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Energy Performance of Hot, Dry Optimized Air Conditioning Systems

Prepared for:
California Energy Commission
Public Interest Energy Research Program

Final Report
July 2008

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ENERGY PERFORMANCE OF HOT, DRY OPTIMIZED AIR-CONDITIONING SYSTEMS

PIER FINAL PROJECT REPORT

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Preface

The California Energy Commission's Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program conducts public interest research, development, and demonstration (RD&D) projects to benefit California.

The PIER Program strives to conduct the most promising public interest energy research by partnering with RD&D entities, including individuals, businesses, utilities, and public or private research institutions.

- PIER funding efforts are focused on the following RD&D program areas:
- Buildings End-Use Energy Efficiency
- Energy Innovations Small Grants
- Energy-Related Environmental Research
- Energy Systems Integration
- Environmentally Preferred Advanced Generation
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Transportation

Energy Performance of Hot Dry Air Conditioners is the final report for the Hot Dry Air Conditioner project (contract number 500-03-002) conducted by Southern California Edison, Proctor Engineering Group, Ltd, Bevilacqua-Knight, Inc. and New Buildings Institute. The information from this project contributes to PIER's Buildings End-Use Energy Efficiency Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/pier or contact the Energy Commission at 916-654-5164.

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Abstract

This project evaluated the energy savings and market potential for hot, dry air conditioning systems for the residential and commercial market sectors. Commercially available air conditioners are designed to meet national performance standards that are roughly based on "average" cooling season weather conditions across the United States. The current design process gives little or no attention to the performance of the air conditioners at the conditions prevalent in California. As a result, substantial energy is wasted by air conditioners in hot dry climates, particularly on peak days.

The PIER program commissioned a project to promote air conditioners specifically selected to perform well at hot, dry conditions. Pacific Gas and Electric Company, Southern California Edison, and Nevada Power commissioned this side-by-side field test of standard SEER 13 air conditioners and major manufacturers' hot, dry air conditioning systems. The study consisted of field monitoring seven SEER 13 air conditioners and their hot, dry air conditioning system replacements during the summer of 2006.

The results from laboratory and field testing demonstrated energy savings of up to 20 percent and peak demand reductions of up to 35 percent. Based on the technical results and an evaluation of the potential market, the team recommends that the state and the local utilities fast track incentives and marketing for units that meet the hot, dry air conditioning system specifications.

Keywords: HVAC, hot dry air conditioner, electric peak reduction, peak energy efficiency ratio, sensible cooling, energy efficiency, rooftop package unit, split air conditioner, air movement efficiency, dry climate, microchannel coil, condenser fan, condenser airflow, furnace airflow efficiency, SEER, EER, PEERs

Executive Summary

Introduction

California is a summer peaking utility region, and air conditioning is the primary cause of the peaks. Residential air conditioning has a ratio of peak load to average load of 3.5 to 1. This is the highest ratio of all end uses. A residential air conditioner produces a peak watt draw 23 times as great as residential lighting with the same annual consumption.

Air conditioning drives the peak energy consumption that results in the highest marginal cost of electricity. California's electric peak demand is almost completely caused by summertime air conditioning loads that show sharp peaks. California Energy Commission. 2002. *2002-2012 Electricity Outlook Report* P700-01-004F.

California's peak electric demand dominates the need for additional power plants, transmission infrastructure, and related environmental issues. Even high-performance air-conditioning systems are not optimized to maximize indoor temperature reduction for each watt-hour of consumption under hot and dry ambient conditions. Reducing peak-electric demand by 20 percent in residential and small commercial air conditioners could save California as much as 71 megawatts per year at a 20 percent market penetration.

Commercially available air conditioners are designed to meet national performance standards that are roughly based on "average" cooling season weather conditions across the United States. For residential air conditioners, the performance metric is the Seasonal Energy Efficiency Ratio (SEER). SEER is based on indoor conditions that require significant dehumidification and an outdoor temperature of only 82ºF. For commercial air conditioners larger than 5 tons, the metric is the Energy Efficiency Ratio (EER) rated at 95ºF, closer to the performance needed in California. The current design process gives little or no attention to the performance of the air conditioners at higher temperatures or where dehumidification is not necessary. The only mandatory test for high temperature is a maximum operating conditions test at 115ºF. The manufacturers do not certify or report the performance of their air conditioners at that temperature.

Purpose

The primary goals of this project were to:

- Demonstrate that peak-electric demand could be reduced by 15 percent to 25 percent due to the installation of new residential and small commercial air conditioners in hot and dry climate regions of California.
- Demonstrate that energy use could be reduced by 10 percent to 25 percent from these new residential and small commercial air conditioners.

The steps initially defined to achieve these goals were to:

- Design, fabricate, and test cost-effective hot, dry air conditioning systems that are optimized for hot, dry climates.

- Develop new engineering design methods for hot/dry climates.
- Evaluate the impact of various control strategies for residential and small commercial hot, dry air conditioning systems on peak electrical demand and electrical energy use.

Key Accomplishments

The following is a summary of the key objectives accomplished during this project:

- Built one 3-ton split air conditioner with reduced peak watt draw and improved life-cycle costs within current manufacturing capabilities.
- Built one 5-ton package rooftop air conditioner with reduced peak watt draw and improved life-cycle costs within current manufacturing capabilities.
- Created "hot/dry low-peak" specification for reduced peak improved life-cycle cost hot, dry air conditioning systems.
- Developed a partnership with manufacturers to build and test air conditioners that meet the "hot/dry low-peak" standards, while meeting human comfort requirements.
- Coordinated and worked with the U.S. Department of Energy, Oak Ridge National Laboratory, and Purdue University in low-peak air conditioner design and control.
- Conducted outreach efforts to publicize the results of the research.

The hot, dry air conditioning system units used for the field testing were provided by Carrier Corporation, Lennox International, and American Standard Companies. Early in the Project a hot, dry air conditioning system performance standard was developed in anticipation that manufacturers would be willing to manufacturer their own hot, dry air conditioning system units. Instead manufacturers chose to provide off-the-shelf equipment that they believed provided equivalent hot, dry air conditioning system performance.

Due to the unforeseen amount of time and effort required to complete the initial phases of the project, the contract term expired before the completion of some of the final tasks. Despite these limitations, this report does attempt to address all the goals and objectives established for the project and provides a full set of recommendations on which to base future efforts to promote and commercialize hot, dry air conditioning systems technology.

Project Outcomes

Detailed test data from a variety of sources was reviewed to determine the appropriate configuration of baseline air conditioners which served two purposes. This information provided data to populate a simulation model for comparing various potential design changes to achieve high efficiency at high temperatures and low humidity. It also provided the initial base of comparison for the performance and economics of the resulting air conditioners.

The anticipated minimum SEER rating in the United States at the start of the hot, dry air conditioning system project was SEER 12. During this project the anticipated minimum SEER rating was increased to 13. In the final analysis the hot, dry air conditioning systems were evaluated against typical SEER 13 R-410A air conditioners.

Two baseline air conditioners were selected. One air conditioner was a 3- ton typical split system residential unit and the other was a 5-ton typical light commercial package unit.

Test data and the physical parameters were used to create a computer model of the baseline units. The Oak Ridge National Lab Heat Pump Design Model was used for modeling the baseline units (ORNL 2002). The model was calibrated to baseline test data when available. These models were the starting points for the design process.

The team researched a broad range of potential component designs for use in the hot, dry air conditioning system units. The potential candidates were evaluated for cost, performance, availability, maintenance issues, durability and reliability. In additional to many standard energy efficiency features the modeling examined specific hot, dry air conditioning system features such as:

- Higher saturation temperature evaporator coil.
- Higher airflow across the evaporator coil.
- Controls to minimize latent capacity under dry indoor conditions.
- Increased condensate retention on the evaporator coil.
- Controls to obtain latent capacity when indoor moisture rises significantly.
- Controls to evaporate moisture off the coil rather than allow condensate drainage.

Extensive airflow optimization was performed at the research group's test facility to optimize hot, dry air conditioning system performance and the results incorporated into the simulation model.

A life cycle cost model of energy consumption was created based on the Residential Alternative Calculation Method (ACM) Approval Manual (2005) and the Nonresidential ACM (2005). The results predicted the life cycle costs for residential and commercial buildings using hot, dry air conditioning systems in seven different California climate zones (Table 1).

Extensive evaluation of different component assemblied and system configureations was made to determine the designed that produced the best hot, dry air conditioning system performance.

Based on the results of the physical testing and simulation modeling, prototype units were constructed for extensive laboratory testing. The 3-ton residential split system was tested at PG&E’s Technical and Ecological Services (TES) facility, and the 5-ton commercial packaged system was tested at SCE’s Refrigeration and Thermal Test Center. Extensive evaluation of different component assemblies and system configurations was made to determine the designs that produced the best hot, dry air conditioning system performance.

Table 1. Average Life Cycle Cost Benefits

	Energy Savings (kWh/year)	Peak Reduction (kW)	Life Cycle Cost Savings over to end user 18 years
Residential			
SEER 12 Baseline	656	1.53	$905 - $1,457
SEER 13 Baseline	231	0.57	$319 - $509
SEER 14 Baseline	85	0.43	$118 - $213
Commerical			
SEER 12 Baseline	1741	1.95	$2,507 - $3,391
SEER 13 Baseline	1619	1.46	$2,331 - $3,037
SEER 14 Baseline	1157	0.51	$1,666 - $2,051

Source: Proctor Engineering Group

Once the laboratory tests were completed, a hot, dry air conditioning systems performance standard was drafted based on the test results. The resulting hot, dry air conditioning systems standard was less than the performance demonstrated by the prototype units, but it was chosen as within reasonable reach by the major manufacturers (Table 2).

Table 2: Hot, Dry Air Conditioner Draft Specifications

Condition #1	Hot Dry 115/80/63
Gross Sensible Capacity (sensible btuh)	75% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 8 btu/watthr
Condition #2	Hot Medium 115/80/67
Gross Sensible Capacity (sensible btuh)	65% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 6.8 btu/watthr

Source: Proctor Engineering Group

A significant outreach effort was executed in parallel with the technical development of the hot, dry air conditioning system units. To increase the opportunities for field testing hot, dry air conditioning systems an extensive outreach effort was undertaken to recruit other utilities in hot/dry states. As a result of these efforts, Nevada Power agreed to participate in addition to SCE and PG&E. Efforts to recruit manufacturers results in participation from Goettl Air Conditioning, Lennox, and Trane chose to formally participate. Numerous other outreach efforts were also made to encourage support and interest from a wide variety of industry and research organizations including the American Refrigeration Institute, Air Conditioning

Contractors of American and the American Society of Heating, Refrigeration and Air-Conditioning Engineers.

The sites selected for field testing included three residential buildings in PG&E territory, one residential and one commercial building in SCE territory, and one residential building in Nevada Power territory. Field testing consisted of an evaluation of an existing baseline unit (Table 3).

Table 3: Sites Selected for Field Testing

Site	Bakers-field	Concord	Madera	Yuba	Furnace Creek	Victor-ville	Las Vegas
House Size (square feet)	1200	1400	1650	1600	1230	1600	1225
Year Built	1941	1970s	2002	1991	NA	2004	NA
Air Handler Location	Bedroom Closet	Hall Closet	Attic	Attic	Package Rooftop	Attic	Attic

Source: Proctor Engineering Group

The goal of a 10 to 25 percent reduction in annual energy use was met in all but two locations (Table 4).

Table 4: Standard vs HDAC Performance Summary

Location	Standard Unit Annual Energy Usage (kWh)	HDAC Unit Annual Energy Usage (kWh)	Energy Savings (kWh)	**Annual Energy Savings (%)**
Las Vegas	2770	2291	478	**17%**
Furnace Creek	11086	9232	1854	**17%**
Victorville	3534	2498	1036	**29%**
Madera	1966	1618	348	**18%**
Yuba	1592	1256	336	**21%**
Bakersfield	3059	3262	-203	**-7%**
Concord	420	443	-23	**-5%**

Source: Proctor Engineering Group

The goal of a 15 percent to 25 percent reduction in the coincident peak demand was met in all but two locations (Table 5). Location of AC equipment, and baseline was very efficient R-22 equipment as well, may be why the comparison not so good.

Table 5: Standard vs HDAC Average Coincident Peak Demand Summary

	Standard Unit Average Coincident Peak Demand (W)	HDAC Unit Average Coincident Peak Demand (W)	Average Coincident Peak Demand Reduction (W)	**Average Coincident Peak Demand Reduction (%)**
Las Vegas	1630	1292	339	**21%**
Furnace Creek	4125	3632	493	**12%**
Victorville	2633	1903	729	**28%**
Madera	1080	902	177	**16%**
Yuba	1041	847	194	**19%**
Bakersfield	1888	2073	-186	**-10%**
Concord	297	312	-16	**-5%**

Source: Proctor Engineering Group

The units in Bakersfield and Concord were intensively monitored, and it was determined that they performed well below the manufacturers' published data.

Conclusions

The Hot, Dry Air Conditioning Systems Project proved the viability of air conditioners designed or selected to operate more efficiently in hot dry climates. The field test results clearly demonstrated the following energy and peak demand benefits:

1. Existing single speed air conditioners utilizing outdoor units, indoor coils, and furnaces selected to meet the performance standards set in the hot, dry air conditioning systems project can produce annual cooling energy savings of 20 percent or more.
2. These hot, dry air conditioning systems machines showed up to a 35 percent peak reduction at system critical peak times.

Air conditioners manufactured to perform equal to or better than the two proof-of-concept hot, dry air conditioning systems units would be cost-effective based on life-cycle costs as long as they are not marked up as premium machines.

The Hot, Dry Air Conditioning Systems Project was successful in producing two units that provided the target energy savings and peak reductions.

The project's laboratory tests proved that air conditioners (the combination of the furnace, outside unit, and indoor coil), when designed with hot, dry climates in consideration, can exceed the following hot, dry air conditioning systems specification.

The Hot, Dry Air Conditioning Systems Project illustrated that there are highly significant additional gains available for dry climates using an extended evaporator fan run time at the end of the compressor cycle.

The major manufacturers are not inclined to produce moderate cost air conditioners that meet the hot, dry air conditioning Systems specifications unless:

- There is a market for the units.
- The market is supported by utility incentives and state- or regionwide marketing.

Recommendations

Given the energy and peak demand benefits of hot, dry air conditioning system units clearly demonstrated as a result of this project, efforts to encourage the adoption of this technology should be made in California and throughout hot dry climate states. The actions recommended in this report are as follows:

- **A California Public Utilities Commission decision to focus attention on climate optimized standards at the regional and national level.** This public declaration of interest will signal to regulators in the Southwest that this is an issue with regionwide interest that will require collaboration across the region.
- **The inclusion of climate optimized efficiency in the development process for the 2011 Title 24 Standards revision process.** There was strong support of the need to optimize cooling equipment and designs for a variety of climates in California. Hot, dry optimization was the dominant theme. This is the first step for the consideration of climate optimized efficiency standards in Title 24.
- **Legislation to establish climate optimized efficiency standards for two climates, hot and dry and hot and humid.** The very fact that California was investigating this potential, both in the laboratory and in field tests, provided sufficient value to substantiate the benefits of proposing the development of federal regional standards.
- **Creation of a National Cooling Initiative to address multiple issues with compressor-based cooling equipment across all market segments.** This initiative would focus on leveraging the work done under this project along with other PIER California Energy Commission work in hot, dry air conditioning systems including the advanced rooftop unit, fault detection and diagnostics and the PIER-sponsored national Hot, Dry Air Conditioning System Roundtable in 2006.

In addition to follow up on the key outcomes noted above, the following key market connection related steps are still needed for full realization of the hot, dry air conditioning system concept's potential:

- Collaboration with the Western Cooling Efficiency Center (WCEC) at UC Davis to move hot, dry air conditioning systems into the marketplace.
- A system for qualifying complete AC units (including evaporator coil, air handler/furnace, and outside unit) should be developed, preferably requiring manufacturers to test and submit qualifying data for their products and using an impartial laboratory to review and spot-test submittals.

- Manufacturers should be strongly encouraged by utilities, regulators, and advocacy groups to provide guidance to their distributors and dealers on proper hot, dry air conditioning system unit specifications. An industry group could be established to validate which combinations met the hot, dry air conditioning system specification through independent spot-checking.
- The Emerging Technologies staff groups in the major California utilities, allied with PIER, should consider joint support of the further development of hot, dry air conditioning system innovations and their integration into practical units for manufacturing.
- Develop alliances with government and industry groups to induce the air-conditioning industry's adoption of hot, dry air conditioning system-related innovations into the next generation of products for home and small commercial uses.
- Approach the Consortium for Energy Efficiency about opportunities to bring this issue to its membership as an initiative.

Benefits to California

This project made it possible to specify the performance of air conditioners for hot, dry climates with a high degree of confidence and that these specifications are attainable with reasonable manufacturing costs. This puts California in the driver's seat for improving high temperature performance and provides an opportunity to reduce peak electrical consumption growth caused by air conditioners in general and R-410A units in particular.

The market potential for new and replacement HDAC units is highly significant, especially when all states within the hot dry region are included. The estimated annual sales figures for residential and commercial units 5 tons and smaller is 930,000 units per year.

- The total 2007 hot-dry region sales potential for residential central air conditioners and heat pumps is approximately 830,000 units. Of that total, 3-ton units account for an estimated 180,000 units or about 22 percent.
- The total 2007 hot-dry region sales potential for commercial central air conditioners and heat pumps is approximately 100,000 units. Of that total, 3-ton (5-ton?) units account for an estimated 23,000 units or about percent.

Reducing peak-electric demand by 20 percent in residential and small commercial air conditioners could save California as much as 71 megawatts per year at a 20 percent market penetration.

1.0 Introduction

1.1. Background and Overview

California is a summer peaking utility region and air conditioning is the primary cause of the peaks. Residential air conditioning has a ratio of peak load to average load of 3.5 to 1. This is the highest ratio of all electric end uses. A residential air conditioner produces a peak watt draw 23 times as great as residential lighting with the same annual consumption.

Air conditioning is the driver of the peak energy consumption that results in the highest marginal cost of electricity. California's electric peak demand is almost completely caused by summer-time air conditioning loads that show sharp peaks. California Energy Commission. 2002. *2002-2012 Electricity Outlook Report* P700-01-004F.

California's peak electric demand dominates the need for additional power plants, transmission infrastructure and related environmental issues. Even high-performance air conditioning systems are not optimized to maximize indoor temperature reduction for each watt-hour of consumption under hot and dry ambient conditions. Reducing peak-electric demand by 20% in residential and small commercial air conditioners could save California as much as 71 megawatts per year at a 20% market penetration.

Commercially available air conditioners are designed to meet national performance standards that are roughly based on "average" cooling season weather conditions across the United States. For residential air conditioners, the performance metric is the Seasonal Energy Efficiency Ratio (SEER). SEER is based on indoor conditions that require significant dehumidification and an outdoor temperature of only 82ºF. Some discussion of the fan energy problems with SEER deserve to be discussed here…especially since a significant amount of the HDAC benefit came from reducing fan energy. For commercial air conditioners larger than 5 tons, the metric is Energy Efficiency Ratio (EER) rated at 95ºF, closer to the performance needed in California. The current design process gives little or no attention to the performance of the air conditioners at higher temperatures or where dehumidification is not necessary. The only mandatory test for high temperature is a Maximum Operating Conditions test at 115ºF. The manufacturers do not certify or report the performance of their air conditioners at that temperature.

The SEER mad EER tests also substantially underestimate the watt draws of the fans used to circulate air through the evaporator coils, furnaces, and duct systems. Fan watt draw becomes more significant as the reminder of the system is improved. As a result air flow improvements and fan watt draw reductions provide fertile ground for energy savings and peak reductions.

1.2. Goals

The primary goals of this project were to:

- Reduce peak-electric demand by 15% to 25% due to the installation of new residential and small commercial air conditioners in hot and dry climate regions of California.

- Reduce energy use by 10% to 25 % from these new residential and small commercial air conditioners.

The purpose of this project was to produce air conditioners that effectively reduce the temperature of the air with a low power draw. For hot dry climates, the appropriate metric of performance is the Peak Energy Efficiency Ratio sensible (PEERs) measured at high outdoor temperatures with low to moderate indoor humidity, and a typical duct system.

1.3. Project Objectives

The objectives of this project were to:

- Enlist and build support in the industry for manufacturing hot/dry climate air conditioners. This involved over 20 component manufacturers and four AC manufacturers.
- Create and prove hot/dry climate low-peak, life-cycle cost optimized air conditioner design methodology. This design methodology covers split and rooftop package units.
- Build one 3-ton split air conditioner with reduced peak watt draw and improved life-cycle costs within current manufacturing capabilities. Compared to the baseline, this unit:
- Provided a demand reduction of at least 15% at high ambient temperatures.
- Reduced electric operating costs by at least 10%.
- Had a reduced life-cycle cost.
- Had equivalent maintenance costs.
- Had a first cost increase of less than 25%.

The 3-ton unit also maintained comfort conditions under occasional indoor conditions different from design.

- Build one 5-ton package rooftop air conditioner with reduced peak watt draw and improved life-cycle costs within current manufacturing capabilities. Compared to the baseline, this unit:
- Provided at least a 15% reduction in energy consumption at peak conditions.
- Shared all the performance objectives of the 3-ton unit.
- Develop a partnership with a manufacturer and commit them to build and test air conditioners that meet the “hot/dry low-peak” standards, while meeting human comfort requirements.
- Coordinate with, utilize and integrate efforts of DOE, Oak Ridge National Laboratory and Purdue University in low-peak air conditioner design and control.

- Conduct outreach efforts to publicize the results of the research including ASHRAE and other publications.
- Build partnerships with utilities, manufacturers, and DOE to further the commercialization of the technology.

