 Compared to 0.5 inches ESP



Reduced AHU Leakage

The baseline AC unit assumes that the return leakage is 4% of the total airflow. Most of this leakage is assumed to occur at the air handler unit. We assume that a robust unit with a tighter AHU cabinet might reduce overall return leakage by 3% of the nominal flow, or by 36 cfm for a 3 ton unit[35].

Table C-9 and Figure C-10 show the simulation results when the return leakage rate is reduced from 4% to 1%. The savings range from 1.5 to 3%.

[35] 4.5% is the average return leakage of several AHUs measured at FSEC (Cummings 2005).

Table C-9. Simulation Results Showing Impact of Reduced AHU Return Leaks on Annual Energy Use

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	722	2,082.5	24.6	2,107.2	4,182	991	5,173	100%
1% Return Leak (Robust AHU)	699	2,020.9	24.5	2,045.4	4,056	962	5,018	97%

Atlanta	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	193	781.1	457.1	1,238.2	1,562	582	2,145	100%
1% Return Leak (Robust AHU)	187	762.2	452.8	1,215.0	1,523	572	2,095	98%

Sterling	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	47	664.1	515.0	1,179.1	1,462	614	2,076	100%
1% Return Leak (Robust AHU)	41	649.0	509.8	1,158.8	1,427	603	2,031	98%

Houston	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	1,015	1,359.0	202.3	1,561.3	2,958	787	3,745	100%
1% Return Leak (Robust AHU)	997	1,317.0	200.7	1,517.7	2,864	765	3,629	97%

Fort Worth	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	131	1,255.7	290.3	1,546.1	2,827	805	3,632	100%
1% Return Leak (Robust AHU)	126	1,219.7	287.9	1,507.6	2,744	785	3,530	97%

Chicago	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	76	596.0	783.8	1,379.8	1,167	649	1,816	100%
1% Return Leak (Robust AHU)	76	584.4	773.6	1,357.9	1,144	639	1,783	98%

Phoenix	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	-	1,544.8	117.5	1,662.3	4,659	1,117	5,777	100%
1% Return Leak (Robust AHU)	-	1,506.0	116.8	1,622.8	4,540	1,091	5,630	97%

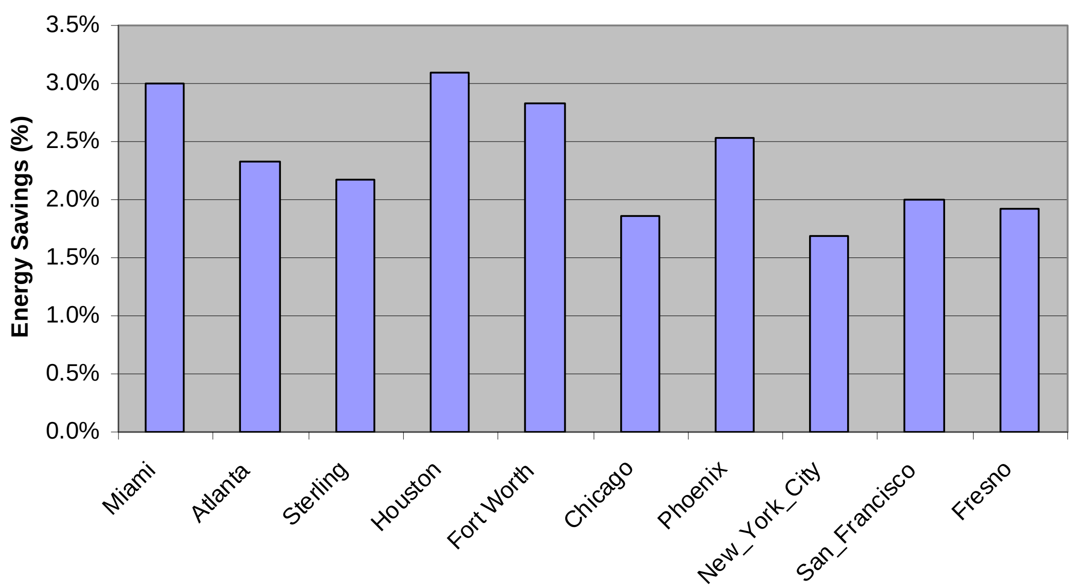
Table C-9. Simulation Results Showing Impact of Reduced AHU Return Leaks on Annual Energy Use (cont.)

New_York_City	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	29	692.7	589.1	1,281.8	1,349	603	1,952	100%
1% Return Leak (Robust AHU)	28	680.6	583.2	1,263.8	1,325	594	1,919	98%

San_Francisco	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	-	1,065.3	327.3	1,392.6	2,400	725	3,126	100%
1% Return Leak (Robust AHU)	-	1,043.4	325.0	1,368.5	2,350	713	3,063	98%

Fresno	Hours above 60% RH (hrs)	AC Runtime (hrs)	Heating Runtime (hrs)	Supply Fan Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
4% Return Leak	-	939.5	299.6	1,239.0	2,413	729	3,142	100%
1% Return Leak (Robust AHU)	-	921.1	297.5	1,218.5	2,365	717	3,081	98%

Figure C-101. Energy Savings from Reduced AHU Return Air Leakage (Decrease from 4% to 1%)



APPENDIX D: CORRELATIONS TO MODEL THE IMPACT OF CHARGE ON EFFICIENCY AND CAPACITY

Proctor, Bresner, and Katsnelson (2005) asserted that the laboratory data collected by Davis and D'Albora (2001a and 2001b) for Pacific Gas Electric are the definitive results for a determining the impact of refrigerant charge on air conditioner performance over a broad range of operating conditions. The two studies looked at both R22 and R410a units and found very similar results for these two refrigerants. We used the measured data from the Appendix of the R22 tests (Davis and D'Albora 2001a) as the basis for the analysis in this section.