Due to the unforeseen amount of time and effort required to complete the initial phases of the project the contract term expired prior to the completion of some of the final tasks. The tasks which were not fully completed included Market Connection, Technology Transfer Activities and the Production Readiness Plan. While a great deal of progress was made on these tasks, the final deliverables could not be completed within the contract period. Efforts were made to amend the contract and extend the completion date however these could not be formalized prior to the expiration of the contract on September 1, 2007. Despite these limitations, this report does attempt to address all of the initial goals and objectives established for the project, and provides a full set of recommendations on which to base future efforts to promote and commercialize HDAC technology.

2.0 HDAC Development

This project included:

1. Determine the critical operating conditions at the time of peak electrical consumption.
2. Determine the efficiency of air conditioners expected to be the baseline after 2005 at conditions important to peak electrical use in hot dry climates.
3. Produce models of the baseline units that closely approximate their performance in the laboratory, with particular attention to hot, dry conditions.
4. Investigate the widest range of components and assemblies potentially capable of producing higher cooling efficiencies in cost-effective conventional refrigerant-based air conditioning.
5. Modify the baseline models using selected components to determine cost-effective combinations for potential incorporation in two proof-of-concept hot, dry air conditioners.
6. Test components and assemblies to determine the final configuration of the two proof-of-concept hot, dry air conditioners.
7. Assemble, develop and test the two hot, dry air conditioners.
8. Test the two hot, dry air conditioners in laboratories capable of measurements that meet the standards of the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE).
9. Optimize the performance of the hot, dry air conditioners by adjusting available parameters such as superheat, evaporator airflow, and subcooling.
10. Produce a performance standard for hot, dry air conditioners based on the laboratory test results.
11. Compare the results of the laboratory tests to available production air conditioners.
12. Work with manufacturers to select existing component assemblies or modified component assemblies that would meet or approach the performance standard for hot dry air conditioners.

2.1. Peak Energy Efficiency Ratio

Air conditioners produce two effects. They lower the temperature (sensible cooling) and they remove moisture (latent cooling). In hot dry climates only the sensible cooling is beneficial under most conditions. For hot dry climates, the appropriate metric of performance is the Peak Energy Efficiency Ratio sensible (PEERs) measured at high outdoor temperatures with low to moderate indoor humidity. Since this addresses the cause of system electrical peak, PEERs is of particular importance.

2.2. Peak Electrical Consumption Conditions

The performance of the air conditioners at electrical peak conditions as seen in most of California, Nevada, Arizona, and West Texas is described by PEERs. These areas have low outdoor humidity under peak summer conditions. In these areas the introduction of outdoor air into the building dries the indoor air below 65 grains of moisture (dew point 55ºF, 0.0093 lb of water per lb dry air).

The research team determined the design indoor moisture from the CheckMe! database of measured conditions at residential and commercial buildings in California. This database includes technician measured air conditioner return air plenum conditions across the State of California under a wide variety of outdoor conditions. There were 8,156 residential data points and 3,894 commercial data points in the analysis.

Peak electrical consumption occurs because of air conditioners in real buildings as opposed to in the laboratory. One important aspect is that the duct system resistance in real buildings is much higher than that assumed for standard laboratory testing. The research team used the measured values from nine field studies to set the design and test point for duct system resistance. This duct resistance produces a static pressure external to the coil and air handler/furnace of

$$\left(\frac{CFM \ \ per\, ton}{495\, CFM \ \ per \ ton}\right)^2 \ \ Inches\ of\ Water$$

2.3. Baseline Unit Efficiency

The baseline air conditioners for this project served two purposes. First, they provided data to populate a simulation model for comparing various potential design changes to achieve high efficiency at high temperatures and low humidity. Second, they provided the initial base of comparison for the performance and economics of the resulting air conditioners.

The anticipated minimum SEER rating in the United States at the start of the HDAC project was SEER 12. During this project the anticipated minimum SEER rating was increased to 13. In the final analysis the HDAC air conditioners were evaluated against typical SEER 13 R-410A air conditioners.

Two baseline air conditioners were selected. One air conditioner was a typical split system residential unit and the other was a typical light commercial package unit.

The factors used to select the baseline air conditioners included:

- Seasonal Energy Efficiency Rating (SEER)
- Type of refrigerant
- Capacity
- Energy Efficiency Ratio at 95°F (EER95)
- Lack of unique features

- Availability of laboratory baseline data

R-410A was selected as the refrigerant since it is anticipated to be the common refrigerant of the future. The current standard refrigerant (R-22) is being phased out due to its effect on the ozone layer and major air conditioning manufacturers are planning to use R-410A in the future. Market research (Knight 2004) has shown that the most common residential unit is a 3-ton split system. The most common light commercial package rooftop unit is a 5-ton combination gas-pack that has an integrated gas furnace with the air conditioning

Baseline performance data were gathered from laboratory tests at: National Institute of Standards and Technology, Pacific Gas and Electric Company, Purdue University, Southern California Edison Company, and Texas A&M University. For each test the electrical power, cycle state points, and flow were measured. At each laboratory, the instrumentation was similar to that in Figure 1.

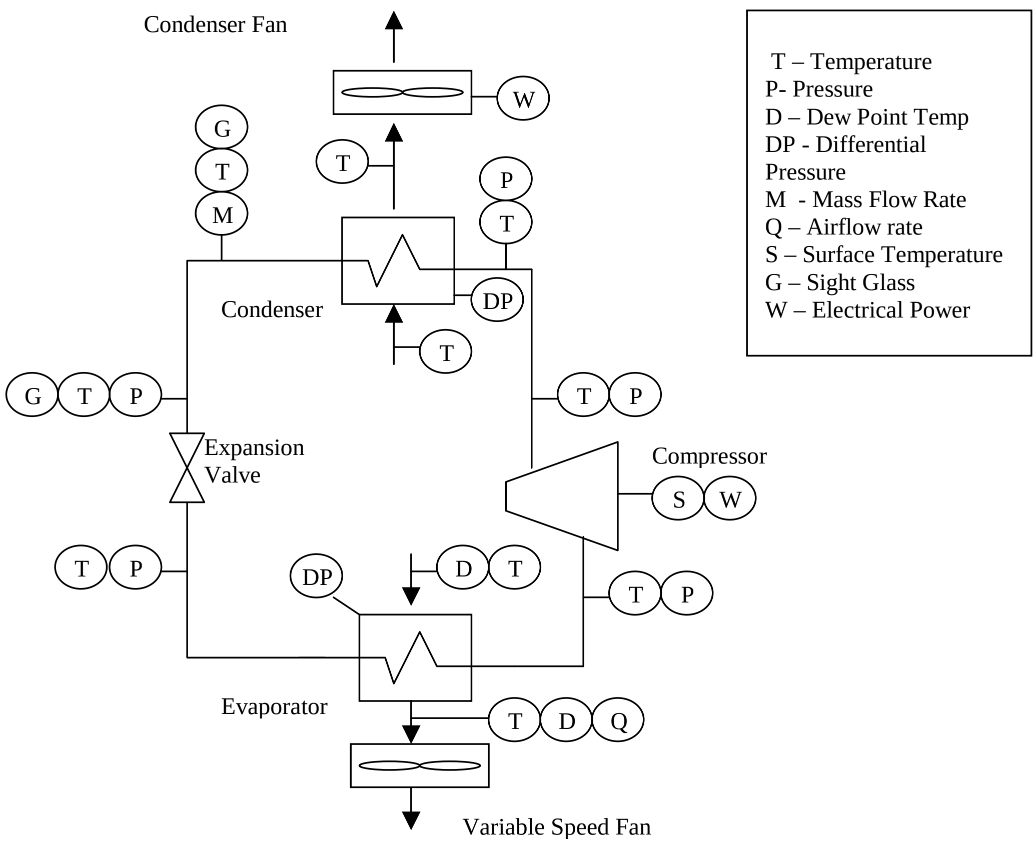

Figure 1: Baseline Testing Instrumentation Schematic

Source: Purdue University

2.3.1. National Institute of Standards and Technology Tests

The National Institute of Standards and Technology (NIST) tested split system 3-ton R22 and R410A residential air conditioners (Domanski and Payne 2002). The selected systems comprised identical evaporators and condensers as well as thermostatic expansion valves (TXVs).

These units were tested at condenser air entering temperatures between 82°F and 135°F. The R22 and R410A systems had a similar capacity and EER at the 82°F (27.8°C) rating point, but the R22 air conditioner was a better performing system at higher temperatures.

2.3.2. Pacific Gas and Electric Company R-410A Comparison Tests

Pacific Gas and Electric Company (PG&E) conducted a series of tests on residential split-system air conditioners using R-22 and R-410A refrigerants (Davis 2001b). The systems were plumbed with both fixed tube orifice and TXV metering devices selectable by refrigerant valves.

These units were tested at condenser air entering temperatures at 95°F and return plenum conditions of 80 °F dry bulb, 67°F wet bulb. The refrigerant charge was varied between 56% and 122% of correct charge. The evaporator airflows were varied between 280 cfm per ton and 400 cfm per ton.

The test results show the system using R-410A has almost exactly the same performance degradation against incorrect charge and lowered airflow as the R-22 system.

The tests showed significant differences between the orifice system and the TXV with respect to refrigerant charge. The TXV performed substantially better than the orifice system when the units were undercharged.

2.3.3. PG&E R-22 Tests

PG&E conducted a series of tests on residential split-system air conditioners using R-22 refrigerant (Davis 2001a). The system was tested with a fixed tube orifice that was converted to TXV metering device for the second set of tests.

The unit was tested at condenser air entering temperatures between 82°F and 115°F with return plenum conditions of 80 °F dry bulb, 67 °F wet bulb as well as 80°F dry bulb, 63°F wet bulb. The refrigerant charge was varied between 60% and 120% of correct charge. The evaporator airflows were maintained at 400 cfm per ton.

The tests quantified the effects of variations in refrigerant charge, evaporator airflow, condenser air entering temperature, and return humidity on the performance of this system when refrigerant was controlled with each of the devices. As expected the tests with lower indoor humidity showed lower Total (sensible + latent) EER and higher sensible EER.

The tests showed significant differences between the orifice system and the TXV with respect to refrigerant charge. When correctly charged, there was little difference in EER between these expansion devices when outdoor temperatures were varied between 95°F and 115°F. The TXV had a somewhat higher EER at 82°F than the fixed metering device. However when undercharged, the TXV consistently had a higher EER and Sensible Heat Ratio than the orifice system. In the undercharged condition, the orifice system draws less power and produces less capacity.

2.3.4. Purdue University Split System Tests

Laboratory testing of the 3-ton split system baseline unit was performed at Purdue University from May 1 to May 31, 2004 (Shen and Jiang 2004a). The unit is described in Table 6.

Table 6: Baseline Split Unit as Tested at Purdue

Item	Split System
Outdoor Unit	Carrier 38EZG036310
Indoor Coil	Carrier FX4BNF036
Furnace	Ruud UGPL-07EBRQR
Capacity	3-ton
Refrigerant	R-410A
SEER	11.5/12
Expansion device	TXV
Compressor	Reciprocating H89B303ABC (Bristol)
Condenser coil	1 row; wavy fin; 2 circuits into 1 subcooled circuit; 25 fins/in 14.8 sf Face Area
Evaporator coil	3 rows; slit fin; 5 circuits; 15 fins/in 3.45 sf.
Condenser fan	1/8 hp, 2400 cfm
Evaporator fan	11 x 10, ¾ HP; PSC
Factory charge	6.25 lbs

Source: Purdue University

Test Conditions

The baseline unit was tested at ARI SEER rating conditions and other test conditions as shown in Table 2.

One objective of this project was to assess and mitigate the effect of normal installation including the low evaporator airflow and incorrect refrigerant charge found in field studies. The Purdue laboratory work includes tests with low airflow and/or incorrect refrigerant charge.

Test #9 was the design point for peak reduction at hot/dry conditions. This test was performed at evaporator airflows of 350 cfm per ton.

Air conditioners located on roofs can experience higher temperatures than ambient.
Test 12 (127°F) condenser air entering temperature determined the effect of extreme conditions beyond the manufacturers' standard tests (Table 7).

Table 7: Purdue Baseline Test Conditions

Test	Outdoor Dry Bulb(ºF)	Indoor Dry Bulb(ºF)	Indoor Wet Bulb(ºF)	Charge	Indoor Airflow(CFM)
1	115	80	67	100%	1200
2	95	80	67	100%	1200
3	115	80	63	100%	1200
4	115	80	63	120%	1200
5	115	80	63	90%	1200
6	115	80	63	81%	1200
7	115	80	63	71%	1200
8	115	80	63	100%	900
9	115	80	63	100%	1050
10	115	80	63	100%	1350
11	115	80	63	100%	1650
12	127	80	63	100%	1200

Source: Purdue University

2.3.5. Package Unit Laboratory Tests

There were no laboratory tests immediately available that evaluated the type of unit or conditions suitable for the baseline of the 5-ton package unit. Two sets of laboratory test data were evaluated for information relevant to this project. These data were generated at Purdue University and Southern California Edison facilities.

Purdue Package Unit Equipment

Laboratory testing of a 3-ton package unit (Carrier 50GL036) was performed at Purdue University from August 1, 2004 to October 31, 2004. The Carrier 50GL036 and the 5-ton package unit Carrier 48GP060090 are similar configurations. The 50GL036 has an open plenum where the 48GP060 has a gas furnace heat exchanger in that plenum. Both units are R-410A machines with 12 SEER.

The test conditions were duplicates of the tests done on the 3-ton split baseline unit as listed in Table 2.

As before, the comparison points were Tests #1 and #9. Test #9 conditions included evaporator airflows of 350 cfm per ton since this is the average airflow found in the field with PSC motors set on their highest speed.

2.3.6. Southern California Edison Company Tests

Southern California Edison's Refrigeration and Thermal Test Center tested six 5-ton package air conditioners manufactured by three companies (Faramarzi et al. 2002). Three of the units were standard efficiency (SEER 10) units and three were high efficiency units. The SEERs of the high efficiency units were 13, 12, and 13. All the units used R-22 refrigerant, which performs better at elevated ambients than R-410A, the baseline refrigerant for this project.

Test Conditions

Testing was performed at standard ARI rating indoor conditions (80°F dry bulb and 67°F wet bulb). The condenser air entering conditions were: 85°F, 95°F, 105°F, 115°F, 120°F, 125°F and 130°F.

None of the tests were performed at the lower humidity ratio targeted for this project. All tests were run between 358 and 388 cfm per ton.

2.3.7. Texas A&M University Tests

Texas A&M conducted a series of tests on ten air conditioners at high outdoor temperatures (Bain et al. 1995). These tests included seven split systems and three package systems ranging in size from two to four tons. Within the ten units there were scroll and reciprocating units, TXV, capillary, and orifice units, as well as heat pumps.

These units were tested at six condenser air entering temperatures ranging from 82°F to 120°F and return plenum conditions of 80°F dry bulb, 67°F wet bulb.

The package units showed more performance (EER) degradation as outdoor temperature rose than the split units possibly because the package units' evaporator sections were exposed to outdoor conditions. The test EERs were compared with the manufacturers' published data and found to be, on average, 2.6% less than the manufacturers' data.

2.4. Baseline Unit Modeling

Test data and the physical parameters were used to create a computer model of the baseline units. The Oak Ridge National Lab Heat Pump Design Model was used for modeling the baseline units (ORNL 2002). The model was calibrated to baseline test data when available. These models were the starting points for the design process.

Phase 1 modeled the gross performance of the units (unit performance without inside fan watt draw and duct system effects).

2.4.1. Oak Ridge Design Model

"A widely used tool developed by DOE through Oak Ridge National Laboratory is the Heat Pump Design Model (Mark VI release). This tool simulates the steady-state cooling and heating performance of air-to-air heat pumps and air conditioners, enabling users to specify such key parameters as type of vapor compressor, type of heat exchanger, air conditions, air flows, and type of refrigerants. The program can be used with R-410A. Manufacturers of heat pump systems and components have used the model extensively in product design and ratings, and contractors have applied it to assessing installation designs." (DOE 2004)

This tool provides adjustment multipliers to fit the model to measured data. The adjustment multipliers include: compressor power, refrigerant mass flow, airside and refrigerant-side coil heat transfer, refrigerant pressure drops and air pressure drops.

2.4.2. Baseline Modeling Sequence

Phase 1 of the modeling started with the gross performance (Capacity and EER) from the test data supplemented by the manufacturer's data at 400 and 350 cfm per ton. The known physical parameters of the units and the measured superheat and subcooling were entered into the model. The manufacturer's supplied compressor map was used in the model.

The model was tuned using the coil heat transfer and pressure drop adjustment factors so it closely matched the measured internal pressures, temperatures, flows, gross capacities, and sensible heat ratios from the test data. Finally, the compressor power and mass flow rate adjustment factors were adjusted to match the EERs at 400 and 350 cfm per ton.

The primary adjustment point was the gross sensible EER at 115/80/63 with 350 cfm per ton. The model was then checked against other conditions and airflows.

Phase 2 modeling estimated how the baseline units would operate under field conditions[1]. Measured values for airflow and external static pressures were used to estimate power draw, EER, and capacity. Since a primary goal was to reduce peak electrical consumption in real installations, it was important to determine performance under field conditions. The resulting models provide baseline sensible capacities and sensible EERs at multiple conditions.

Table 8 shows parameters that were used in the Phase 2 models.

Table 8: Baseline Modeling Net (including evaporator blower effects)

	Phase 2 3-Ton Split Baseline Model	Phase 2 5-Ton Package Baseline Model
Evaporator Flow (CFM)	1050	1750
External Static Pressure[2] (IWC)	0.50	0.50
Evaporator Fan/Motor Eff.	19.2%	18.6%4
Condenser Fan/Motor Eff.	4.6%	7.5%

Source: Proctor Engineering Group

Phase 2 modeling adjusts the gross EERs in the preceding section by adding the fan power and fan heat into the equations as well as reducing the fan flow to 350 cfm per ton.

These models served as the starting point for modeling potential new designs. The research team emphasizes that these models were only design tools. The performance of the proof-of-concept units in the laboratory was initially compared to the baseline laboratory tests and then

[1] Field measured values were used for evaporator fan watt draws rather than manufacturers' published ratings because: 1) Rated fan watt draws are not available for most furnaces that have PSC motors, 2) Split units are generally rated without a specific furnace, and 3) The field measured data is a true picture of performance, which can vary significantly from the manufacturers' data based on the wide variety of duct configurations in the field.

[2] Duct and assessor static pressure – this excludes the evaporator coil.

to manufacturer's published data for revised baseline units with SEER 13 (the new national standard).

Baseline 5-Ton Commercial Package System

Test data were not available for a SEER-12 5-ton R-410A package unit. Manufacturer’s published data were used in conjunction with the SCE and Purdue Lab data to tune the ORNL model.

The known physical parameters of the baseline 5-ton package unit (Carrier 48GP060090) and the measured superheat and subcooling of the Purdue tested 3-ton package unit (Carrier 50GL036) were entered into the model and adjusted to avoid out of range results. The model was adjusted so it closely matched the measured internal pressures and temperatures of the Purdue tested unit. The model was further adjusted so it closely matched the gross Capacities and EERs of the baseline 5-ton unit at 400 and 350 cfm per ton.

2.5. Component Investigation and Selection

The team researched a broad range of potential component designs for use in the HDAC units. The potential candidates were evaluated for cost, performance, availability, maintenance issues, durability and reliability.

The components and their effects on the cost and performance of the designs were evaluated using; manufacturers' data and research reports; published technical reports; modeling of refrigerant cycles, including simple models as well as EVAP-COND (Domanski and Payne 2002) and vender proprietary models including a detailed system model composed of extensively validated component models based on a Newton-Raphson solver. The model employed airside heat transfer and pressure drop correlations developed for microchannel heat exchangers (Bullard et al. 2006).

2.6. Modeling HDAC Units

These baseline models were the starting points for the design process. The design point for this project was: 115°F outdoor temperature, 80°F return plenum dry bulb temperature, and 63°F return plenum wet bulb temperature.

Through iteration the input parameters were adjusted to provide designs that met the performance criteria of this project. Tradeoffs were made between design options (such as larger condenser coils vs. larger evaporator coils) based on the relative cost effectiveness of the options. Over 900 iterations were performed in the design process. These iterations included step-by-step interactive incremental changes, and on occasion, major jumps in most of the input parameters. The occasional major jumps reduce the likelihood of producing local optimums that have less efficiency than other optimums discernable when the model approaches from another “angle”.

2.6.1. Design Criteria and Considerations

The main design criteria were to:

- Reduce Lifecycle Cost
- Reduce Peak Electrical Consumption by 15% to 25% in Hot Dry Climates
- Energy Saving of 10% to 20%
- Initial Cost increase 5% to 25% of Baseline
- Produce a Buildable and Marketable Design

The following considerations were included in the process:

1. Design for easy adoption
    - component availability,
    - smallest effective deviation from existing practice,
    - potential for mainstreaming (avoid technologies that manufacturers shy away from)
2. Proven performance
    - in previous tests,
    - in modeling
3. "Drop in" replacement for baseline unit
    - cooling capacity,
    - footprint,
    - supply and return locations
    - heating capacity

2.6.2. Design Features and Revisions

A large number of features and revisions were evaluated by analysis and modeling for the HDAC. The revisions that received the greatest attention in the final design process are cataloged in the following lists. The first list contains features that improve the efficiency of air conditioners at most conditions. These features were optimized within the modeling to high outdoor temperature and lower indoor humidity conditions trading off with other features.

General efficiency features include:

- Increased UA evaporator coils
- Increased UA condenser coils
- Increased efficiency compressors
- Lower saturation temperature condenser coil
- Reduced suction superheat

- Improved evaporator blower/motor efficiency
- Improved condenser fan/motor efficiency
- Reduced inside cabinet and heat exchanger air restrictions
- Reduced outside cabinet and heat exchanger air restrictions
- Increased liquid subcooling
- Higher airflow across the condenser coil
- Oval tube heat exchangers
- Rifled tube augmentation
- Microchannel heat exchangers
- Suction superheat, liquid subcooling heat exchange
- Improved two speed PSC blower motor (non-slip speed control)
- Airfoil axial evaporator blower
- Start up and shut down controls to manage refrigerant migration

The second list contains features that focus more on increasing the efficiency of air conditioners at high outdoor temperature and lower indoor humidity conditions.

Hot dry efficiency features include:

- Higher saturation temperature evaporator coil
- Higher airflow across the evaporator coil
- Controls to minimize latent capacity under dry indoor conditions
- Increased condensate retention on the evaporator coil
- Controls to obtain latent capacity when indoor moisture rises significantly
- Controls to evaporate moisture off the coil rather than allow condensate drainage

2.7. Component and Assembly Tests

2.7.1. Condenser Airflow Testing

The efficiencies of condenser fan assemblies are low. Since the watt draw of the condenser fan is small compared to the watt draw of the compressor, little attention has been paid to this portion of the system. As peak electric power and improvements on the refrigerant side of the system become more costly there will be increased emphasis on this portion of the system, particularly if the performance at high temperatures is given a higher priority. The efficiency of the airflow through the condenser offers two opportunities. First there is the possibility of lowering the watt draw of the condenser fan and second there is the possibility of increasing the airflow across the condenser coil, which lowers the watt draw of the compressor and increases the capacity of the air conditioner.

Extensive airflow optimization was performed at the research group's test facility. Test results were applied to the U.S. DOE ORNL heat pump model to predict the system efficiency effects of various condenser airflow revisions.

Test Facility

The researchers constructed a condenser airflow test apparatus capable of accommodating various condenser fans and flow configurations. The primary method of comparing changes in configuration was recorded changes in the measured face velocity at multiple points on the condenser coil. In the case of the 5-ton package condenser tests, these velocities were used to estimate airflow in CFM.

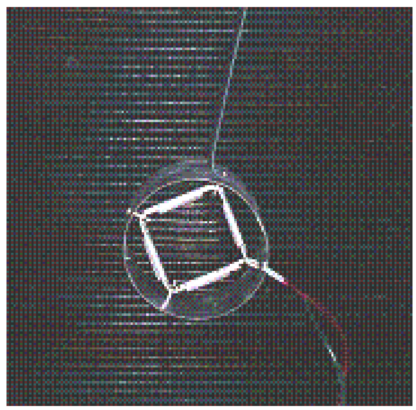

Figure 2: Circular Flow Station at Surface of the Condenser Coil

Source: Proctor Engineering Group

A 6-inch circular flow station manufactured by vanEE (Figure 2) was used for the velocity and velocity distribution measurements. Measurements were taken at consistent points on the face of the condenser coils. These points were at the top, middle and bottom of the coils along the vertical centerline of each face. There were 12 consistent points for the package unit, which had four sides, and 9 consistent points for the split unit, which had three sides.

For the 5-ton package unit, tests were run to estimate the actual flow. For these tests, the condenser fan was replaced with a calibrated Minneapolis Blower Door fan. With this configuration, the face velocities were measured and a face area weighted average velocity calculated. This data was obtained for 12 tests at various flow rates and the measured changes in flow correlated to within 3% with the measured changes of the weighted average face velocities. The Minneapolis Blower Door fan was then replaced with the test condenser fans and the flows estimated by the ratio of the weighted average velocities.

This test apparatus is sufficient to measure the relative airflows for comparisons between configurations.

The fan electrical power, fan speed, average condenser coil face velocity, dry bulb temperature, and line voltage were measured.

Fan speed was measured by placing a LED and receiver on opposite sides of the fan blades. The receiver output an electrical signal when light from the LED was detected. As each fan blade passed between the receiver and LED, light from the LED was obscured and the receiver signal became very small. An oscilloscope was used to measure the receiver output signal oscillation frequency. Fan rpm was calculated as a function of the number of fan blades and the signal oscillation period.

Fan input power was measured with an Ohio Semitronics GH-series watt transducer accurate to ± 0.2%.

2.7.2. 3-Ton Split Evaporator Airflow Testing

Efficiency of residential air handlers is low, typically 10 to 15%. Poorly designed cabinets and restrictive duct systems can further decrease efficiency to only 7%. In the past, little emphasis was placed on air handler efficiency because it was not required in the SEER rating, and represented a relatively small portion of total energy use. As air conditioners and furnaces have become more efficient in recent years, the impact of air handlers on overall system efficiency has grown.

Efficiency detractors in typical air handlers include:

- Inefficient electric motors
- Fan and housing not designed for efficiency
- Air handler cabinet restrictive at fan inlet
- Furnace heat exchanger restrictive at fan outlet
- Poorly designed duct systems

The 3-ton Hot/Dry Air Conditioner (HDAC) air handler was designed for high efficiency at realistic duct system resistance: 0.5 inch water column static pressure at a flow rate of 1050 cfm.

Extensive airflow optimization was performed at the research group's test facility. Test results were applied to the U.S. DOE ORNL heat pump model to predict the air handler design for optimum air conditioner PEERs at hot/dry conditions.

Test Facility

The researchers constructed an air handler test apparatus capable of accommodating various fan and furnace configurations. The test apparatus consisted of a 14.5 foot long plenum with a 20″ by 20.5″ cross section. Static pressure was measured at four locations along the plenum, and airflow was measured at one location. Windows were built into the plenum at the furnace heat exchanger and evaporator coil so that airflow characteristics could be studied visually (Figure 3).

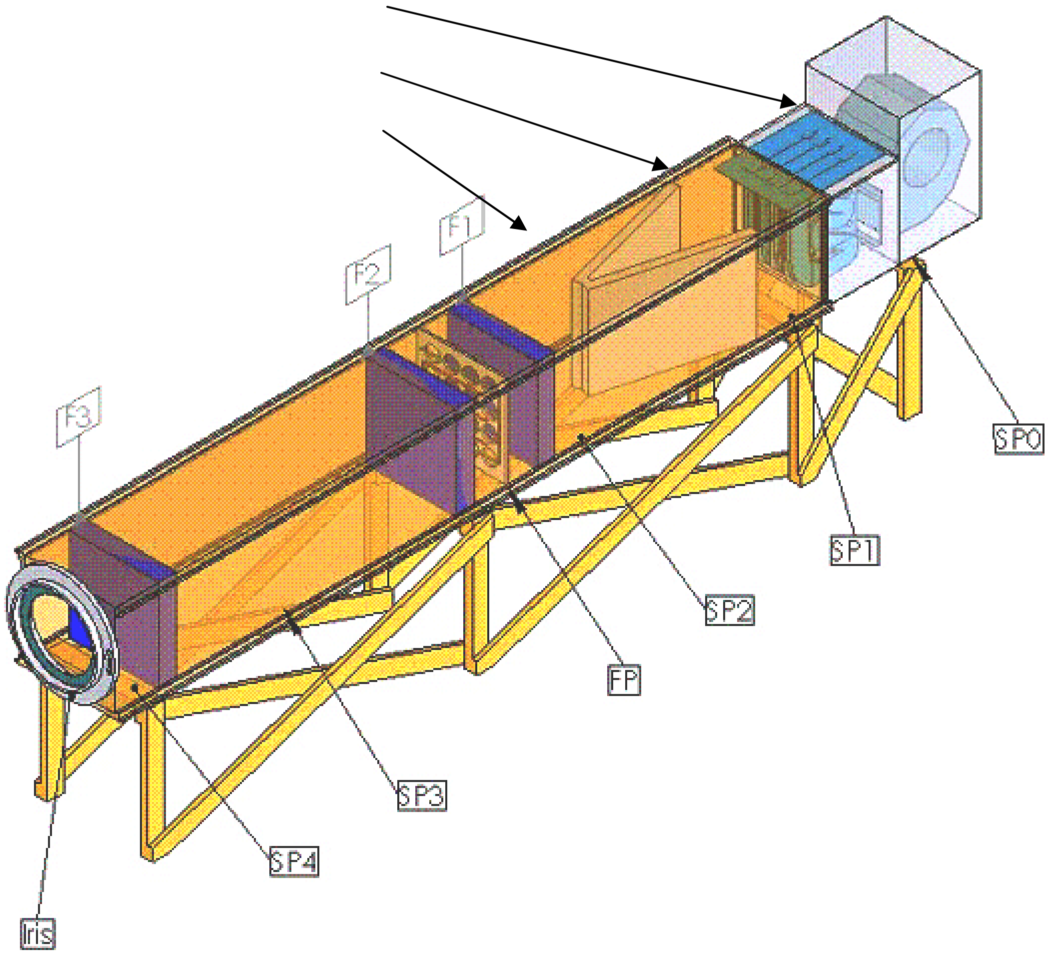

Figure 3: Furnace/ Air Handler Test Apparatus

Source: Proctor Engineering Group

Figure 3 Notations:

- SP0, SP1, SP2, SP3, SP4 — Static Pressure
- F1, F2, F3 — Flow straightener (honeycomb)
- FP — Flow Plate
- Iris — Iris damper

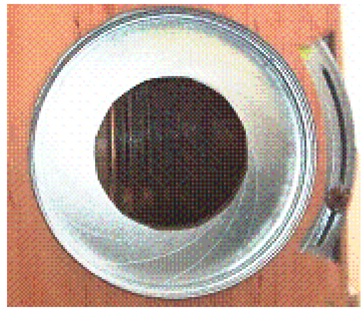

Figure 4: Iris Damper

Source: Proctor Engineering Group

Airflow resistance was controlled by an iris damper located at the outlet of the test apparatus (Figure 4). Opening the iris decreased airflow resistance, while closing the iris increased resistance.

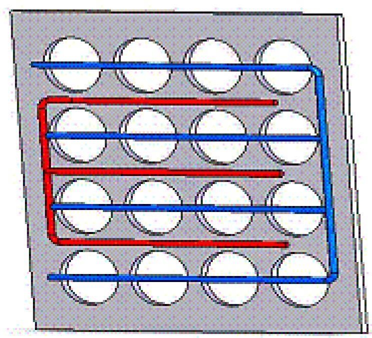

Figure 5: Flow Plate

Source: Proctor Engineering Group

Airflow and static pressures were measured with an Energy Conservatory DG-3 digital pressure gauge. Airflow was measured using a flow plate permanently installed in the test apparatus (Figure 5). Honeycomb flow straighteners were installed before and after the flow plate to improve measurement accuracy.

This test apparatus is sufficient to measure the relative airflows for comparisons between configurations.

Fan speed was measured by placing a LED and receiver on opposite sides of the fan blades. The receiver output an electrical signal when light from the LED was detected. As each fan blade passed between the receiver and LED, light from the LED was obscured and the receiver signal became very small. An oscilloscope was used to measure the receiver output signal oscillation frequency. Fan rpm was calculated as a function of the number of fan blades and the signal oscillation period.

Fan input power was measured with an Ohio Semitronics GH-series watt transducer accurate to ± 0.2%.

Testing and Analysis Protocol

Test apparatus system resistance was measured so that performance could be evaluated at appropriate conditions. Static pressure was measured at varying airflow rates for each iris damper position. Measured static pressure was curve fit against airflow to define system resistance curves. Design system resistance was 0.5″ water column static pressure at 1050 cfm airflow. Performance was also evaluated at a higher and a lower resistance to verify the air handler would maintain efficiency over a range of conditions. The system resistance curves used to evaluate air handler performance is shown in Figure 6.

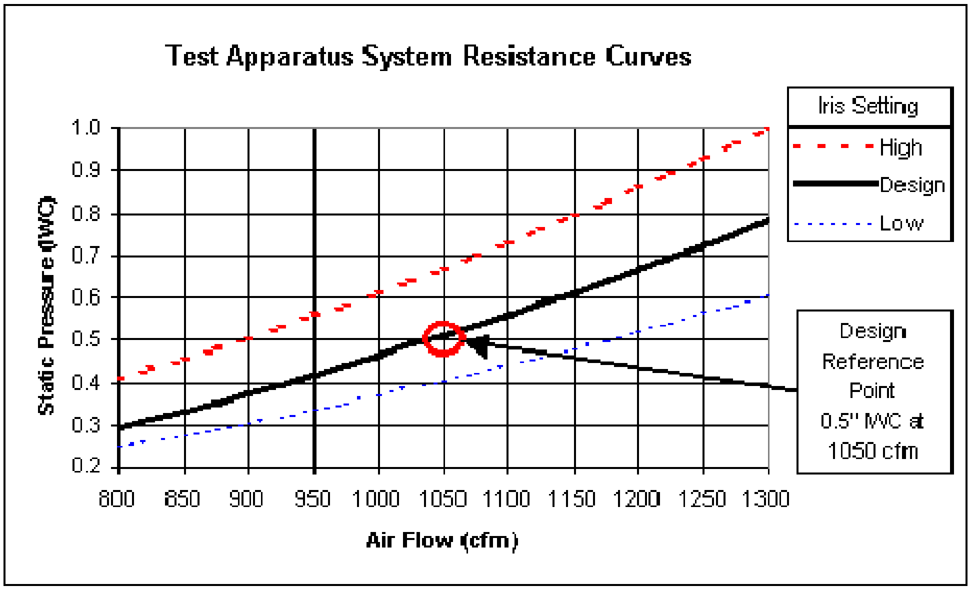

Figure 6: Test Apparatus System Resistance

Source: Proctor Engineering Group

Improvements were tested individually. All tests were performed using an 825 rpm 1/3 Hp PSC motor. The tests were performed with the furnace heat exchanger and evaporator coil in place to provide realistic conditions for improvement evaluation.

Each air handler improvement was tested at varying iris damper settings. Static pressure was curve fit against airflow to calculate the intersection with the system resistance curves and determine the operating point. Power input was curve fit against airflow, and watt draw at the operating point was calculated.

Efficiency was calculated as: h = Q*Dp/W

Where: Q = Volumetric air flow

Dp = Pressure difference across the fan

W = Power input

Airflow, input power, and efficiency were measured at design system resistance as well as higher and lower than design system resistance. Test results were applied to the DOE ORNL model to determine the air handler design for optimal PEERs.

2.7.3. 5-Ton Package Evaporator Section Airflow Testing

The goal of the package unit evaporator section airflow testing was a configuration that would potentially provide higher than average evaporator coil airflow at lower than average fan watt draws. Simultaneously the configuration had to provide a reasonably uniform distribution of airflow velocities across the coils.

Package rooftop air conditioners have the same efficiency detractors that are typical in residential furnaces/air handlers. In addition they are confined to a small footprint.

Four metrics of airflow were used: airflow rate, external static pressure, fan motor power draw, and evaporator coil flow distribution. Package air conditioners generally have two options for the positions of the supply and return openings, bottom and side. For all of the testing, the side openings were used, though the bottom openings were left unobstructed so the product would retain dual inlet/outlet orientations.

Extensive airflow optimization was performed at the research group's test facility. Test results were applied to the U.S. DOE ORNL heat pump model to predict the air handler design for optimum air conditioner PEERs.

Test Facility

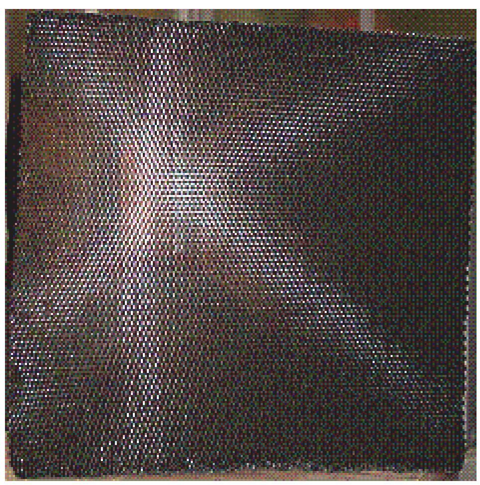

Figure 7: Honeycomb Flow Conditioner

Source: Proctor Engineering Group

To obtain consistent pressure and flow measurements, two 4-ft sections of duct were attached to the supply and return inlets. Each duct had 2 honeycomb flow straighteners as shown in Figure 7. A 20"x 20" TrueFlow© flow-plate was mounted between the flow straighteners in the return duct. A micro-manometer measured the flow plate pressure difference and the flow was calculated using the standard TrueFlow equations (The Energy Conservatory 2006). The TrueFlow flow meter is accurate to ± 7% of the full-scale flow.

External static pressure was measured using static probes mounted in the ducts at the package unit inlet and outlet. These probes were attached to a DG-700 Energy Conservatory micro-manometer. The accuracy of this digital pressure gauge is ±1%.

Fan input power was measured with an Ohio Semitronics GH-series watt transducer accurate to ± 0.2%.

Evaporator coil airflow distribution was investigated with three techniques: Pitot tube measured face velocities, smoke pen flow visualization, and infrared camera thermal imaging.

The Pitot tube measurements were most consistent when situated directly downstream of the coil.

The smoke pen method allowed visual confirmation of airstreams, and areas of turbulence, but was only viable with airflow rates less than 500 CFM.

The thermal imaging camera provided the most effective method of judging distribution across the coil and the effects of the distribution. With the air conditioner operating, airflow misdistribution and/or refrigerant flow was determined from infrared images of the coil. Regions receiving greater than average airflow were at higher temperatures, while regions with insufficient airflow were at lower temperatures. To determine which observed differences were due to airflow distribution, diverters and turning vanes were installed, and the temperature regions moved as expected. Figure 8 shows an image from the thermal camera.

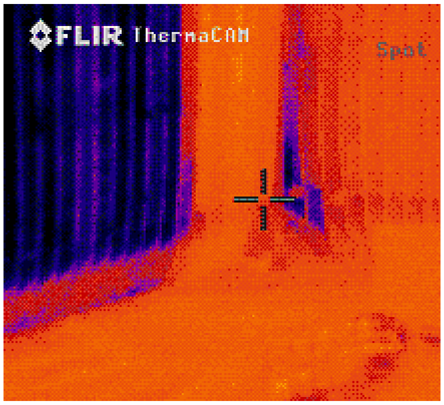

Figure 8: Thermal Image of an Evaporator Coil

Source: Proctor Engineering Group

2.8. HDAC Assembly and Development Tests

The HDACs were assembled in the research team's laboratory and tested under high ambient temperatures. Assembly used standard brazing and mechanical fastening techniques.

The development laboratory does not have psychrometric rooms. High ambient temperatures were created by recirculation of the heated air exiting the condenser. Condenser exiting air was trapped by a large tent surrounding the unit. The air was circulated within the tent and a controlled amount of ambient air was introduced to stabilize the condenser air entering temperature.

The airflows were measured with the test apparatus described in Sections 2.6.2 and 2.6.3. Inlet and outlet air temperatures were measured by grids of thermocouples. Refrigerant temperatures were measured with thermocouples thoroughly insulated from air temperatures and in direct contact with the refrigerant lines.

With this test apparatus, the team was able to compare the performance when components and configurations were changed. This apparatus was used to select between available condenser and evaporator coils and to determine the effects of a suction line liquid line heat exchanger.

The suction line liquid line heat exchanger was plumbed into the system such that the heat exchanger could be bypassed.

2.9. Laboratory Test HDACs and Optimize Parameters

Performance data were gathered from laboratory tests at Southern California Edison Company for the 5-ton unit and at Pacific Gas and Electric Company for the 3-ton unit. For each test the electrical power, cycle state points, and flow were measured. The psychrometric rooms for each of the laboratories were equipped as described in
Section 2.2 Baseline Unit Efficiency.

2.9.1. Test Facilities

The SCE Refrigeration and Thermal Test Center is a 3,800 square-feet testing facility located in SCE's CTAC complex in Irwindale, California. It contains supermarket, HVAC, and refrigerated walk-in test chambers.

The PG&E Technical and Ecological Services (TES) is a 13-acre facility serving the needs of Pacific Gas and Electric Company. TES provides specialized testing and analytical services across a wide spectrum of disciplines including HVAC systems.

The HVAC testing areas consist of indoor and outdoor psychrometric rooms in which the temperature and humidity can be controlled to simulate a range of inside and outside conditions. The "Inside Room" acts as a space from which the test unit draws air representing the conditions inside the building. The "Outside Room" is used to create the outside air condition. The 5-ton package unit was placed in the outside room for testing with inside room air ducted to the unit. The condenser section of the 3-ton split unit was placed in the outside room with refrigerant lines running to the inside room for the evaporator section. The 3-ton unit indoor section was placed in the indoor room. Evaporator airflow measurements were made with airflow chambers compliant with national standards. The PG&E outdoor room also had a compliant airflow chamber to measure the condenser airflow. Each chamber is equipped with a variable speed booster fan to compensate for the added resistance of the chamber.

2.9.2. Test Conditions

The HDAC units were tested at Hot Dry conditions as well as those necessary to establish SEER. These tests determined:

- Performance and optimum refrigerant charge and subcooling under hot and dry conditions
- Performance and optimum evaporator airflow rate under hot and dry conditions
- Performance under standard conditions for SEER
- Performance utilizing latent recovery

Optimum Refrigerant Charge and Subcooling for Hot, Dry Conditions

Refrigerant charge optimization was done at the design point for the HDAC 115°F outside, 80°F dry bulb and 63°F wet bulb inside. Airflow through the evaporator section was constant with 0.50 in. of duct system static pressure across the unit.

For this series of tests, subcooling and the amount of refrigerant were changed step by step to establish the optimum subcooling, superheat, and refrigerant charge for each unit. When data had been collected for a range of subcooling and refrigerant charges, the optimum settings were determined by the highest PEERs.

Optimum Airflow Rate for Hot, Dry Conditions

For this series of tests, the outdoor environment room was maintained at 115°F and the indoor room at 80°F dry bulb and 63°F wet bulb. The refrigerant charge and controlled subcooling were set to the optimum performance determined in the Optimum Refrigerant Charge and Subcooling Tests. Airflow through the indoor section was controlled to a static pressure external to the indoor section of $\left(\frac{CFM\ \ per\,ton}{495\,CFM\ \ per\ ton}\right)^2\ \textit{Inches of Water}$ while airflow was varied. When data had been collected for a range of airflows, the optimum flows were determined by the highest PEERs (net sensible EER).