Figure D-1 is a duplicate version of Figure 14 in Davis and D'Albora (2001a). The EER is normalized (by the original authors) by dividing by the EER of the system at 80/67/95 with a charge fraction of 1.0 (100% of the nominal charge). Similarly Figure D-2 is a duplicate of the Figure 13 for "Dry" coil conditions (80/63 entering).

Figure D-1. Data from Davis and D'Albora (2001a), Figure 14: Normalized Efficiency at 80/67

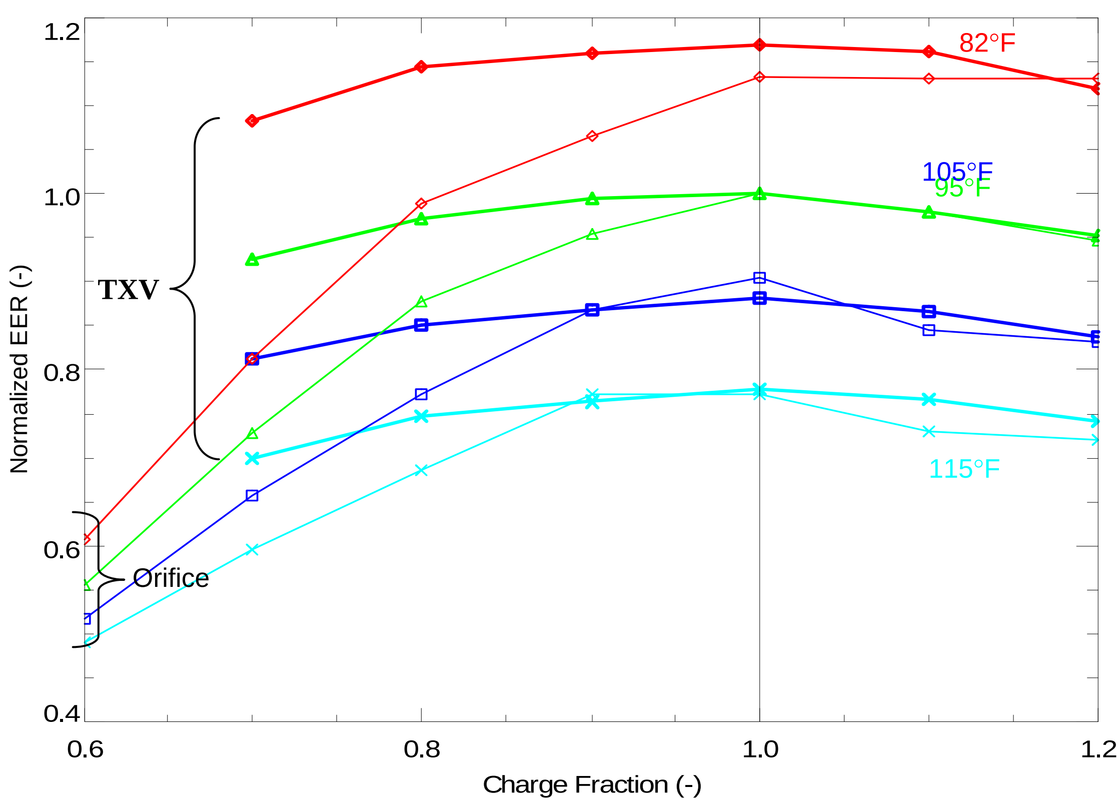
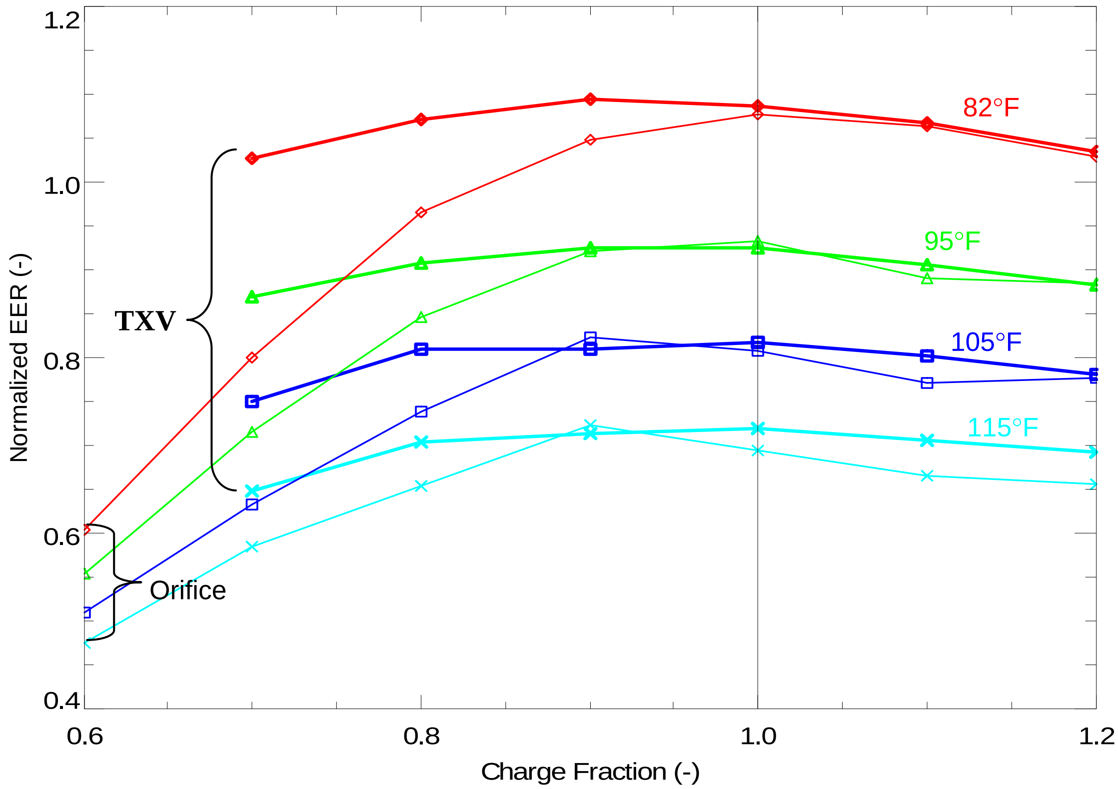