Performance at Standard Rating Conditions

Once the optimal combination of refrigerant charge, subcooling, and airflow rate were determined at 115°F, additional tests were completed to obtain SEERs. The tests for determining SEER are specified in ARI Standard 210/240 (ARI 2003). The three tests used to determine SEER on single speed equipment are called Test B, Test C, and Test D. The test conditions are listed in Table 9.

Table 9: ARI SEER Test Conditions

Test	Evaporator Air Entering Dry Bulb	Evaporator Air Entering Wet Bulb	Condenser Air Entering Dry Bulb	External Static Pressure
B Cooling Steady State	80ºF	67ºF	82ºF	0.15 for 3-ton 0.20 for 5-ton
C Cooling Steady State Dry Coil	80ºF	57ºF or less	82ºF	0.15 for 3-ton 0.20 for 5-ton
D Cooling Cyclic Dry Coil	80ºF	57ºF or less	82ºF	0.15 for 3-ton 0.20 for 5-ton

Source: Air-Conditioning, Heating, and Refrigeration Institute

Since the SEER test uses much lower external static pressures than used in the HDAC design and testing, the HDACs' airflows were reoptimized for SEER. Higher SEERs are obtained by manufacturers using a hard shut off valve to separate the high pressure and low-pressure sides of the system. The HDAC SEER tests were run with a hard shut off on the metering valve.

2.9.3 Performance with Latent Recovery

In dry climates like California there is wasted cooling capacity that is sent "down the drain" in the form of condensate from the evaporator coil. The air conditioner cycle can be changed to harvest this cooling capacity by running the fan at the end of the compressor cycle which evaporatively cools the air returning to the building. The test conditions used for this test are shown in Table 10.

Table 10: Latent Recovery Test Conditions

Test Description	Outdoor DB (°F)	Indoor DB (°F)	Indoor WB (°F)	Airflow (CFM)	Static Pressure (IWC)
Latent Recovery	115	80	67	1750 (5-ton) 1100 (3-ton)	0.5

Source: Proctor Engineering Group

2.10. HDAC Performance Standard

The Proof-of-concept HDACs were designed to investigate and illustrate potential improved performance in air conditioners when they are optimized to hot/dry conditions. Once the laboratory tests were completed, an HDAC performance standard was drafted based on the test results. The resulting HDAC standard was less than the performance demonstrated by the proof-of-concept units, but it was chosen as within reasonable reach by the major manufacturers (Table 11).

Table 11: Hot, Dry Air Conditioner Draft Specifications

Condition #1	Hot Dry 115/80/63
Gross Sensible Capacity (sensible btuh)	75% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 8 btu/watthr
Condition #2	Hot Medium 115/80/67
Gross Sensible Capacity (sensible btuh)	65% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 6.8 btu/watthr

Source: Proctor Engineering Group

2.11. Compare Laboratory Tests to Production Air Conditioners

Following the laboratory tests, the results were compared to production air conditioners. Since the manufacturers do not publish performance data for their machines at the HDAC conditions, the published "expanded performance tables" which list estimated cooling capacities and unit powers at various indoor conditions, outdoor conditions and evaporator airflows had to be used. These were used in combination with published performance for evaporator coils and forced air furnaces.

The process followed these steps:

1. Interpolate the total capacity, sensible capacity, and unit watt draw for the HDAC evaporator air entering wet bulb temperature.
2. For units rated as net capacities and without a specific evaporator fan, increase the sensible and total capacities by 1.25 times the airflow in cfm[3]. This produced gross capacities.
3. For units rated as net capacities and with a specific evaporator fan, determine the fan watt draw at the ARI minimum external static pressure and increase the sensible and total capacities by 3.414 times the watt draw of the evaporator fan. This produced the gross capacities.
4. For units rated as net watt draws add the watt draws of the evaporator fan as determined in steps 2 or 3 above. This produced the gross watt draws.
5. Using data on the actual evaporator fan in the proposed HDAC unit, curve fit the relationship between the airflows and the static pressures at the proposed fan settings. Plot the result.
6. For split units curve fit the relationship between the evaporator airflows and the static pressure drop across the evaporator coil. Add the static pressure drop of the typical duct

[3] Standard ARI assumption for units tested without a specified evaporator fan.

system $(\frac{CFM\ \ per\, ton}{495\, CFM\ \ per\ ton})^2$ *Inches of Water* to the pressure drop across the evaporator coil and plot the result with the results from step 6. The intersections are the potential system operating points.

7. For package units plot the static pressure drop of the typical duct system $(\frac{CFM\ \ per\, ton}{495\, CFM\ \ per\ ton})^2$ *Inches of Water* against the evaporator airflow with the results from step 6. The intersections are the potential system operating points.

8. Determine the blower power for the evaporator fan for the potential operating points of interest.

9. Determine $\text{PEERs} = \frac{\text{Gross Sensible Capacity - 3.412 * (Blower Power)}}{\text{Compressor Power + Blower Power + Condenser Fan Power}}$ for each potential operating point of interest.

2.12. Select Standard Manufactured Component Assemblies for Field Test

Within the timeframe of the project, four manufacturers expressed an interest in HDACs, particularly if a combination of their existing components could meet the HDAC standard.

The research team evaluated products by seven manufacturers including both split and package units using the methodology in Section 12.11. From that evaluation, units that approached the standard (none met the standard outright) were selected for field test against production SEER 13 units.

3.0 Laboratory Prototype Testing Results

3.1. Component and Assembly Test Discussion and Results

3.1.1. 3-Ton Split Indoor Airflow Test Results

Fan Housing

A standard sheet metal fan housing was compared to a molded plastic housing. The same fan and motor were used for the tests, only the housing was changed. Figure 9 and Figure 10 represent only the housing differences.

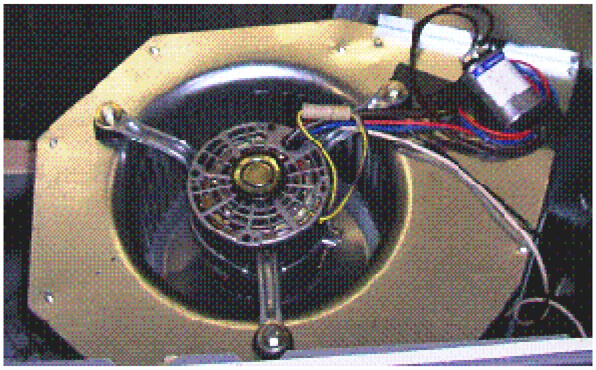

Figure 9: Sheet Metal Fan Housing

Source: Proctor Engineering Group

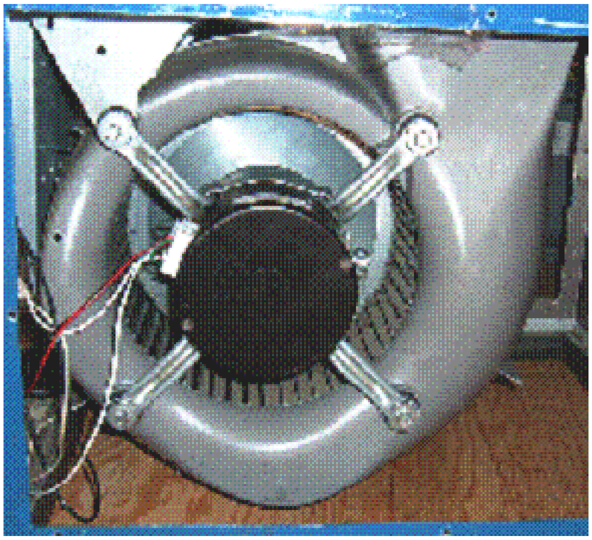

Figure 10: Molded Plastic Fan Housing

Source: Proctor Engineering Group

The molded plastic housing delivered 160 cfm more airflow at the design system resistance (Figure 11). The air handler consumed less power per unit airflow with the plastic fan housing than with the sheet metal housing. The difference increased with increasing static pressure (Figure 12 and Table 12).

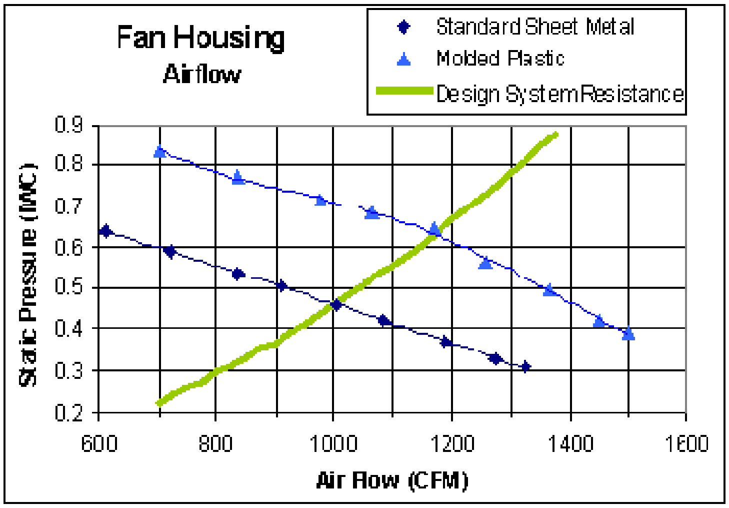

Figure 11: Fan Housing Airflow

Source: Proctor Engineering Group

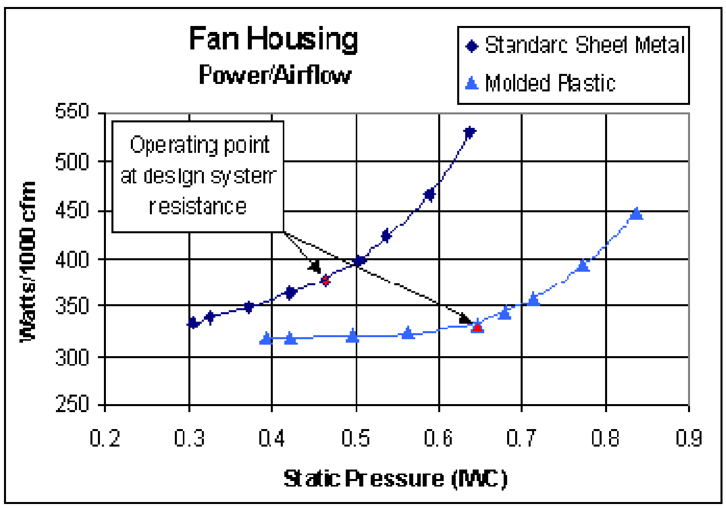

Figure 12: Fan Housing Effect on Power Consumption

Source: Proctor Engineering Group

Table 12: Fan Housing Comparison

	System Resistance	Metal Housing	Plastic Housing	% Difference
Airflow (cfm)	Low	1079	1256	16.3
	Design	1001	1161	16.0
	High	904	1060	17.2
Power (W)	Low	394.0	399.2	1.3
	Design	379.7	376.9	-0.7
	High	363.3	356.1	-2.0
Efficiency (%)	Low	13.6	20.8	52.1
	Design	14.3	22.2	55.4
	High	14.8	23.1	56.0

Source: Proctor Engineering Group

Air handler efficiency was 55% higher with the molded plastic fan housing than with the standard sheet metal housing. The plastic housing was more efficient at all conditions, but the improvement increased with increasing system resistance.

Furnace Heat Exchanger

A tube heat exchanger was compared to a clamshell style heat exchanger (Figure 13). The "fins" used to block air from bypassing the heat exchanger were also studied. A curved fin was compared to the standard flat fin to test if a more aerodynamic flow path would improve efficiency (Figure 14 and Figure 15).

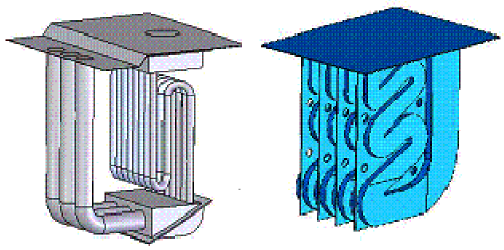

Figure 13: Furnace Heat Exchangers: Tube (Left) and Clamshell (Right)

Source: Proctor Engineering Group

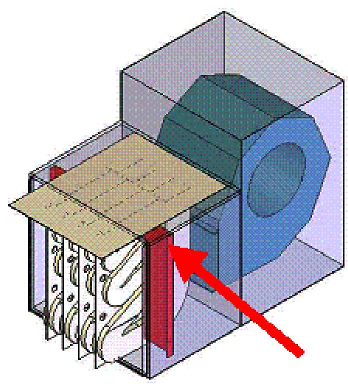

Figure 14: Heat Exchanger 'Fins'

Source: Proctor Engineering Group

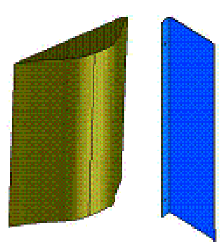

Figure 15: Curved (Left) & Flat (Right) 'Fins'

Source: Proctor Engineering Group

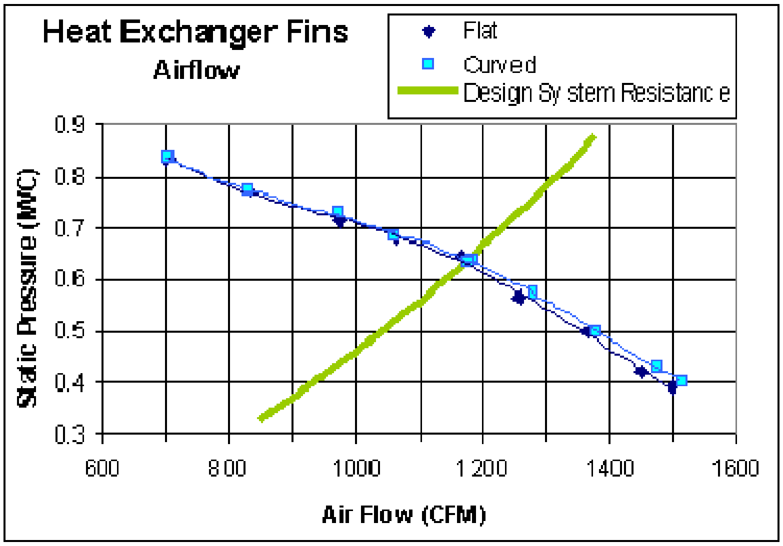

Figure 16: Furnace Heat Exchanger Airflow

Source: Proctor Engineering Group

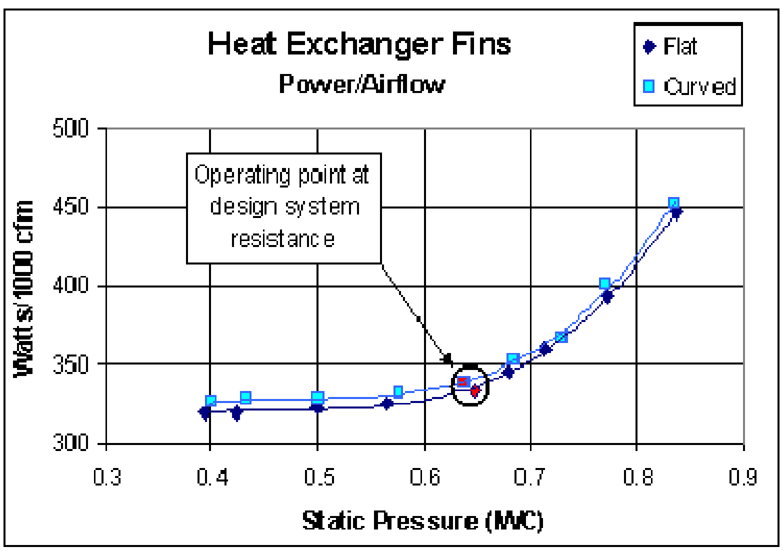

Figure 17: Furnace Heat Exchanger Power Consumption

Source: Proctor Engineering Group

Airflow at design system resistance was 36 cfm higher with the clamshell heat exchanger than with the tube heat exchanger (Figure 16). Power consumption per unit airflow was lower with the clamshell heat exchanger than with the tube heat exchanger (Figure 17).

Modifying the heat exchanger fins did not significantly change air handler performance (Figure 18 and Figure 19). Airflow and power consumption were virtually unchanged. Various curve shapes were tested and none were found to improve performance (Table 13).

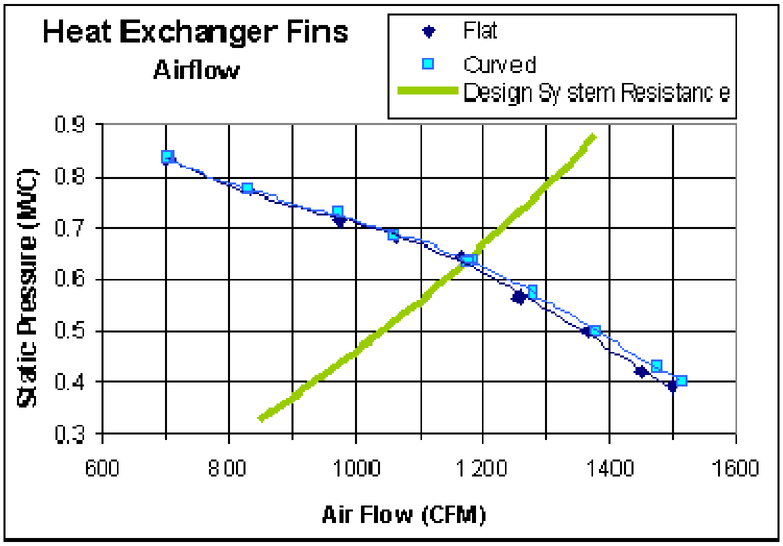

Figure 18: Furnace Heat Exchanger Fins Airflow

Source: Proctor Engineering Group

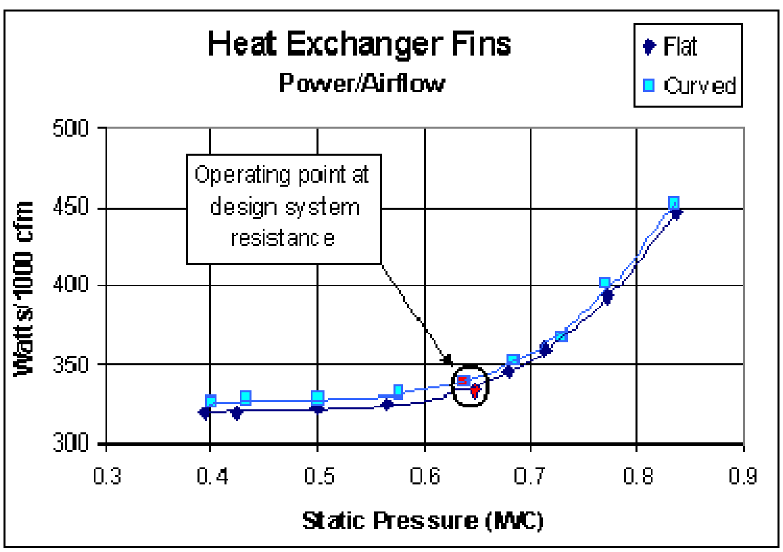

Figure 19: Furnace Heat Exchanger Fins Power Consumption

Source: Proctor Engineering Group

Table 13: Furnace Heat Exchanger and Fins Comparison

	System Resistance	Heat Exchanger: Tube	Heat Exchanger: Clamshell	Heat Exchanger: % Difference	Heat Exchanger Fins: Flat	Heat Exchanger Fins: Curved	Heat Exchanger Fins: % Difference
Airflow (cfm)	Low	1209	1256	3.9	1256	1271	1.2
	Design	1125	1161	3.2	1161	1171	0.9
	High	1029	1060	2.9	1060	1064	0.4
Power (W)	Low	401.0	399.2	-0.5	399.2	423.1	6.0
	Design	381.9	376.9	-1.3	376.9	398.4	5.7
	High	362.7	356.1	-1.8	356.1	374.9	5.3
Efficiency (%)	Low	18.8	20.8	10.6	20.8	20.5	-1.2
	Design	20.3	22.2	9.5	22.2	22.0	-1.1
	High	21.4	23.1	8.2	23.1	22.9	-1.2

Source: Proctor Engineering Group

Efficiency at design system resistance was 9.5% higher with the clamshell furnace heat exchanger compared to the tube heat exchanger. Airflow increased by 36 cfm. The improvement was greater at low system resistance. Replacing the flat heat exchanger fins with curved fins did not significantly change efficiency or airflow.

Fan Inlet Straightening Vanes

Swirling airflow at the fan inlet detracts from efficiency (Figure 20). Inlet swirl is caused by a variety of factors including duct configuration, duct shape, fan position in the cabinet, and airflow obstructions in the cabinet.

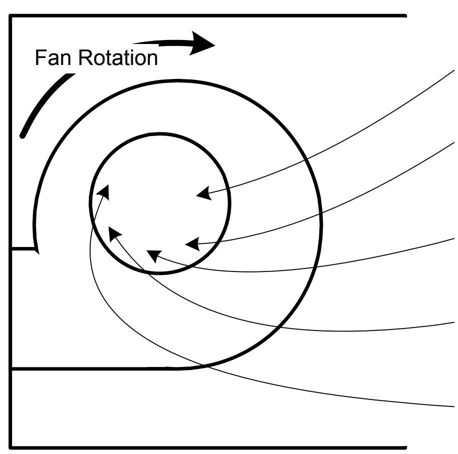

Figure 20: Fan Inlet Swirl

Source: Proctor Engineering Group

Various methods are used to straighten airflow entering the fan, including improved duct design, turning vanes, and fan inlet straightening vanes or splitters. Turning vanes must be designed to match the ductwork, and so are not appropriate for residential systems where the ductwork layout is unknown. Inlet straightening vanes (or splitters) can be implemented as a simple flat vane in line with the fan axis, extending from the cabinet wall to the fan housing. By splitting the intake in half along the fan axis, swirl around the fan is reduced.

Two straightening vane configurations were tested: extending the full length of the cabinet on either side of the fan, and extending only from the fan center to the fan outlet cabinet wall (Figure 21).