Figure D-2. Data from Davis and D'Albora (2001a), Figure 23: Normalized EER at 80/63 Dry Conditions



If the curves above are normalized to the efficiency corresponding to the nominal charge for each outdoor temperature condition, the curves collapse to a single trend as shown in Figure D-3 for wet conditions and Figure D-4 for dry conditions. The large "+" symbols indicate the average at each charge point. These average points are given in Table D-1 below. Figure D-5 compares the data for wet and dry conditions from Table D-1. The differences between the wet and dry conditions appear to be minor, which implies that the impact of charge is essentially independent of operating conditions.

Table D-1. The Average Value of the Normalized Efficiency Ratio at Each Level of Charge

		Refrigerant Charge						
		60%	**70%**	**80%**	**90%**	**100%**	**110%**	**120%**
Wet	Orifice	0.576	0.737	0.874	0.964	1.000	0.965	0.950
	TXV	-	0.918	0.970	0.989	1.000	0.984	0.954
Dry	Orifice	0.618	0.784	0.915	1.005	1.000	0.964	0.953
	TXV		0.926	0.984	0.997	1.000	0.981	0.956

Figure D-3. Impact of Charge on Normalized Efficiency at Wet Conditions (80/67)

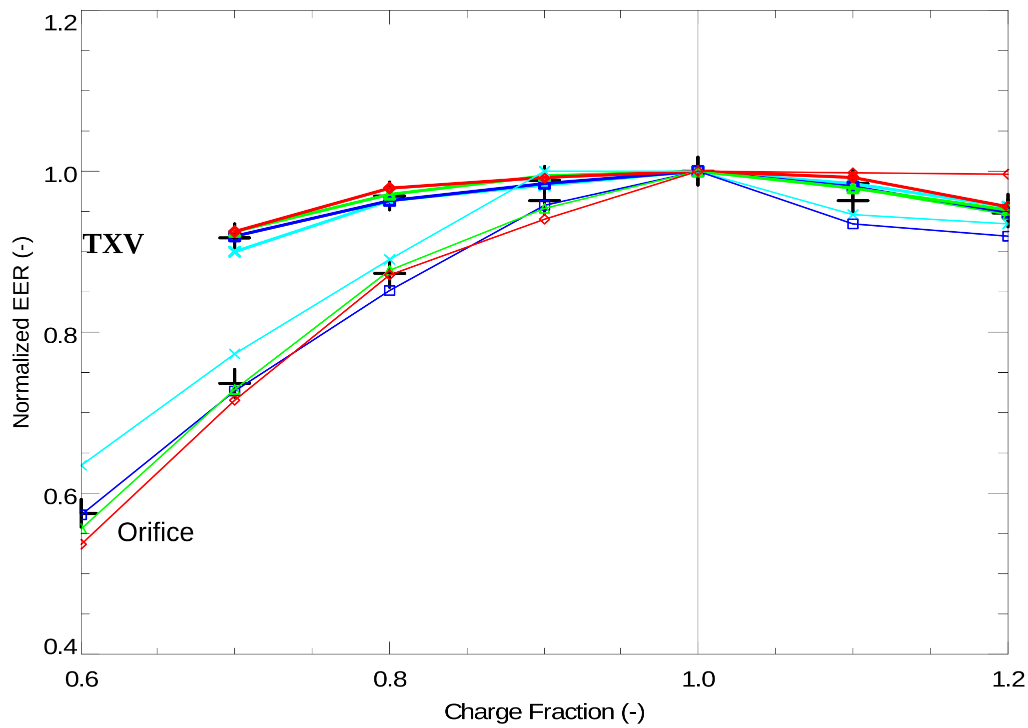


Figure D-4. Impact of Charge on Normalized Efficiency at Dry Conditions (80/63)