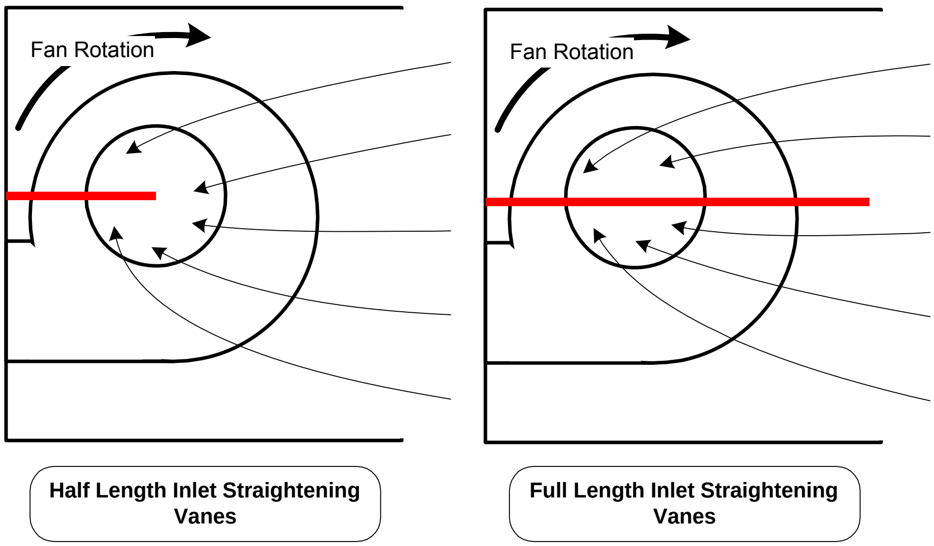

Figure 21: Fan Inlet Straightening Vanes

Source: Proctor Engineering Group

Both types of straightening vanes increased airflow at design system resistance (Figure 22).

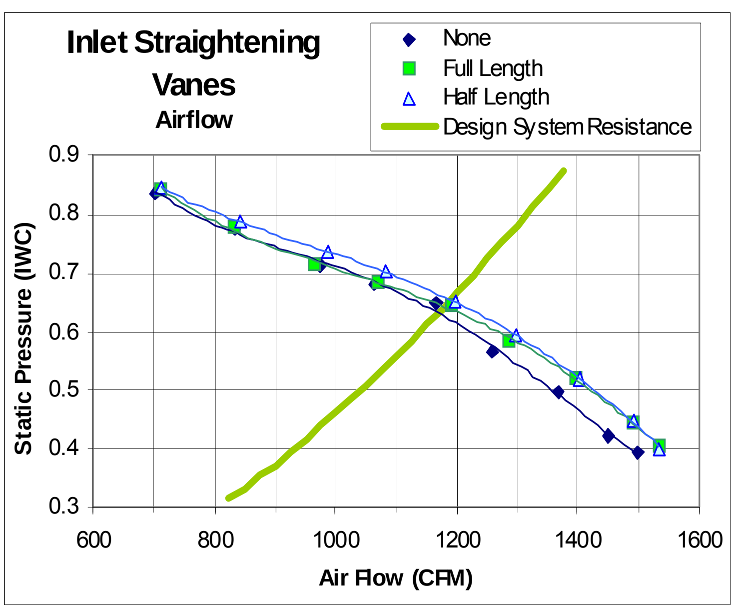

Figure 22: Fan Inlet Straightening Vanes Effect on Airflow

Source: Proctor Engineering Group

Half length vanes increased airflow more than full length vanes. Power consumption per unit airflow increased with both types (Figure 23 and Table 14).

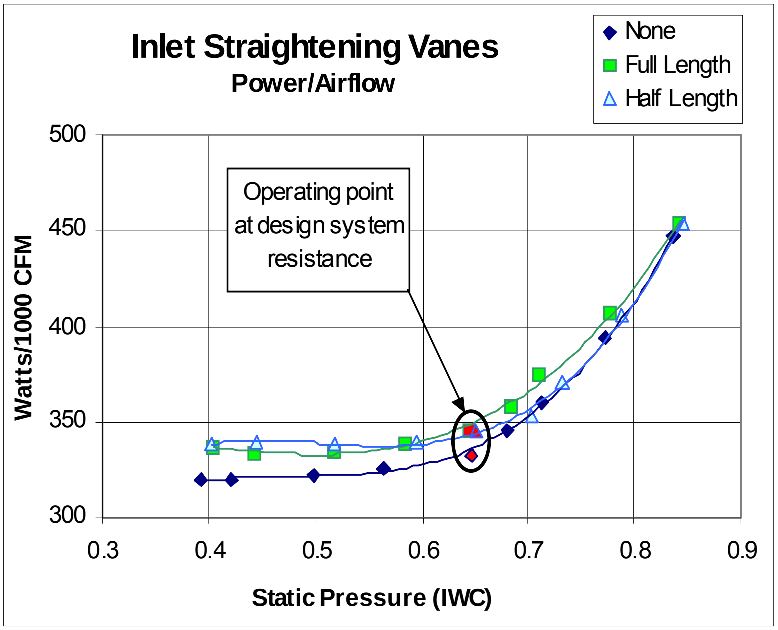

Figure 23: Fan Inlet Straightening Vanes Airflow

Source: Proctor Engineering Group

Table 14: Fan Inlet Straightening Vanes Comparison

	System Resistance	None	Full Length	% Difference	Half Length	% Difference
Airflow (cfm)	Low	1256	1273	1.4	1289	2.7
	Design	1161	1172	1.0	1189	2.5
	High	1060	1067	0.7	1083	2.2
Power (W)	Low	399.2	431.5	8.1	437.5	9.6
	Design	376.9	405.4	7.6	410.0	8.8
	High	356.1	381.5	7.1	384.7	8.0
Efficiency (%)	Low	20.8	20.5	-1.1	20.6	-0.6
	Design	22.2	21.8	-1.9	22.2	-0.2
	High	23.1	22.5	-2.7	23.1	-0.1

Source: Proctor Engineering Group

Inlet straightening vanes increased both airflow and power consumption. As a result, efficiency decreased. Air handler performance was better with straightening vanes extending half of the cabinet length (from fan axis to fan outlet) than with vanes extending the full length. Half length straightening vanes increased airflow by 28 cfm, with little efficiency loss.

The test apparatus return plenum provided a straight airflow path into the air handler cabinet. Straightening vanes may result in a greater improvement in systems where return duct configuration is more conducive to system effect losses (Figure 24). Future work should include return duct configuration as a factor in evaluating the impact of inlet straightening vanes or splitters.

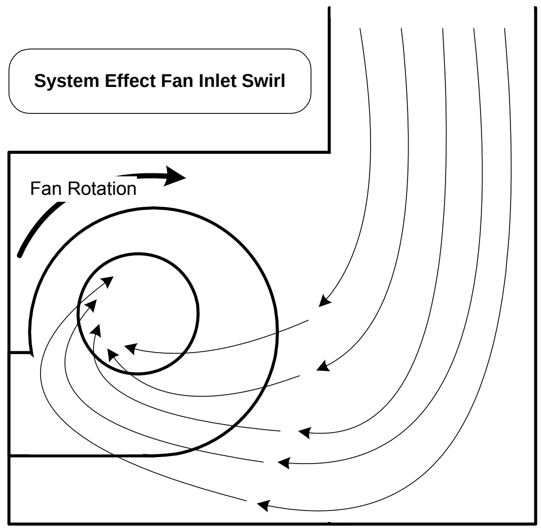

Figure 24: Duct Configuration Causing Fan Inlet Swirl

Source: Proctor Engineering Group

Fan Inlet Clearance

Recent studies have shown that insufficient clearance between the air handler cabinet and fan inlet greatly reduces airflow and efficiency (Walker 2004) (Figure 25). Tests were performed at various fan inlet conditions to determine if efficiency could be improved by offsetting the fan in the cabinet or using a wider cabinet to increase clearance (Table 15, Figure 26 and Figure 27). Testing was not performed with a severe fan inlet restriction, as in Walker 2004.

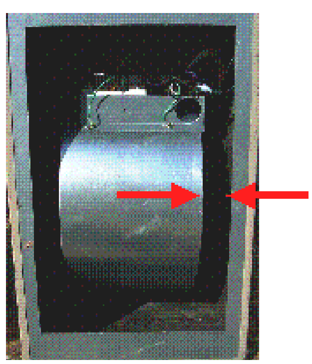

Figure 25: Clearance Between Fan Housing and Cabinet

Source: Proctor Engineering Group

Table 15: Fan Inlet Clearance Tests

Air HandlerCabinet Width (Inches)	Fan Inlet Clearance (Inches)	
	Motor Side	Opposite Motor
20	2	3.5
20	2.75	2.75
20	3.5	2
22.5	4	4

Source: Proctor Engineering Group

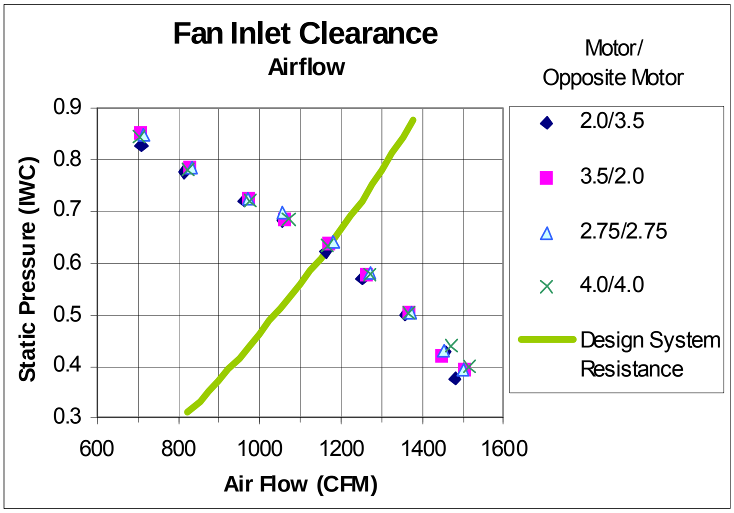

Figure 26: Fan Inlet Clearance Airflow

Source: Proctor Engineering Group

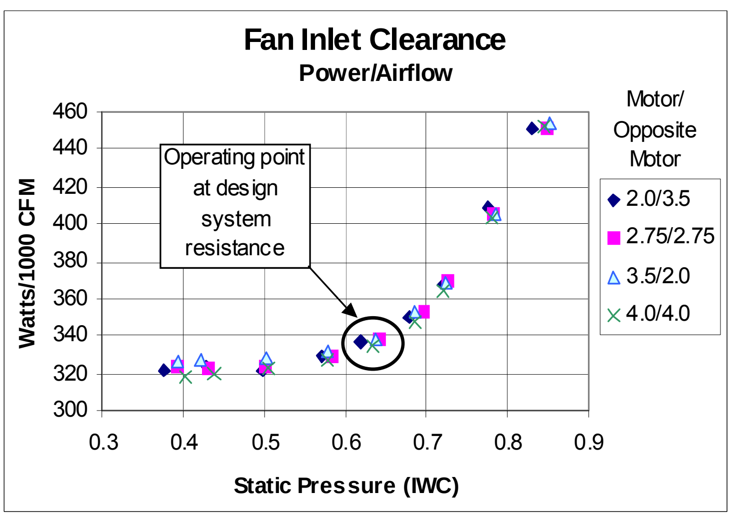

Figure 27: Fan Inlet Clearance Power Consumption

Source: Proctor Engineering Group

Modifying the clearance between the fan inlet and the air handler cabinet had very little effect on system performance. Airflow, power consumption, and efficiency were unchanged. Test results indicated that the fan used in the HDAC air handler had sufficient inlet clearance in a 20″ cabinet. Increased cabinet width did not improve performance.

Fan Type

A tube axial fan was tested for comparison to the standard forward curved centrifugal fan. The axial fan was equipped with variable pitch blades to adjust airflow at varying system resistance (Figure 28).

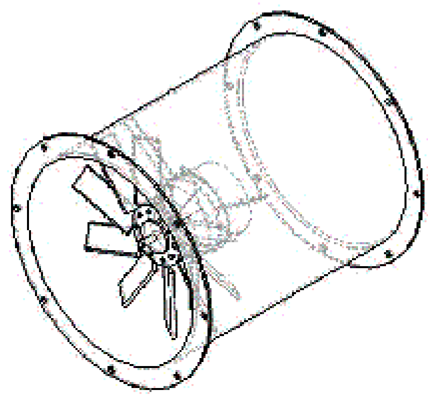

Figure 28: Tube Axial Fan

Source: Proctor Engineering Group

Tests were conducted at 18° and 25° blade angles. Motor mount and shaft diameter were incompatible with the centrifugal fan, so the axial fan was tested with a 1/2 Hp PSC motor instead of the 1/3 Hp PSC motor used in other tests.

The axial fan provided less airflow at design system resistance than the centrifugal fan. Increasing the blade angle to 25° improved airflow, but the fan consumed more power per unit airflow (Figure 29 and Figure 30).

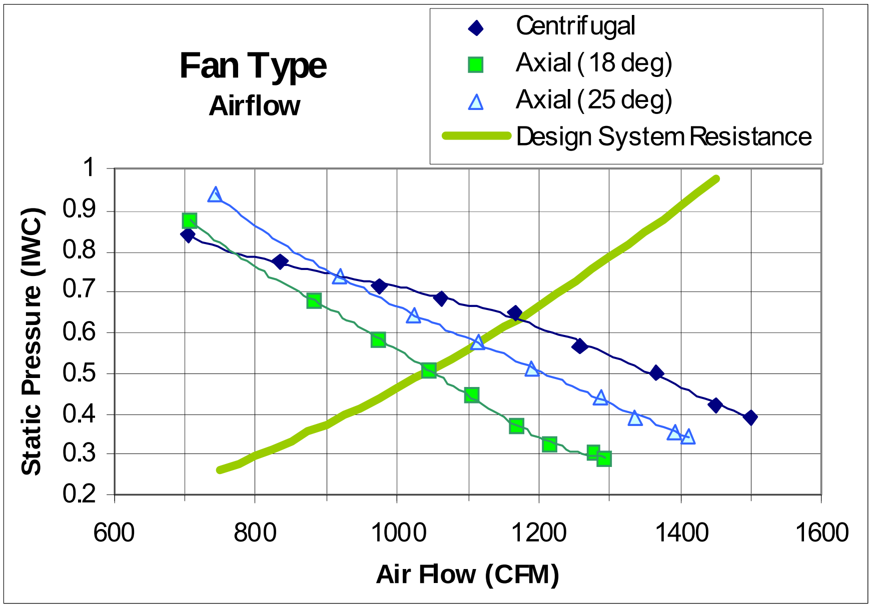

Figure 29: Axial Fan Airflow

Source: Proctor Engineering Group

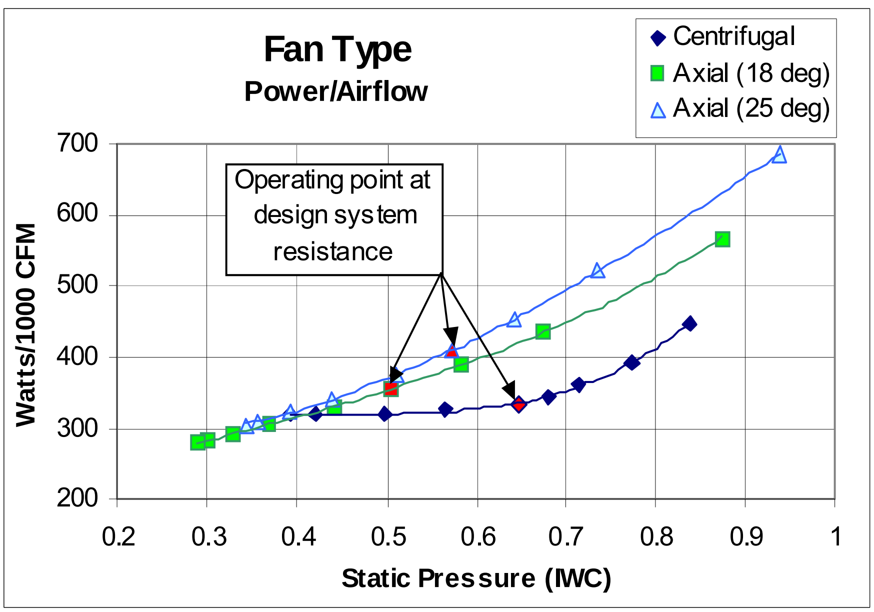

Figure 30: Axial Fan Power Consumption

Source: Proctor Engineering Group

Efficiency cannot be directly compared to the centrifugal fan since a different motor was used. The combination of axial fan and motor was 24% less efficient than the centrifugal fan/motor at design system resistance.

The axial fan was very noisy compared to the centrifugal fan. The noise level was unacceptable for residential applications.

Return Plenum Orientation

Various return duct/plenum configurations were tested, as shown in Figure 31.

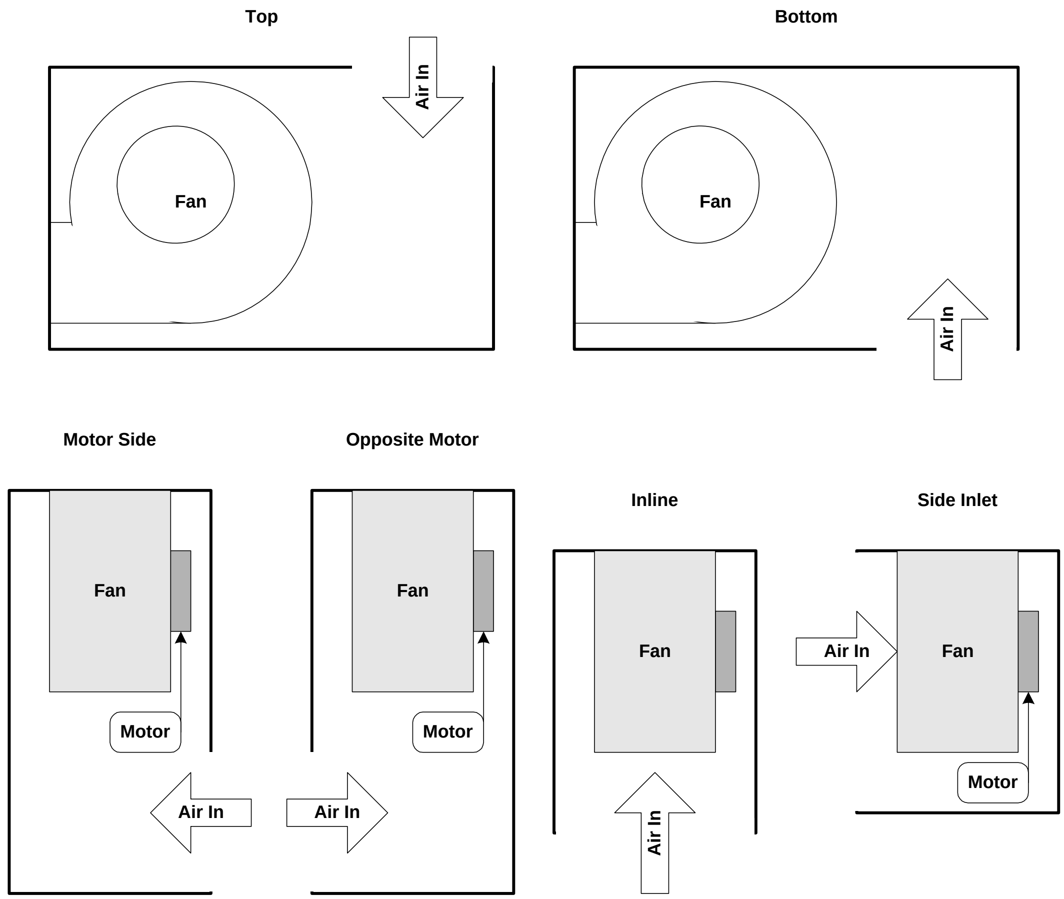

Figure 31: Return Duct/Plenum Orientation

Source: Proctor Engineering Group

The air handler performed better with the return duct entering perpendicular to the fan's axis of rotation than parallel to the fan's axis. Average air handler efficiency at design system resistance was 15% higher in the 'Top' and 'Bottom' configuration compared to the 'Opposite Motor' and 'Motor Side' configuration. Airflow was 3.6% higher. These tests were performed with a different motor than previous tests. The motor ran at a higher rpm, so airflow was higher. Average airflow at design system resistance was 1300 cfm/ton.

Performance was better with the return plenum in line with the air handler than perpendicular to the air handler. Efficiency at design system resistance was 3.6% higher in the 'Inline' position compared to the 'Side Inlet' position.

Fan Motor

Typical permanent split capacitor (PSC) motors for residential air handlers are designed to operate at 1100 rpm on high speed. Most are equipped with several speed selections for airflow adjustment. However, PSC motors consume nearly as much power at the low speed settings as at the high-speed setting. Efficiency is greatly reduced when these motors are not run at their rated rpm.

The HDAC air handler delivers appropriate airflow for a 3-ton air conditioner between 800 and 900 rpm at design system resistance. To avoid the efficiency loss associated with running a PSC motor at reduced speed, an 825 rpm PSC motor was acquired for testing. The motor was 1/3 Hp and required approximately half as much power as a standard efficiency 1/2 Hp, 1100 rpm PSC motor running at reduced speed.

An electronically commutated motor (ECM) was also tested. ECM is a brushless DC motor that can be configured to maintain constant torque. Motor speed is modulated to maintain torque, delivering steady airflow over a range of static pressures. The ECM motor was equipped with a controller for manual speed/torque adjustment.

The PSC motor was rated at 1/3 Hp, 825 rpm. The ECM motor was rated at 1/2 Hp and was variable speed. Table 16 compares air handler performance with each motor at 860 rpm, 1150 cfm airflow, and 0.63" water column static pressure. Tested at similar conditions, the ECM motor increased air handler efficiency by 17% compared to the smaller PSC motor.