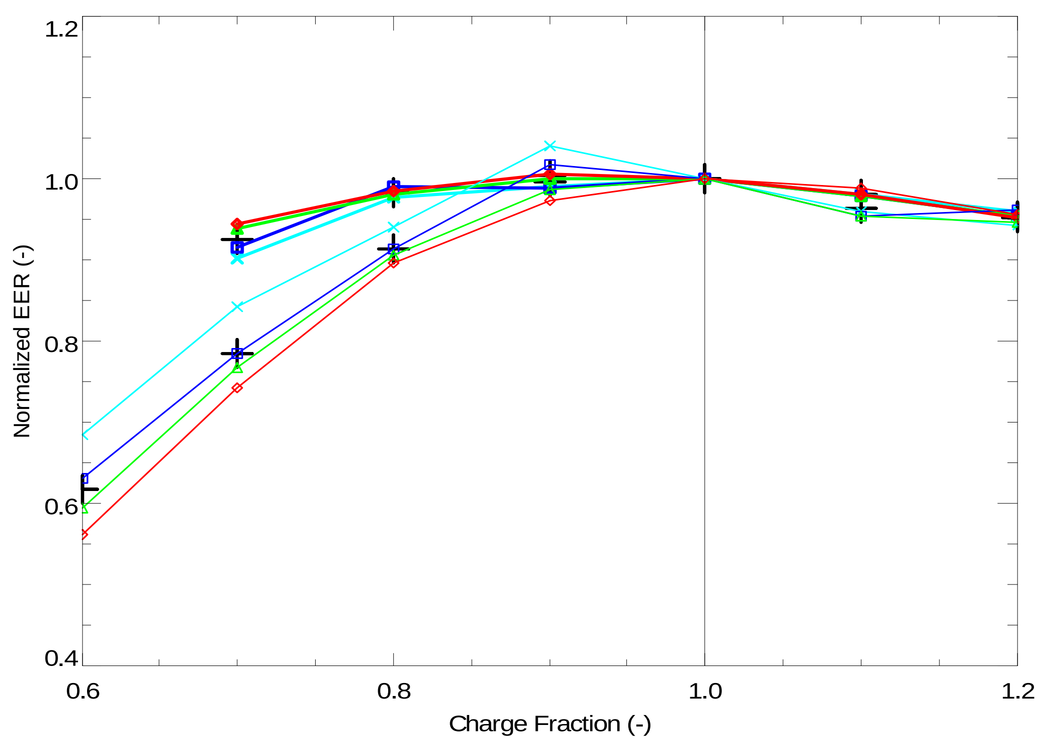
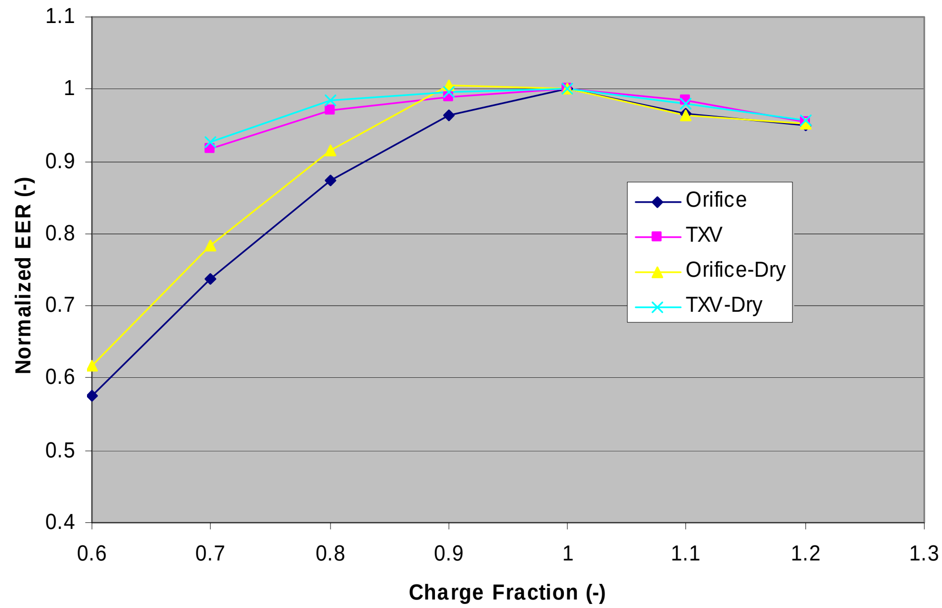


Figure D-52. Comparing Normalized Impact of Charge for Wet and Dry Coil Conditions

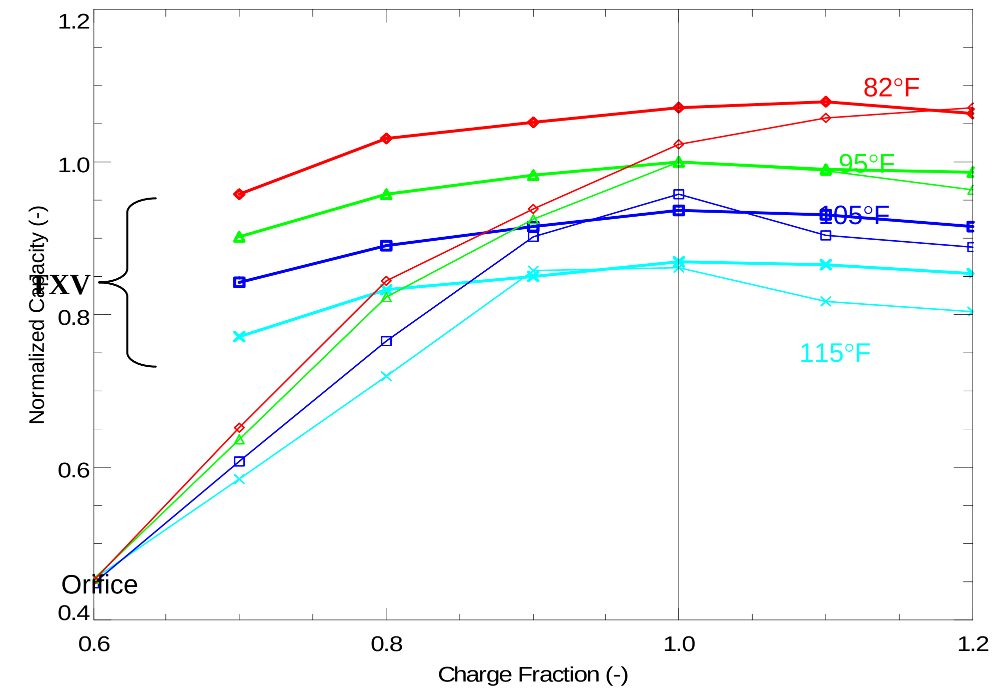


The same process was repeated for the normalized capacity. The data presented in Davis and D'Albora (2001a and 2001b) is shown in Figure D-6 and Figure D-7. Again the curves collapse to a single trend when normalized using the capacity corresponding to 100% of full charge at each temperature.

Table D-2. The Average Value of the Normalized Capacity Ratio at Each Level of Charge

		Refrigerant Charge						
		60%	**70%**	**80%**	**90%**	**100%**	**110%**	**120%**
Wet	Orifice	0.473	0.647	0.821	0.945	1.000	0.979	0.968
	TXV		0.891	0.956	0.980	1.000	0.997	0.985
Dry	Orifice	0.515	0.696	0.865	0.991	1.000	0.970	0.959
	TXV		0.908	0.971	0.987	1.000	0.990	0.984

Figure D-6. Normalized Capacity at 80/67



Source: Data from Davis and D'Albora (2001a), Figure 12

Figure D-7. Data from Davis and D'Albora (2001a), Figure 19: Normalized Capacity, Dry 80/63

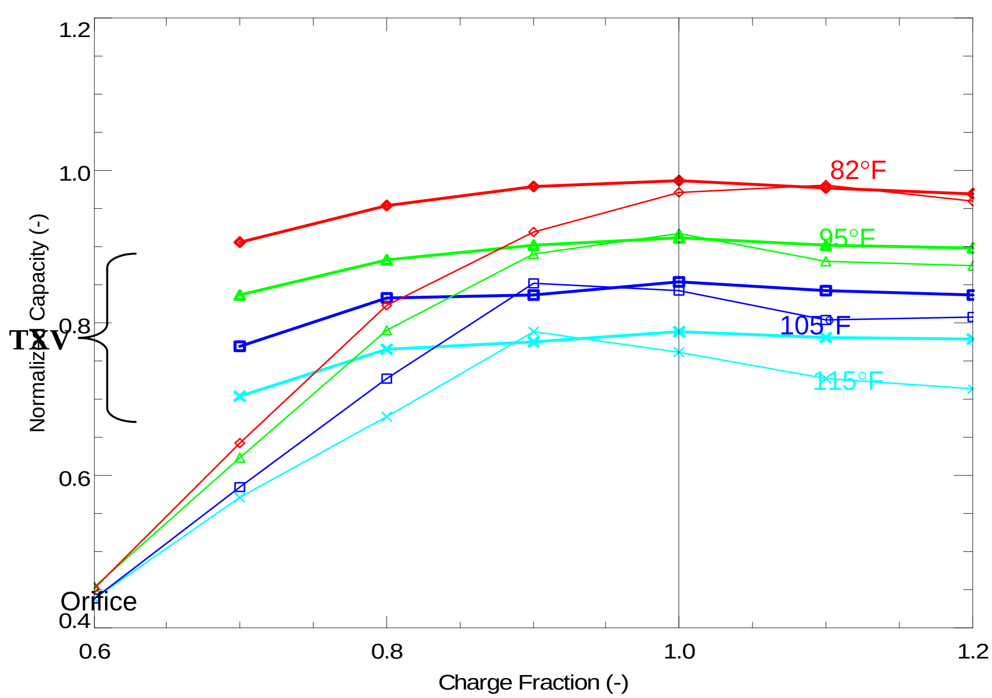


Figure D-83. Impact of Charge on Normalized Capacity at Wet Conditions (80/67)

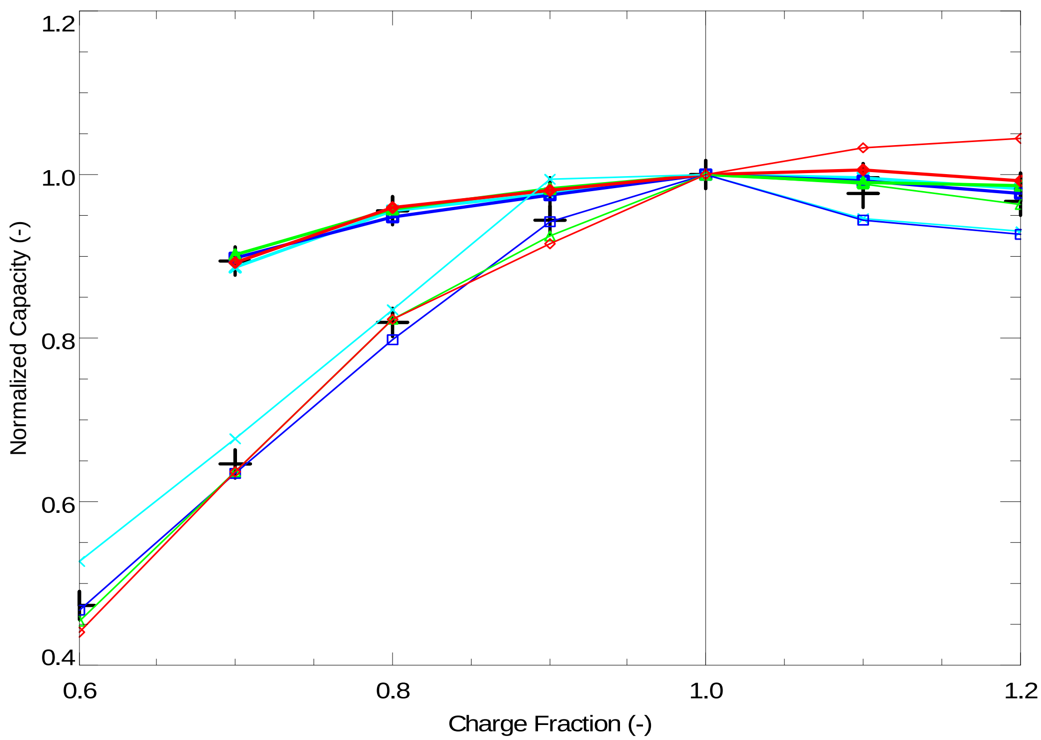


Figure D-9. Impact of Charge on Normalized Capacity at Dry Conditions (80/63)