Table 16: Split System Air Handler Fan Motor

Motor	Power (W)	Watts/1000 cfm	Air Handler Efficiency (%)
1/3 Hp PSC	377	325	22.5
1/2 Hp ECM	323	281	26.4

Source: Proctor Engineering Group

A different type of brushless DC motor was also tested. The motor was rated at 1/3 Hp. At low static pressures, it performed similarly to a 1/3 Hp ECM. Unlike ECM, the motor was not designed to maintain constant torque, and airflow dropped rapidly with increasing static pressure. Airflow at design system resistance was too low to fully test this motor.

Final Design

Test results were applied to the DOE ORNL model at hot/dry conditions (80 °F return dry bulb, 63 °F return wet bulb, 115 °F outside ambient). Components that provided the highest predicted system EER and sensible EER were selected for the final design. The following components were selected:

- Plastic fan housing – this housing increased EER by 2.7% and sensible EER by 7% compared to the standard sheet metal housing.
- Clamshell style heat exchanger – this heat exchanger increased EER by 0.7% and sensible EER by 1.5% compared to the tube heat exchanger.
- 1/2 Hp ECM motor – this motor increased EER by 2% and sensible EER by 1.8% compared to a 1/3 Hp, 825 rpm PSC motor. It should be noted that the 825 rpm PSC motor is significantly more efficient than a standard 1100 rpm PSC running at a lower

speed setting. The ECM motor improved EER by 15% over a standard efficiency PSC motor running at reduced speed.

Modified furnace heat exchanger fins and inlet turning vanes did not affect system efficiency and were not included in the final design.

Air handler cabinet width was set at 20″ since increased cabinet width did not improve performance.

Total improvement over the baseline 3 ton split air conditioning unit, as predicted by the DOE ORNL model at hot/dry conditions, is 9.8% higher EER and 11.6% higher sensible EER.

3.1.2. 5-Ton Package Indoor Section Airflow Test Results

The package proof-of concept HDAC unit began as a Carrier Infinity 13™ 5-ton package unit. The indoor section of the unit was substantially modified during the airflow and development testing. The original configuration of the indoor section is shown in Figure 32.

Figure 32: Package Unit Initial Indoor Section Configuration

Source: Proctor Engineering Group

Evaporator Coil

The original tube and fin evaporator coil was compared to a microchannel evaporator coil of the same face area and configuration. Figure 33 shows the configuration of these coils.

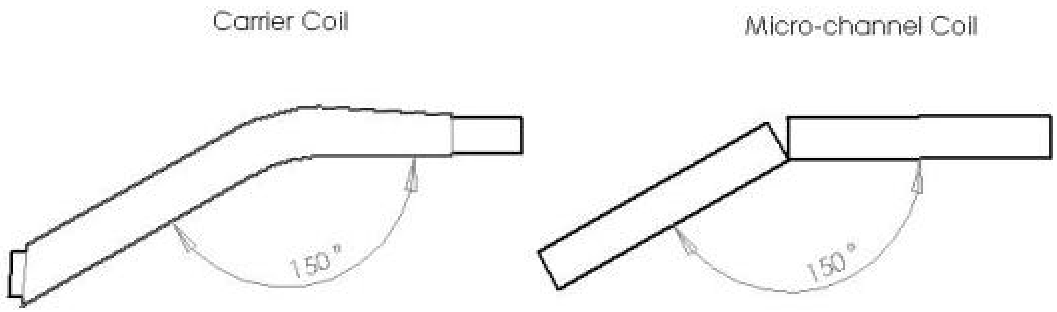

Figure 33: Package Unit Evaporator Coil Configuration

Source: Proctor Engineering Group

The microchannel coil achieved a higher flow rate by 130 cfm at design system resistance. The indoor section consumed 120 Watts less power at 1750 cfm with the micro-channel coil.

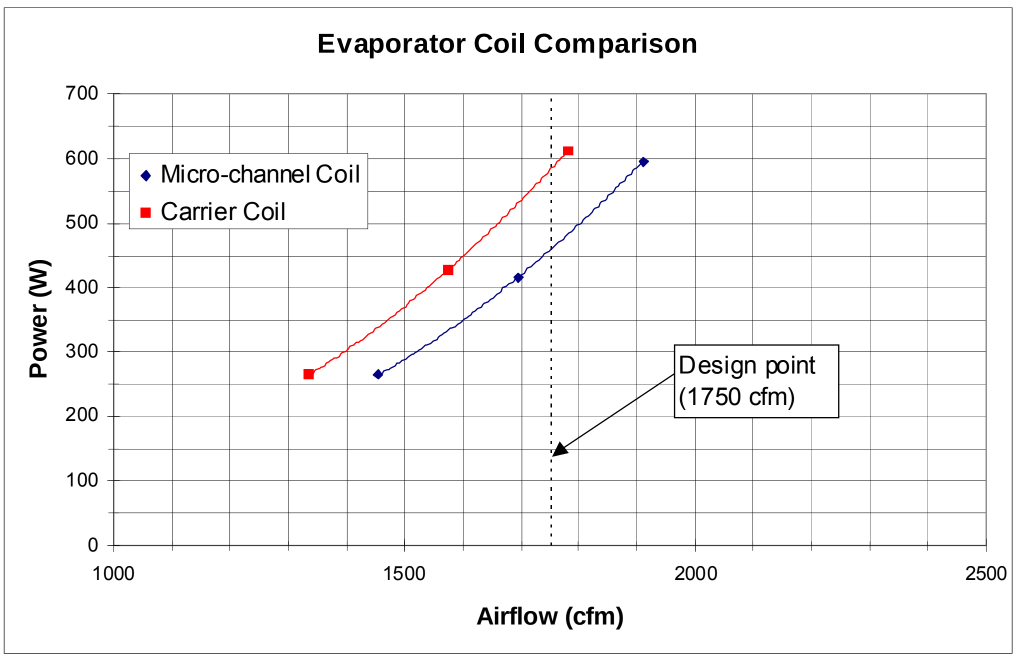

Figure 34. Evaporator Coil Effect on Power Consumption

Source: Proctor Engineering Group

Furnace Heat Exchangers

A clamshell heat exchanger of the same heat input rating was compared to the original tube heat exchanger. The clamshell was tested in different orientations to see if it would provide less resistance to airflow in any possible orientation. These tests could not determine the suitability of the clamshell heat exchanger for use in the unit since additional tests in the heating mode would be needed to determine that suitability. These were only to determine if further investigation was warranted for airflow considerations within the current project.

In the best orientation of the clamshell heat exchanger, the airflow was 200 CFM lower than the original tube heat exchanger.

Evaporator Fan Modifications

Three types of fans were tested: a standard forward-curved centrifugal ("squirrel cage") fan, a vane-axial fan and a backward-inclined centrifugal fan. The standard forward curved fan was chosen for the final configuration, but the backward curved fan showed some promise and deserves further testing.

Six motors were tested using the forward-curved fan. These motors included Permanent Split Capacitor (PSC) motors, DC motors, and Electrically Commutated Motors (ECMs). The results of these tests are shown in Figure 35. An ECM motor was selected for the final configuration based on watt draw and true variable speed capability.

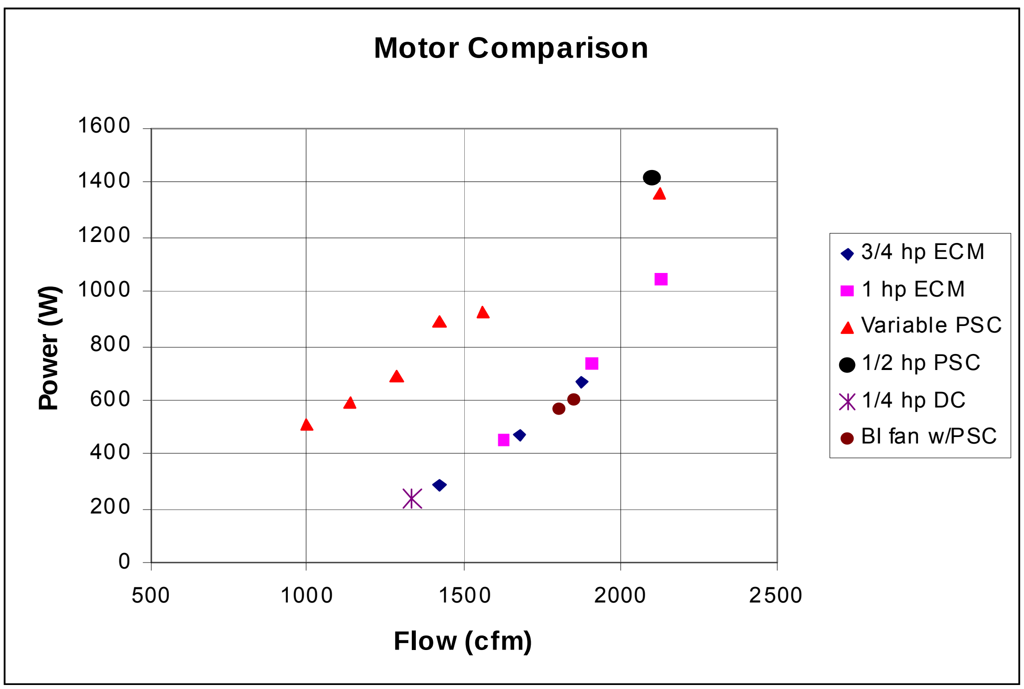

Figure 35. Evaporator Fan Motor Comparison

Source: Proctor Engineering Group

Five modifications to the evaporator fan were tested. These included:

- Flipping the entire fan assembly 180º around the discharge axis.

- Moving the fan assembly further away from the furnace heat exchanger[4].
- Opening the discharge angle of the fan housing and widening the opening into the furnace heat exchanger plenum to obtain more direct airflow to the furnace heat exchanger.
- Removing the flow diverter at the outlet of the fan housing[5].
- Smoothed the corners of the fan housing.

These modifications are shown in Figures 36 and 37.

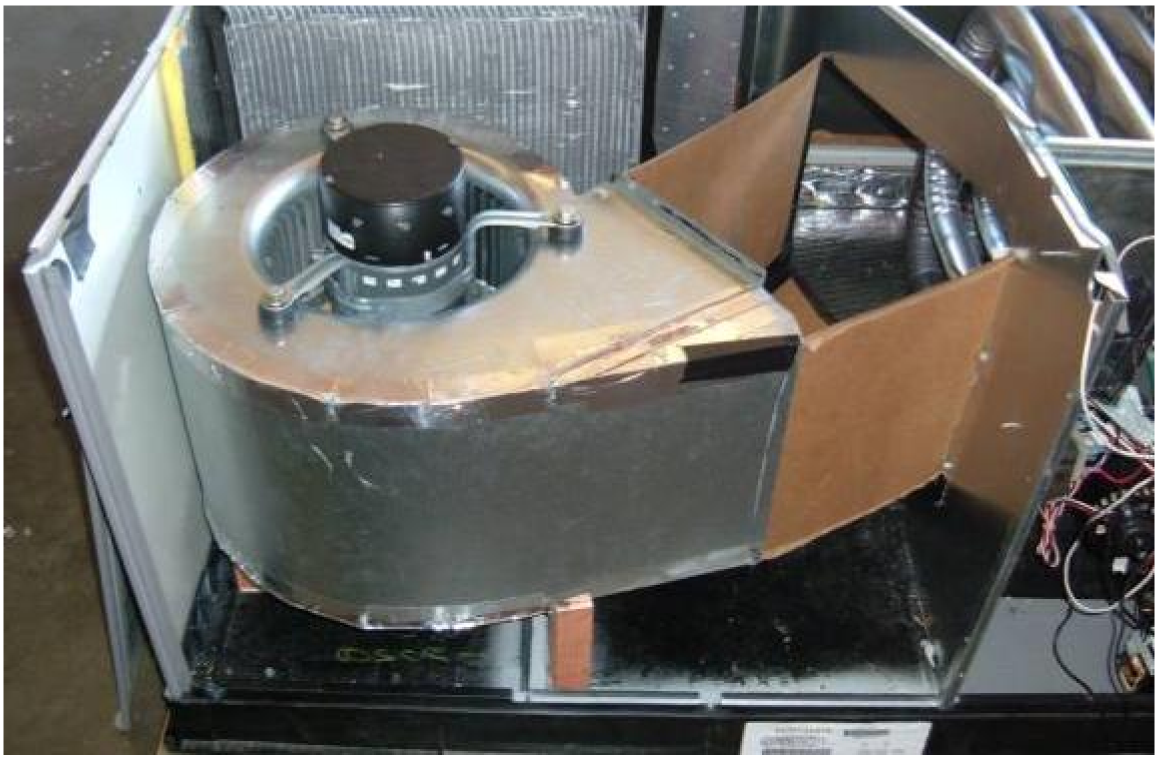

Figure 36. Evaporator Fan Assembly Revisions

Source: Proctor Engineering Group

[4] This was made possible by the relocation of the compressor to the outdoor section of the unit.

[5] Each change was tested separately to ensure that each modification was advantageous.

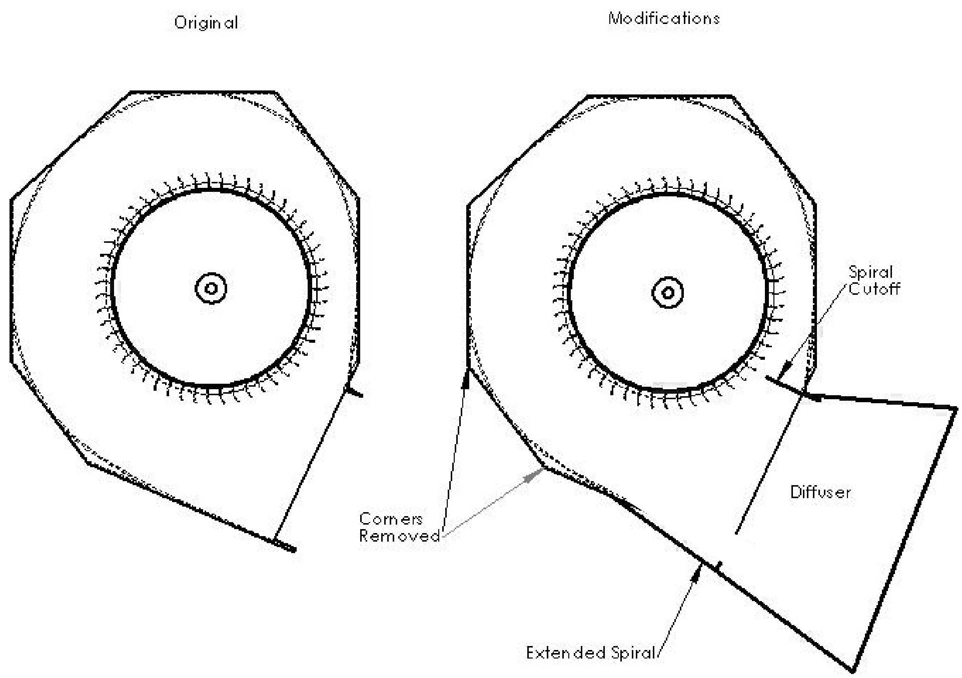

Figure 37. Evaporator Fan Housing Revisions

Source: Proctor Engineering Group

Evaporator Airflow Distribution Modifications

Even airflow distribution through the evaporator coil in package units is difficult to achieve due to the geometry and space restrictions. Preliminary testing showed that most of the air was flowing through the half of the coil directly in line with the cabinet return air inlet. Multiple methods were tested to achieve more even airflow distribution through the evaporator coil. These methods included: turning vanes, coil realignment, airflow dividers, and a larger return plenum opening.

Increasing the return plenum opening and realigning the coil so that the approach angle of the air was closer to perpendicular improved the airflow distribution significantly. Figure 38 displays the 73% increase in the return inlet area.

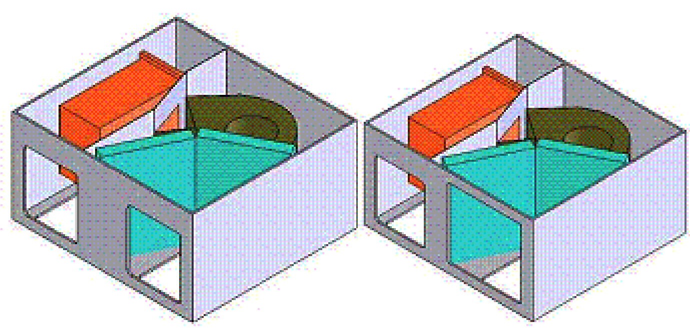

Figure 38. Return Plenum Inlet Revision (original on left, revision on right)

Source: Proctor Engineering Group

Configurations

Over 130 tests were run on 20 component configurations to determine the best placement of components. These tests improved the maximum airflow with a ½ Hp PSC motor from 1785 cfm to 2100 cfm, an improvement of 17.5%. The changes interact so the exact result of each change is not independent of the configuration of the other components. Table 17 estimates the flow effect of individual improvements for the ½ Hp PSC motor at full speed.

Table 17. Estimated Indoor Section Airflow Changes from Modifications

Change	Improvement
Removed Corners on Blower housing	15 cfm
Micro-channel evaporator coil	100 cfm
Expanded blower housing	150 cfm
Flipped Fan and rotation	50 cfm
Expanded return inlet	75 cfm

Source: Proctor Engineering Group

Final Indoor Section Airflow Design

The airflow improvements for the indoor section are compared to the original configuration in Figure 39. The results of these changes as modeled with the ORNL modeling software are shown in Table 18.

Table 18. Effect of Indoor Section Airflow Modifications

Result	Baseline	HDAC
Flow	1730 cfm	1910 cfm
Fan Power	694 Watts	596 Watts
Watts per 1000 cfm	401	312
Airflow Efficiency	0.13	0.23
EER	6.93	7.15
Sensible Heat Ratio	0.81	0.83
Sensible EER	5.62	5.97

Source: Proctor Engineering Group

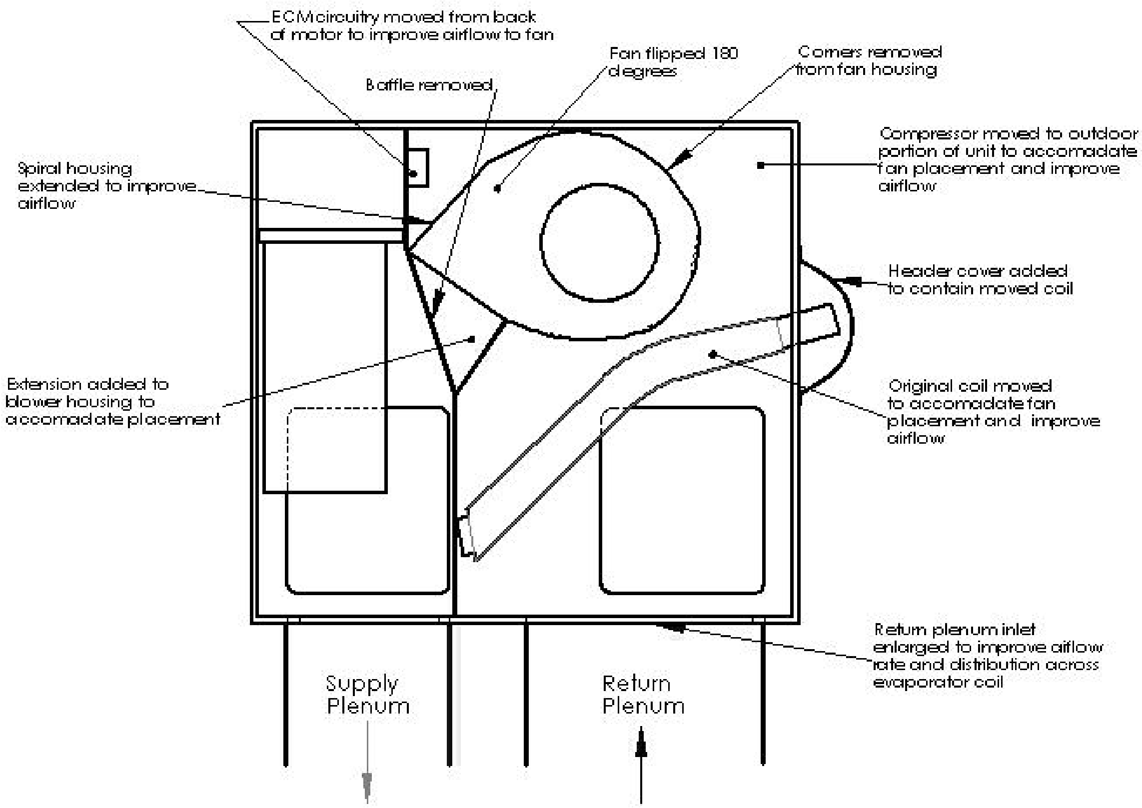

Figure 39. Final Indoor Section Configuration with Revisions Noted

Source: Proctor Engineering Group

3.1.3. Condenser Airflow Test Results

The split unit and the package unit underwent similar testing and produced similar results for the condenser sections.

Compressor Location

The split outdoor unit had a U shaped condenser coil as shown in Figure 40.

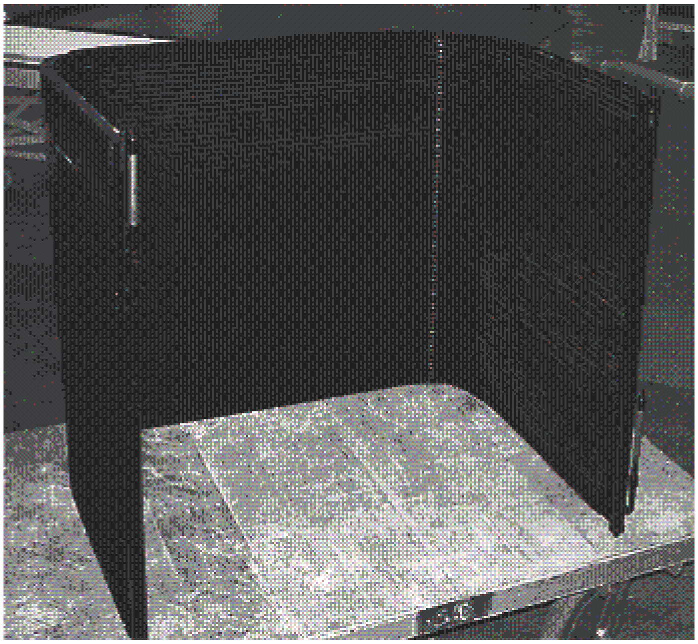

Figure 40. Split Unit Condenser Coil

Source: Proctor Engineering Group

The original design called for the compressor to be housed in a chamber at the "back" of the outdoor unit similar to some production models. This configuration was simulated as shown in Figure 41.