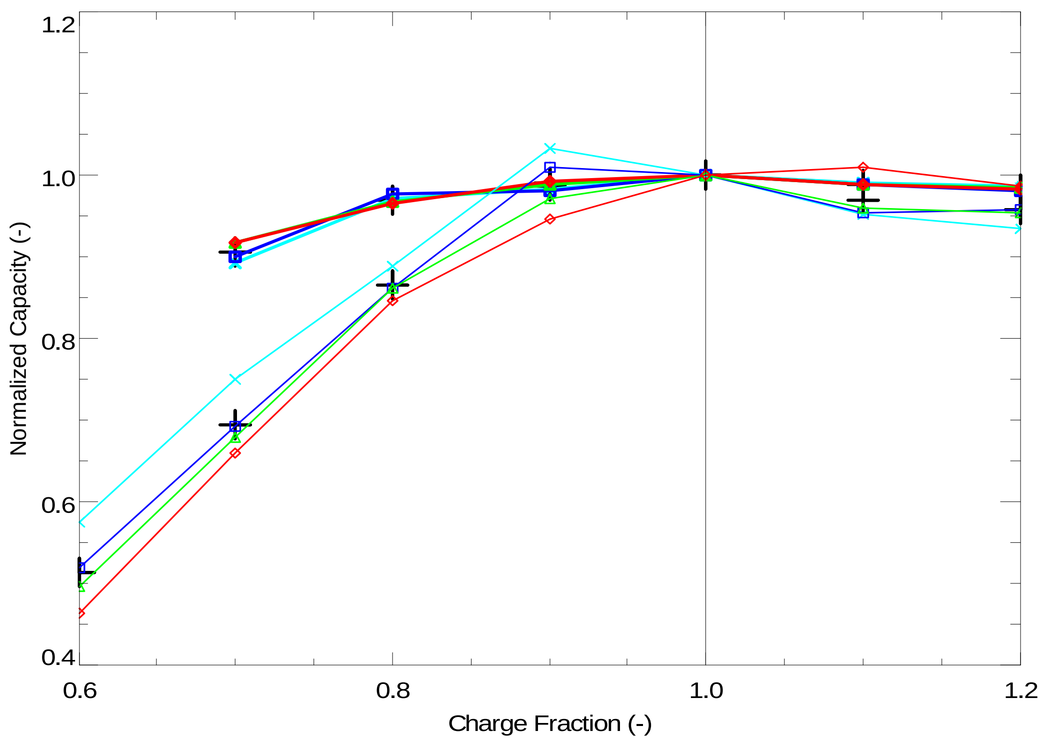
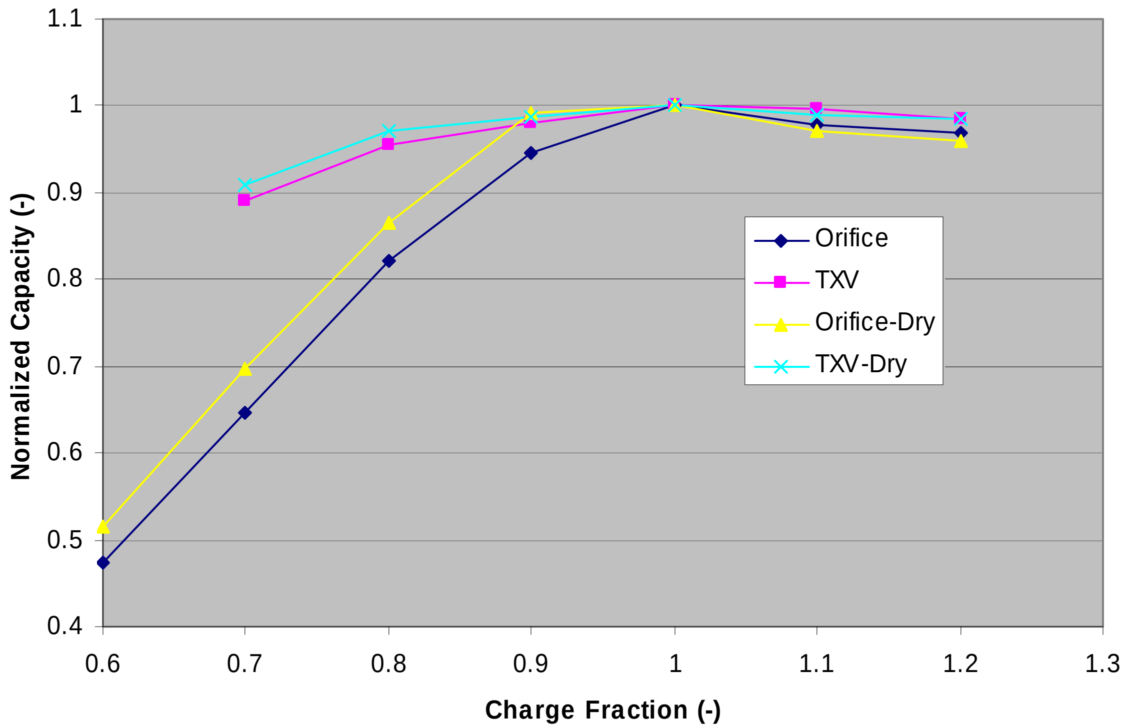


Figure D-10. Comparing Normalized Impact of Charge for Wet and Dry Coil Conditions



The same process was repeated for the Sensible Heat Ratio (SHR) using the wet coil data presented in the Davis and D'Albora (2001a and 2001b).

Figure D-11. Impact of Charge on SHR Ratio at Wet Conditions (80/67)

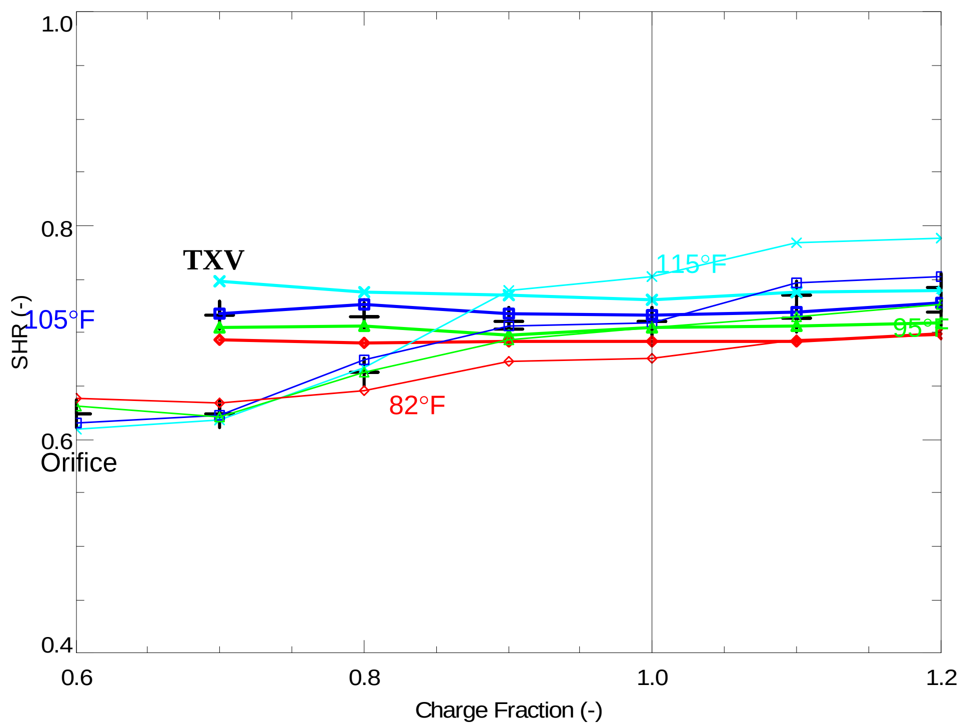
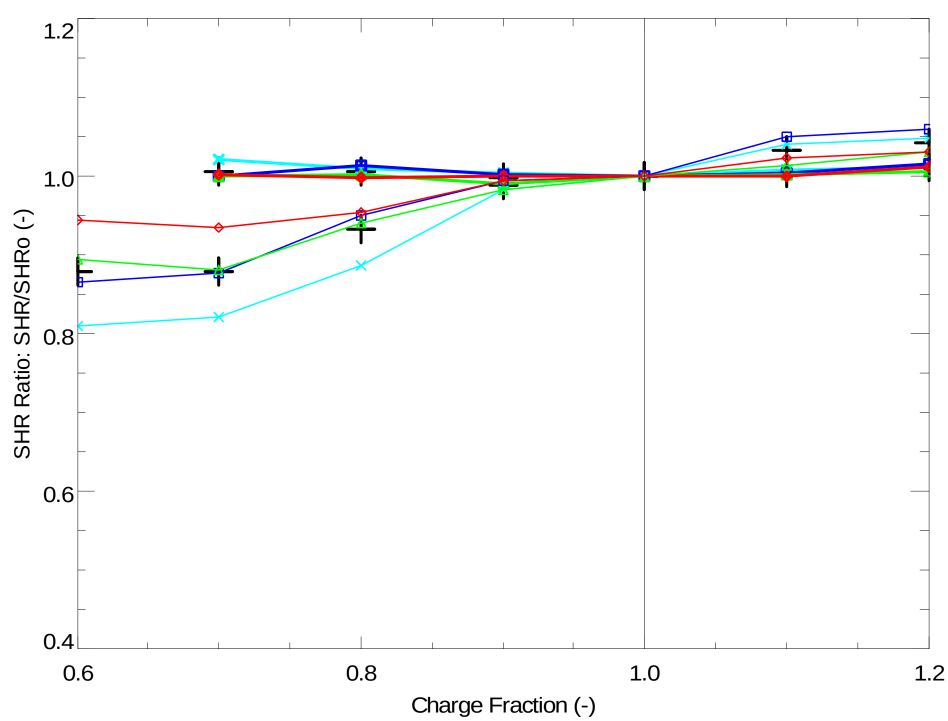


Figure D-12. Impact of Charge on SHR at Wet Conditions (80/67)



The figures below fit the average values for wet coil conditions to a quadratic curve. The simple quadratic curves generally fit the averaged data points very well.