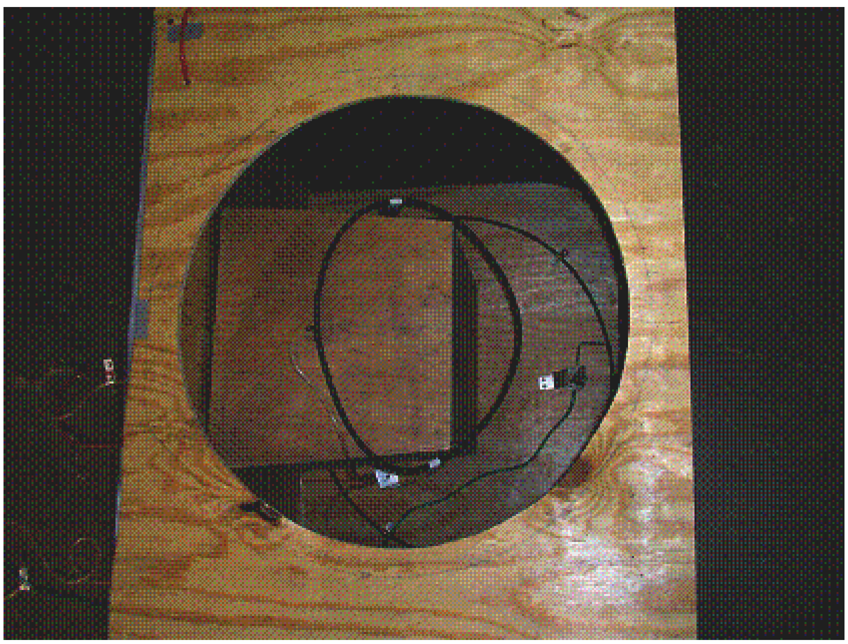

Figure 41. Split System Proposed Compressor Location (within the box at the left)

Source: Proctor Engineering Group

Testing compared the proposed location to alternative locations, including centered directly below the vertical axis of the fan. The centered location was shown to have better airflow and lower watt draw.

The package outdoor section had two L-shaped condenser coils as shown in Figure 42. The compressor location was moved from a cubical in the lower section to the center of the outdoor section in line with the vertical axis of the fan.

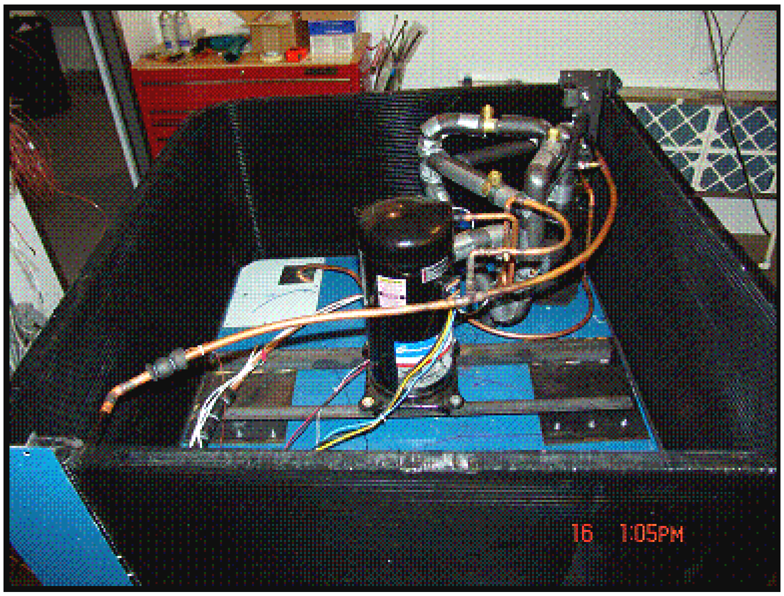

Figure 42. Package System Condenser Coils and Compressor Location

Source: Proctor Engineering Group

Condenser Fan Design

Eight fan blades were tested, including five 2-blade fans and three 3-blade fans with blade pitches varying from 18-34°. In addition, the tests included two patented aerodynamically shaped fans designed by Florida Solar Energy Center (Parker 2005). The basic fan types are shown in Figure 43.

Figure 43. Condenser Fan Types

Source: Proctor Engineering Group

Table 19 shows the results of the tests on the two and three blade stamped metal 26" diameter condenser fans and three different motors on the outdoor section of the package HDAC.

Table 19. Condenser Fan and Motor Performance

Motor	Blades	Pitch (degrees)	Flow (cfm)	Power (W)	RPM	W/1000 cfm
Baseline 21" diameter fan and 1/4 Hp PSC motor (no diffuser)						69.9
1/3 Hp ECM	2	22	5230	122	836	23.3
1/8 Hp PSC	2	22	5116	188	825	36.7
1 Hp ECM	2	34	6817	267	835	39.2
1 Hp ECM	2	30	6685	235	838	35.1
1 Hp ECM	2	20	5076	114	834	22.5
1 Hp ECM	2	18	4972	109	840	21.9
1 Hp ECM	3	30	6933	247	830	35.7
1 Hp ECM	3	32	7338	297	834	40.4

Source: Proctor Engineering Group

The reduction in fan watts per 1000 cfm made it possible to improve condenser fan airflow without a large fan watt draw penalty. The 2-blade fan with a pitch angle of 22° was chosen based on its ability to provide sufficient flow with a 1/8 Hp PSC motor. This configuration was too noisy prior to the addition of the diffuser described below. The laboratory tests were run with the 1/3 Hp ECM so that variations of condenser airflow could be tested. Due to the higher cost of the ECM, it may be desirable in some cases to use the PSC.

Table 20 details the differences between the FSEC fan assemblies and the HDAC assembly

Table 20. Detail on FSEC and HDAC Condenser Fan Assemblies

	FSEC	HDAC
Fans	Shaped molded 4 blade and5 blade fans	Stamped Lau2 blade fan
Fan diameter	28 inch	26 inch
Fan blade pitch	Shaped – multiple pitches	22 degrees
Diffuser	12degree12.5 inch	10 degree23 inch
Inlet	2.5 inch high Trane production shaped inlet	HDAC 8 inch high shaped inlet

Source: Proctor Engineering Group

Figure 44 compares the two FSEC fan assemblies with the HDAC fan assembly. The fan assemblies were roughly comparable over their common range of airflow. The FSEC assemblies were not able to provide the flow that was achieved by the HDAC assembly. The FSEC assemblies and the HDAC assembly were designed for different applications (condenser pressure drop, lower fan speed, etc.) The HDAC fan assembly has a taller diffuser and a taller inlet. It is possible with equal diffuser height and the resultant additional pressure recovery the FSEC fan could outperform the HDAC fan.

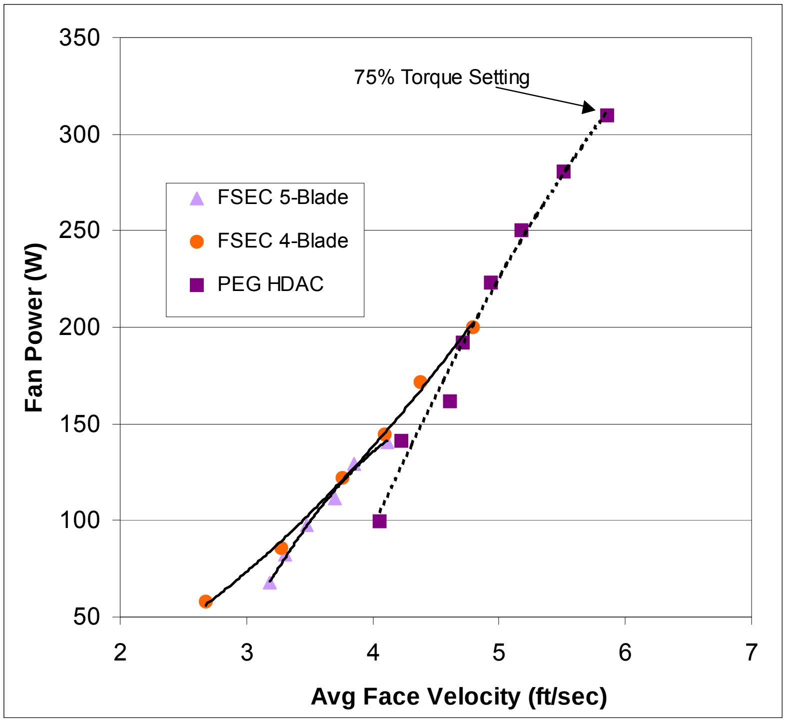

Figure 44. Airflow vs. Power for FSEC and HDAC Condenser Fan Assemblies

Source: Proctor Engineering Group

Condenser Fan Location, Inlet, and Outlet Conditions

The primary tests of the condenser fan location (height above the top of the condenser coil), inlet conditions, and outlet conditions were run in the split system outdoor unit.

It was determined that raising the fan above the top of the condenser coil, adding a shaped inlet, and adding a discharge diffuser improved the watt draw to airflow ratio and reduced the noise of the fan to acceptable levels.

The shaped inlet 8" tall proved to be a good compromise. This inlet is shown in Figure 45.

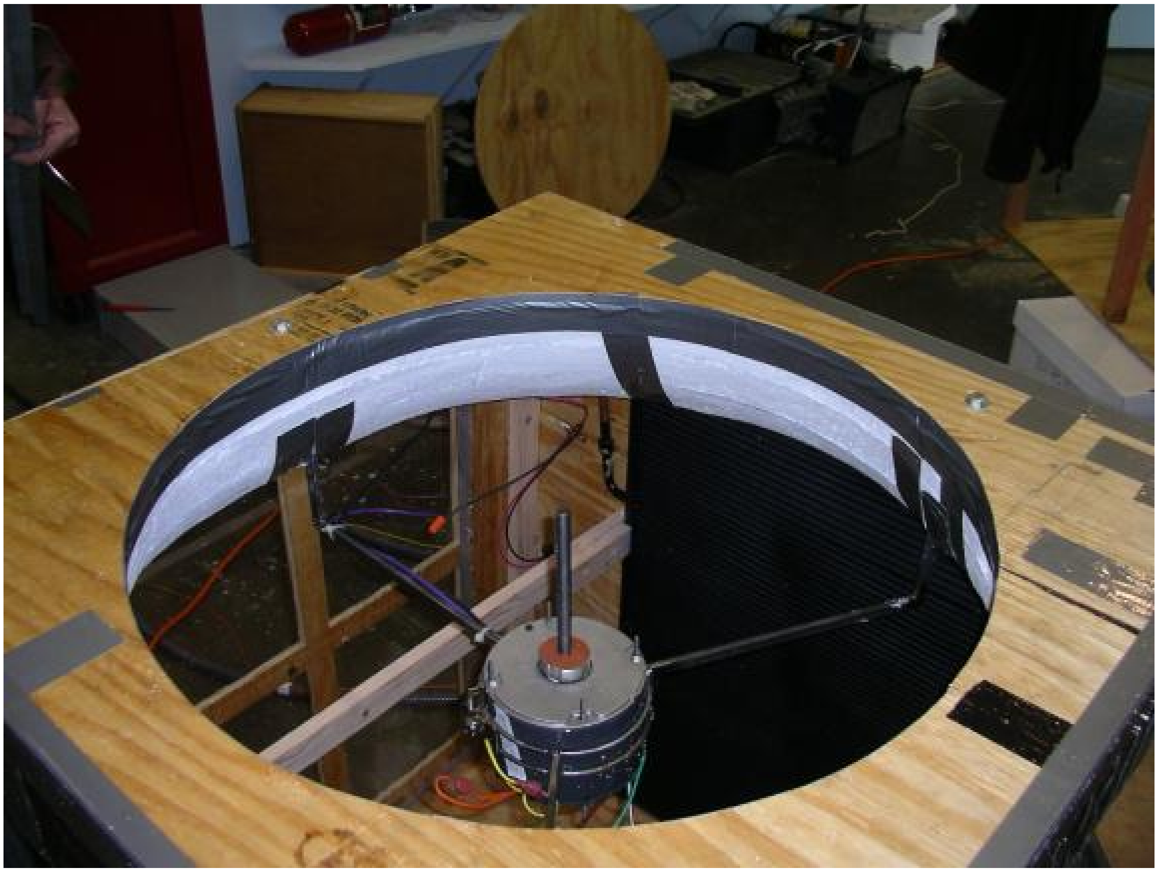

Figure 45. Shaped Inlet for Condenser Fan

Source: Proctor Engineering Group

A variety of diffuser heights and angles were tested. The results are shown in Table 21.

Table 21. Condenser Fan Diffuser Test Results

Diffuser Angle	Length	Flow (cfm)	Power (W)	Watts/1000cfm
none	na	4031	192	47.63
7	36	4604	176	38.23
7	24	4444	178	40.05
7	18	4443	179	40.29
10	23	4574	178	38.92
10	18	4529	178	39.30
10	12	4451	181	40.67

Source: Proctor Engineering Group

The diffuser used in the laboratory tests was 23" tall with a 10º expansion angle as shown in Figure 46.

Figure 46. Diffuser for Condenser Fan

Source: Proctor Engineering Group

3.1.4. Development Tests

The development testing produced the following changes in the units delivered for laboratory testing:

- Both Units
- Refrigerant metering control was changed from subcooling with a flooded evaporator to constant subcooling with low superheat.
- Suction/liquid line heat exchanger was plumbed out of the system.
- Split Unit
- Microchannel evaporator coil was tested against a 9mm coil. The microchannel coil was used because it produced higher efficiencies. The 9mm coil performance was degraded due to poor refrigerant distribution between the circuits, a problem that could potentially be solved.
- Package Unit
- The standard fin/tube evaporator coil was tested against the microchannel coil. The standard coil was used because it produced higher efficiencies. The microchannel coil performance was degraded due to poor refrigerant distribution associated with a major and unanticipated difference in airflow through sections of the evaporator coil, a problem that could be solved by altering the circuit configuration.

3.1.5. Unit Tests 5-Ton Package (SCE RTTC)

Charge Optimization

This series of tests was conducted to determine the refrigerant charge for optimal performance. Optimum performance was gauged by analyzing the EER as refrigerant was gradually removed from the unit. Figure 47 shows the EER for several charge levels with 115°F ambient air and indoor conditions of 80°F dry bulb and 63°F wet bulb. The optimum charge was determined to be 9 lbs 8 ounces.

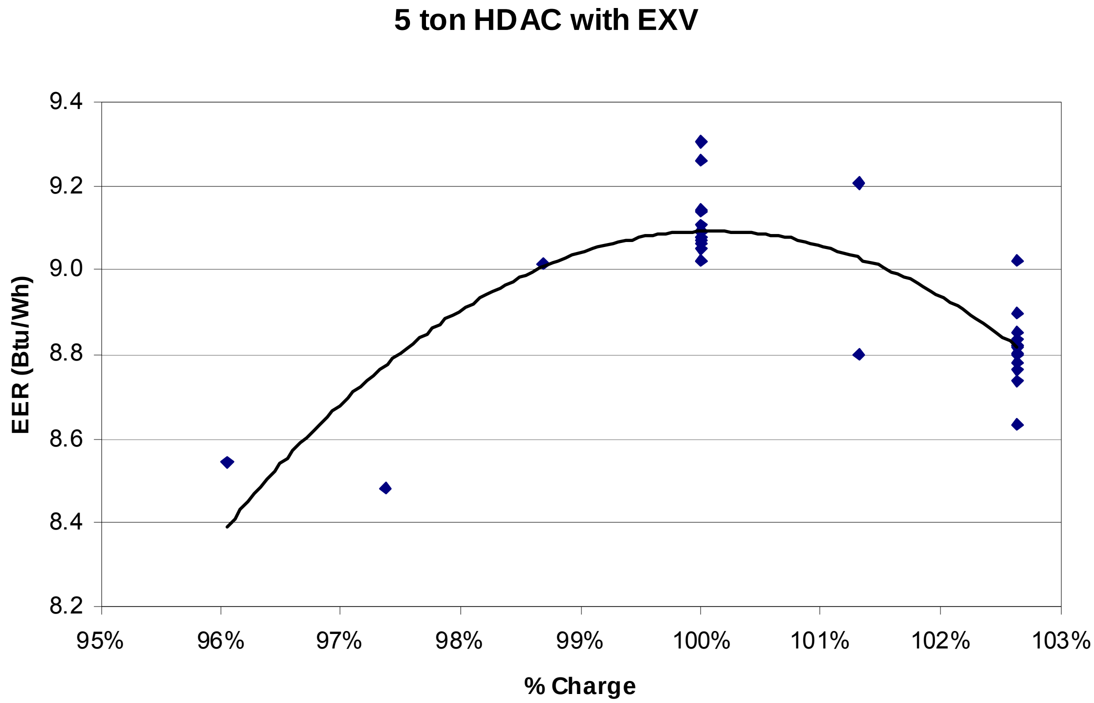

Figure 47. Charge Optimization for Package HDAC

Source: Southern California Edison

Airflow Optimization

This series of tests was conducted to determine the airflow for optimum performance. Since the metric of interest in this project is the net sensible EER at hot dry conditions (PEERs) this metric was used to determine the optimum airflow. Higher airflow rates result in higher sensible cooling, however the watt draw of the evaporator fan increases very rapidly since the static pressure increases as the square of the flow.

Performance

The performance of the package unit exceeded expectations as shown in Table 22.

Table 22. Performance of the Package HDAC

	Outdoor DB [°F]	Indoor DB [°F]	Indoor WB [°F]	EER [btu/Whr]	Sensible EER [btu/Whr]	SHR [btu/btu]	Sensible Capacity [btu]
HDAC Design	115	80	63	9.30	8.60	0.92	50,251
Hot “Normal”	115	80	67	9.96	7.08	0.71	41,421
HOT Test	120	80	67		6.68		
ARI Test A	95	80	67	14.24	10.19	0.72	49,330
ARI Test B	82	80	67	17.45	11.99	0.69	50,827
ARI Test C	82	80	59	15.29	15.11	0.99	62,112
ARI SEER				SEER 16.75			

Source: Southern California Edison

The performance of the HDAC package unit is compared to baseline units in Section 3.3.

Latent Recovery

At peak conditions with moderately dry indoor air (Hot "Normal") above, the air conditioner stores some of its cooling capacity in the form of water collected on the evaporator coil. This is shown in Table 23 as the sensible heat ratio (SHR) less than 1 in the Hot "Normal" test. In that test 29% (1-0.71) of the cooling energy of the unit was stored as moisture removal. This stored capacity will be lost down the condensate drain unless it is recovered at the end of the compressor cycle. By running the evaporator fan after the compressor is off, the stored cooling capacity can be recovered from the water evaporating off the coil. This process is latent recovery.

The amount of moisture that is converted to sensible cooling is dependent on the airflow and the length of time the fan runs after the compressor is off (the tail). The efficacy of the tail depends on the watt draw of the evaporator fan motor.

Latent recovery and its counterpart latent enhancement have been studied in laboratory tests the field monitoring (Faramarzi & Mitchell 2006; Proctor and Brezner 2006; Shirey, Henderson & Raustad 2006).

Table 23. Measured Sensible EER and Savings from a 5 minute Time Delay

Compressor Cycle	End of Compressor Cycle Sensible EER	Sensible EER after 5 minute fan delay	Savings
5 minutes	6.0	8.5	29%
10 minutes	6.3	8.0	21%
15 minutes	6.6	7.75	15%

Source: Southern California Edison

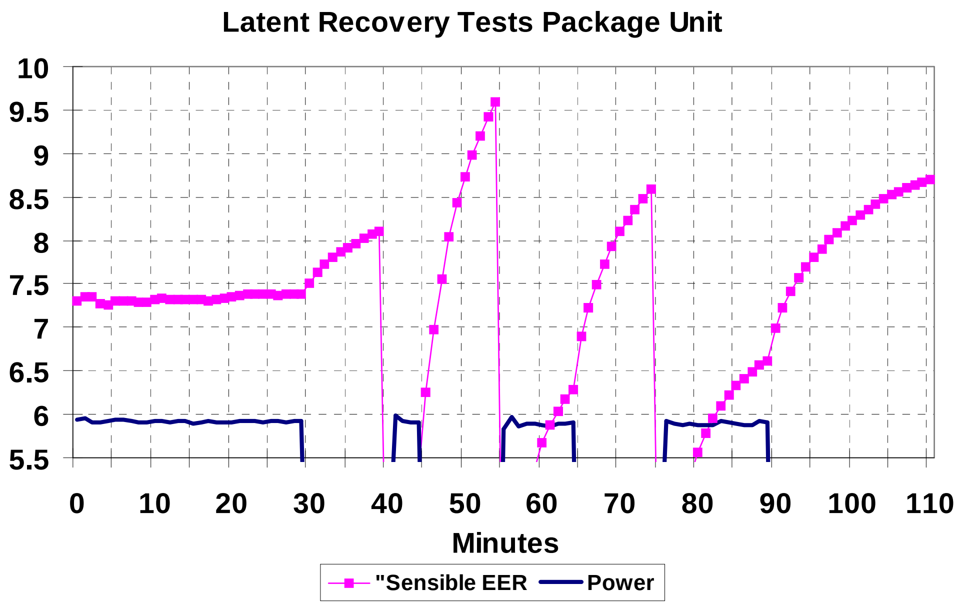

Figure 48. Package Unit Latent Recovery Laboratory Tests

Source: Southern California Edison

Figure 48 Notes:

- As elapse time proceeds (X axis) the unit is running at steady state with a sensible EER of 7.4. At 30 minutes the compressor shuts off (the watt draw drops from 5.9 kW to the indoor fan only power). From the 30 minute mark, the indoor fan continues to run for 10 minutes evaporating water off the coil and delivering sensible capacity. At the end of the "tail" the overall sensible EER has risen to 8.1.