Figure D-13. Comparing Quadratic Curve Fit to Normalized EER Data for Wet Coil Conditions

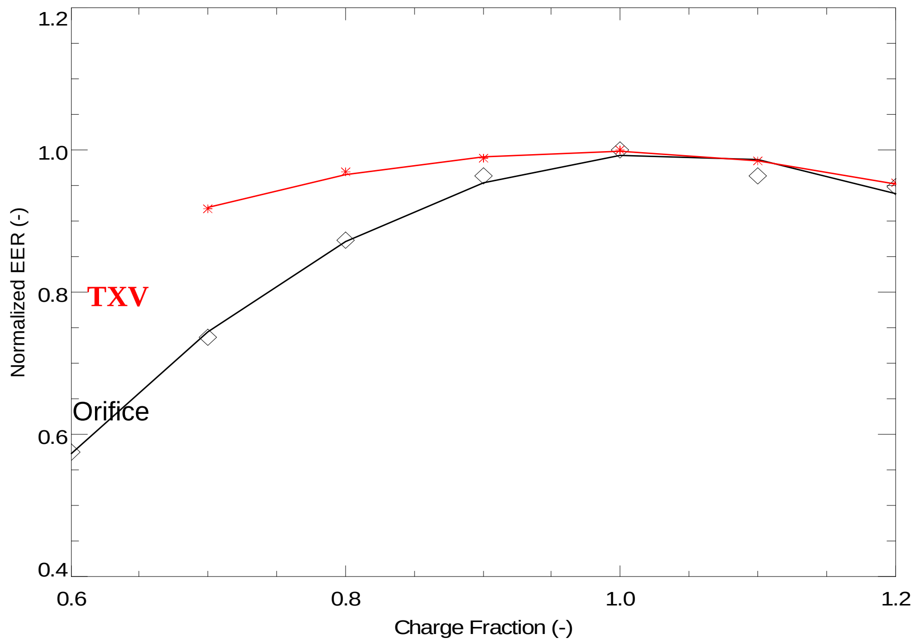
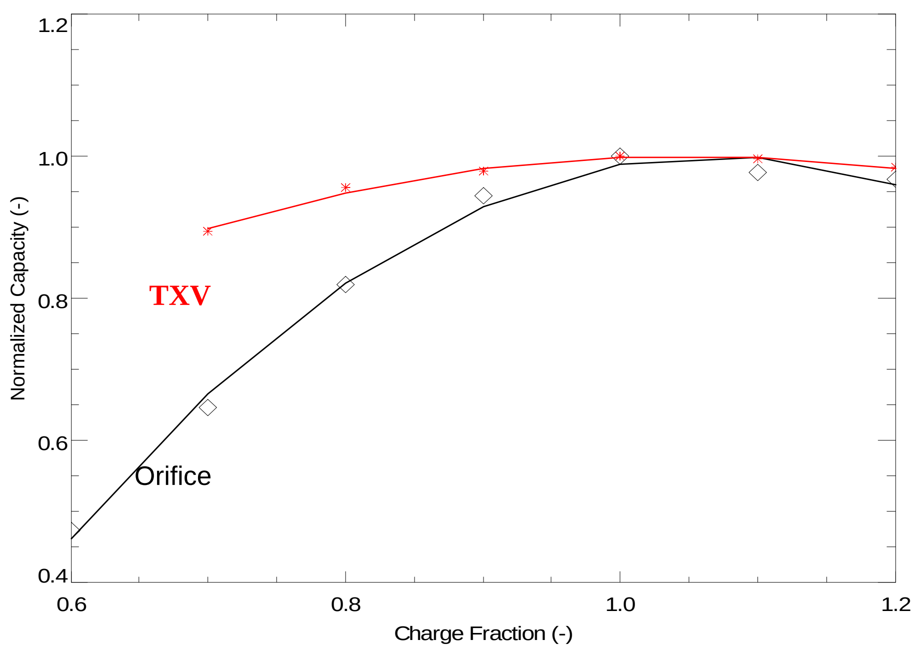


Figure D-14. Comparing Quadratic Curve Fit to Normalized Capacity Data for Wet Coil Conditions



Modeling Approach

The model coefficients corresponding to the curves above are listed in Table D-3 below.

Table D-3. Functions Used to Predict Impact of Refrigerant Charge Level (CR=1.0 at 100%)

Normalized Efficiency

$ER = c0 + c1*CR + c2*CR^2$	Orifice	TXV
Coefficients - c0	-1.3744450	0.051466055
c1	4.5664984	1.9260348
c2	-2.1987032	-0.97870189

Normalized Capacity

$QR = c0 + c1*CR + c2*CR^2$	Orifice	TXV
Coefficients - c0	-1.7810837	0.082344576
c1	5.1948408	1.7502776
c2	-2.4247714	-0.83302159

Normalized SHR

$SR = c0 + c1*CR + c2*CR^2$	Orifice	TXV
Coefficients - c0	0.55515179	1.0
c1	0.61807582	0
c2	-0.17174450	0

To model the impact of charge we "superimposed" the relationships developed here with the standard AC performance map. The standard performance map determines the capacity, efficiency and sensible heat ratio as a function of operating conditions: entering dry bulb (Tin), entering wet bulb (Wbin), ambient temperature (Tamb), airflow ratio (fratio) and expansion device type (dtype). The overall model will be of the following form:

Qt = F1(Tin, Wbin, Tamb, fratio, dtype) x F2(CR, dtype)

Where:
F1 () – standard performance map for AC capacity ()
F2 () – correction factor for capacity as function of charge ratio and type of expansion device

Similar performance maps and correction factors were used for the EER and sensible heat ratio.