- At the 40 minute mark, the compressor comes on and runs for 5 minutes. At the end of that 5 minute compressor cycle, the cumulative sensible EER is less than 6. From the 45 minute mark, the indoor fan continues to run for 10 minutes delivering sensible capacity. At the end of the "tail" the over sensible EER has risen to 9.6.

- At the 55 minute mark, the compressor comes on and runs for 10 minutes. At the end of that 10 minute compressor cycle, the cumulative sensible EER is delivering sensible capacity. At the end of the "tail" the overall sensible EER has risen to 8.6. It is evident that the longer compressor on cycle would require a longer "tail" to approach the efficiency achieved by the 5 minute compressor on cycle within a 10 minute "tail."

- At the 75 minute mark, the compressor comes on and runs for 15 minutes. At the end of that 15 minute compressor cycle, the cumulative sensible EER is 6.6. From the 90 minute mark, the indoor fan continues to run for 20 minutes delivering sensible capacity. At the end of the "tail," the overall sensible EER has risen 8.7.

3.1.6. Unit Tests 3-Ton Split (PG&E)

The primary test results are summarized in this section.

The results are presented in the following order:

- Optimized Unit Performance
- Refrigerant Charge and Subcooling Optimization
- Evaporator Airflow Optimization
- Airflow Effects Under ARI Test Conditions
- SEER
- Condenser Airflow Testing
- Optimized Unit at Hot and ARI Conditions

The performance of the unit met expectations as shown in Table 24.

Table 24. Split Unit Performance

	Outdoor DB [°F]	Indoor DB [°F]	Indoor WB [°F]	EER [btu/Whr]	Sensible EER [btu/Whr]	SHR [btu/btu]	Sensible Capacity [btu]
HDAC Design	115	80	63	8.22	8.22	1.0	25,818
Hot "Normal"	115	80	67	8.48	6.91	0.81	21,947
HOT Test	130	80	67	6.35	6.35	1.0	21,941
Dehumidification	82	80	67	14.45	9.07	0.63	20,410
ARI Test A	95	80	67	12.82	9.51	0.74	25,560
ARI Test B	82	80	67	15.83	11.22	0.71	27,088
ARI Test C	82	80	59	14.45	14.45	1.0	34,465

Source: Pacific Gas and Electric Company

Sensible EER and capacity were within 0.5% of the prediction from the ORNL model at the design conditions. Of equal interest were the sensible capacity and EER at somewhat more moist conditions, since some of the units will experience moisture removal most the time and all will remove moisture some of the time. The performance was as expected, and can be improved by latent recovery discussed below.

Lowering the evaporator airflow by almost 50% to 660 cfm to increase latent capacity (the dehumidification test) provided a marginally lower sensible heat ratio than the ARI Test B that was run at the same conditions (except with an evaporator airflow of 1100 cfm).

Refrigerant Charge Optimization

The amount of refrigerant and the metering device settings affect the performance of the air conditioner. Using nearly all the evaporator coil for phase change (near zero superheat) improves performance. The unit's refrigerant charge and metering device settings were optimized at the design hot/dry condition of 115/80/63. To optimize refrigerant charge the superheat was fixed at 6°F and the refrigerant charge was adjusted to obtain subcooling readings of 5,8, 10,12, & 13°F. The unit performance was measured at each refrigerant charge level and the results are plotted in Figure 49. Because of the dramatic change in performance at 12.75°F, the charge and metering device were set to obtain 12°F of subcooling and 2°F of superheat.

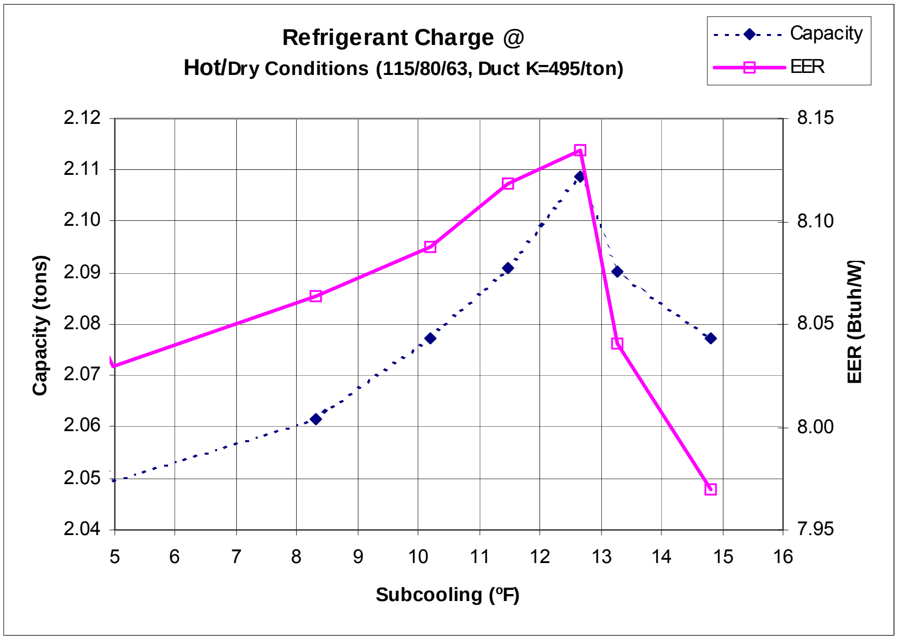

Figure 49. Refrigerant Charge Optimization – Sensible Capacity vs. Subcooling

Source: Proctor Engineering Group

Evaporator Airflow Optimization

Achieving high sensible capacities (high sensible heat ratios) has traditionally been accomplished by increasing the airflow across the evaporator coil. Taken in the narrow view this can be quite effective. However, in the broader view of the air conditioner installed in a home, higher airflows can be counterproductive. Since the average duct system provides a fixed resistance to flow, the duct system static pressures increase as the square of the airflow (as do the coil pressure drop and losses within the cabinet). As a result of the increased static pressures, the work of the fan motor increases as the cube of the airflow, leading to very high fan motor watt draws that can cancel the positive effect of higher sensible capacity. Therefore, the evaporator coil airflow needs to be optimized for the duct system, coil, and air handler/furnace. Figure 50 shows the results of the airflow optimization tests. These tests were done at hot dry design with a constant external duct resistance. The maximum PEERs was obtained at a little less than 1100cfm (about 350 cfm per ton). This was a somewhat surprising result given that standard recommendations for hot dry climates are 450 cfm per ton or higher.

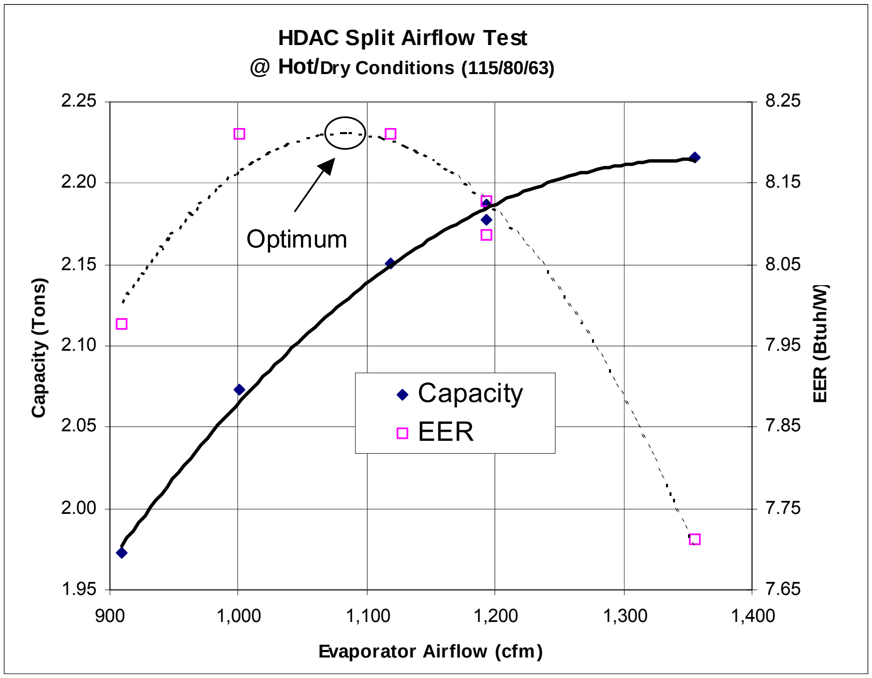

Figure 50. Evaporator Airflow Optimization – Sensible EER vs. CFM

Source: Proctor Engineering Group

Airflow Effects under ARI Test Conditions

Figure 51 shows the results of the ARI tests at different evaporator airflows.

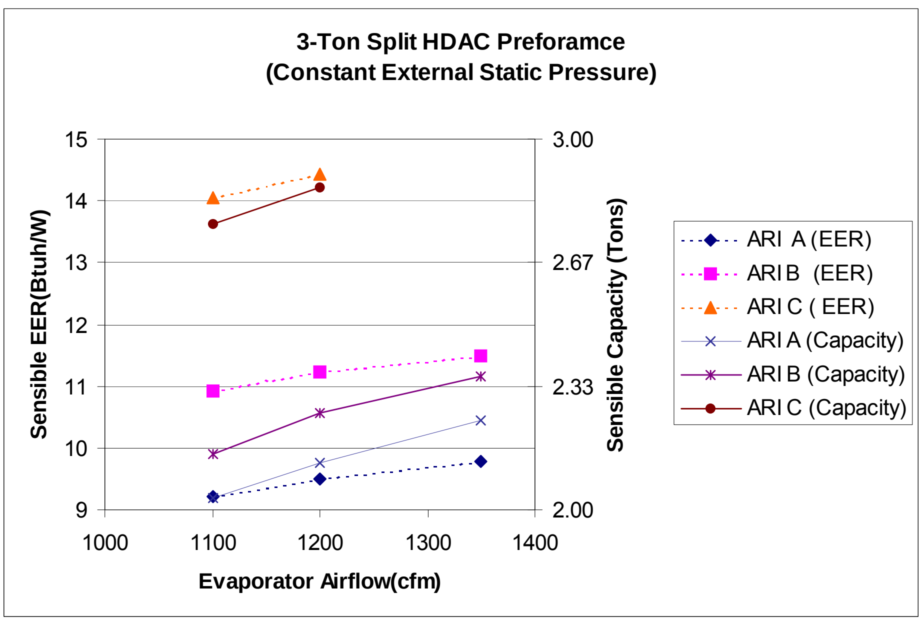

Figure 51. ARI Tests at Various Evaporator Airflows and Constant External Static Pressure

Source: Pacific Gas and Electric Company

SEER

The SEER is mathematically derived from ARI Tests B, C, and D. Conditions were not favorable for a full series of development tests to optimize the SEER by changes in the metering valve control and evaporator fan off time delay. With a two-minute fan off delay (which was less than optimum) and the metering valve partially open when the compressor was off (also less than optimum), the SEER was 14.6 as shown in Table 25.

Table 25. Split Unit SEER Calculation Results

	Qs (Btu/hr)	EER	CLF	Cd	SEER
ARI B		15.82			
ARI C steady state	34,450				
Cycle 1	3,160	12.70	0.1835	0.147	14.66
Cycle 2	3,137	12.66	0.1821	0.150	14.64
Cycle 3	3,126	12.69	0.1815	0.147	14.66
Cycle 4	3,131	12.59	0.1818	0.156	14.59

Source: Pacific Gas and Electric Company

Given further development time and based on the results on the 5-ton package unit, the team would expect a fully developed unit to have an SEER of over 15.

Condenser Airflow

The condenser airflow was tested at 3000 cfm and 4200 CFM. The sensible EER at hot dry design conditions was 6% higher at the higher speed and it had acceptably low condenser fan noise.

Latent Recovery

As discussed in the Latent Recovery subsection of Section 3.1.5, an air conditioner stores some of its cooling capacity as moisture removed from the house air and collected on the evaporator coil. In dry climates like California, Arizone, and Nevada were the indoor moisture is low there is no need to remove moisture from the house air. In those climates it is advantageous to recover the capacity stored in the moisture by using the evaporator coil as an evaporative cooler at the end of the cycle. Figure 52 displays the improvement in sensible EER when latent recovery was tested on the split unit.

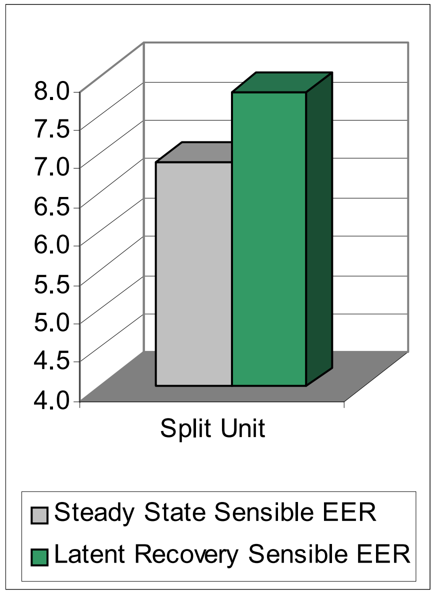

Figure 52. PEERs Improvement with Latent Recovery

Source: Pacific Gas and Electric Company

The laboratory test conditions for Figure 52 were 115ºF outside, 80ºF/67ºF inside.

3.2. Life Cycle Cost

The life cycle costs are based on a model of energy consumption contained in the Residential Alternative Calculation Method (ACM) Approval Manual (2005) and the Nonresidential ACM (2005). This methodology is used to calculate the hourly energy consumption for an air conditioner. The inputs to the model include standard building designs, hourly weather data, and air conditioner performance under different conditions. The details of these calculations are contained in *Supplementary HDAC Life Cycle Cost and Comparative Performance Report* (Proctor et al. 2005).

The California Energy Commission (CEC) is required by law to develop and maintain energy efficiency standards that are "cost effective, when taken in their entirety, and when amortized over the economic life of the structure when compared with historic practice"(Chief Counsel's Office 2005). In support of this requirement the CEC maintains a Life Cycle Cost (LCC) methodology for standards development. The version used in the 2005 building standards development (CEC 2005) was used for these LCC calculations.

The life cycle costs for this project were calculated two ways, the annual life cycle cost method and the hourly life cycle cost method using time dependent valuation (TDV). Time Dependent Valuation (TDV) accounts for time-of-use in determining cost effectiveness and requires estimated energy savings on an hourly basis. The CEC has published hourly TDV energy factors for both residential and nonresidential buildings for each of the 16 climate zones in California (Heschong Mahone Group 2002).

3.2.1. Energy Consumption Model

Because most of the climates in California are dominated by sensible cooling loads (cooling is required for temperature reduction not dehumidification), the model is a sensible model. The hourly sensible cooling loads are calculated for prototype buildings. The electricity needed to cool a typical residence or small business each hour was calculated by taking the sensible cooling load required for that hour and dividing it by the calculated Sensible EER corresponding to the given outside temperature:

$$CoolingElectricity\,(W \cdot h) = \frac{Sensible\,Cooling\ Load\,(Btu)}{Sensible\ EER\,(\frac{Btu}{W \cdot h})}$$

Within the model there are three regions of operation. The Sensible EER in each region is a function of the outside temperature, the Sensible SEER, the Sensible EER at 95ºF, and the slope of the Sensible EER line above 95ºF (by assumption or from air conditioner performance data. Figure 53 shows the temperature regions of operation and how the Sensible EER varies with outside air temperature.

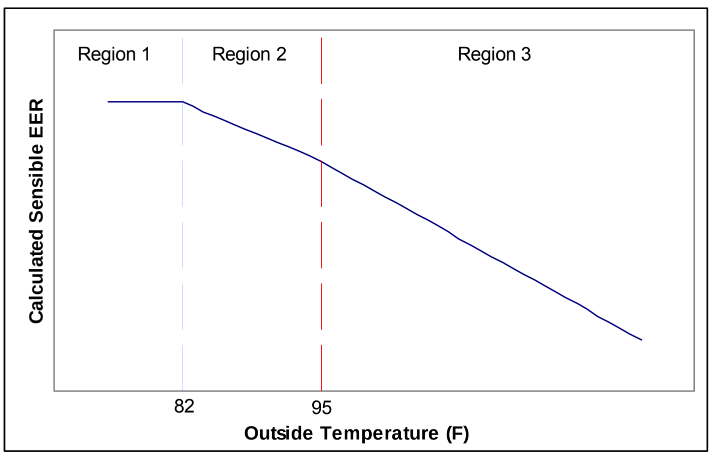

Figure 53. Piecewise Sensible EER Function.

Source: Proctor Engineering Group

The Sensible EER for outside temperatures lower than or equal to 82° F (Region 1) is equal to the calculated SEER for the unit. At temperatures below 82° F the cycling degradation is at its maximum so the Sensible EER equals the Sensible SEER.

$$Sensible\ EER_{\leq 82F} = Sensible\ SEER$$

Where

$$Sensible\ SEER\ = SEERxSHR\,@\,82F$$

From 82° F to 95° F the effect of cycling is assumed to decrease linearly. This temperature range is illustrated in Region 2 of Figure 53 and is described by:

$$SensibleEER_{82-95F} = SensibleSEER@82F + (SensibleEER@95F - SensibleSEER@82F) \times \frac{OutsideTemp\,F - 83F}{13}$$

At temperatures greater than or equal to 95° F the cycling losses are assumed to be small. In Region 3 then:

$$SensibleEERSlope = \frac{(SensibleEER@115F) - (SensibleEER@95F)}{115F - 95F}$$

and

$$SensibleEER_{>95F} = SensibleEER@95F + (SensibleEERSlope) \times (OutsideTemperatureF - 95F)$$

Air Conditioner Performance Inputs

For the purpose of this model the performance of an air conditioner is defined by the Sensible SEER, the Sensible EER at 95° F, and the Sensible EER at 115°F. These sensible energy efficiencies are defined as those that are likely to exist in air conditioners installed in California. This definition includes the airflow resistance of the ducts and external devices (other than the evaporator coil) found to be typical in field tests.

Based on field tests (Proctor and Parker 2003; Wilcox and Chitwood 2005), the external resistance (excluding the evaporator coil) to airflow shows the following relationship.

External Static Pressure (IWC) = $\left(\frac{CFM\ per\,ton}{495\,CFM\ \ per\ ton}\right)^2$

The Sensible EERs and Sensible SEERs were determined in one of the following ways (in order of preference):

1. Laboratory test data at the specified airflows, static pressures, indoor and outdoor conditions.
2. Laboratory test data adjusted for specified airflows and static pressures.
3. Manufacturer's published data on the outdoor unit, the evaporator coil, and the furnace blower, corrected to the external static pressure.
4. Oak Ridge National Lab (DOE/ORNL) Heat Pump Design Model Mark VI.

Tables 26 and 27 describe the units and the source of their performance data.

Table 26. Split Units' Descriptions and Data Sources

Unit	Manufacture, Model #	Source	Adjustments
SEER 12 Baseline	Outside Unit: Carrier, 38EZG036310Inside Unit: Carrier, FX4BNF036Furnace: Ruud, UGPL-07EBRQR	Purdue Laboratory Testing (Shen et al. 2004)	Data adjusted for typical ductwork.
SEER 13 Baseline	Outside Unit: Lennox, HS26-036Inside Unit: Lennox, C33-44CFurnace: Lennox, G40UH036A-040	Manufacturers Data Sheets	Data adjusted for typical ductwork.
SEER 14 .25 (R-22)	Outside Unit: Lennox, HS27-036Inside Unit: Lennox, C33-38A/CFurnace: Lennox, G60UH-36B-090	Manufacturers Data Sheets	Data adjusted for typical ductwork.
SEER 15 .3 (R-22)	Outside Unit: Lennox, HS27-036Inside Unit: Lennox, C33-38A/CFurnace: Lennox, G50UHI-36B-090/ECM	Manufacturers Data Sheets	Data adjusted for typical ductwork.
SEER 14 .2 (R-22)	Outside Unit: Lennox, HS27-036Inside Unit: Lennox, C26/51/65Furnace:Lennox, G60UH-36B-090	Manufacturers Data Sheets	Data adjusted for typical ductwork.
HDAC (SEER 14.7)	PEG, 3-Ton split HDAC	PG&E Laboratory Testing	

Source: Proctor Engineering Group

Table 27. Package Units' Descriptions and Data Sources

Unit	Manufacture, Model #	Source	Adjustments
SEER 12 Baseline	Carrier, 48GP060090	Purdue Laboratory Testing (Proctor et al 2005)	Adjusted to 5-ton unit. Tested unit was a 3-ton system.
SEER 13 Baseline	Carrier, 48HJE006- -31	Manufacturers Data Sheets	Data adjusted for typical ductwork.
SEER 14	Carrier, 48PG06	Manufacturers Data Sheets	Data adjusted for typical ductwork.
HDAC (SEER 17.5)	PEG, 5-Ton Package HDAC	SCE Laboratory Testing	

Source: Proctor Engineering Group

Data used in Life Cycle Cost calculations are shown in Table 28.

Table 28. Baseline, Comparison, and HDAC Performance Inputs to LCC Model

System	Sensible Heat Ratio	SEER	Sensible SEER	Sensible EER@ 95/80/67 (ARI Static)	Sensible EER@ 115/80/63 (Field Static)
3-Ton Split SEER 12	0.68	12	8.12	6.82	6.22
3-Ton Split SEER 13	0.72	13.15	9.36	8.47	7.18
3-Ton Split SEER 14	0.74	14.25	10.25	9.1	7.04
3-Ton Split HDAC	0.71	14.7	10.47	9.51	8.22
5-Ton Package SEER 12	0.75	12	9.05	7.29	5.98
5-Ton Package SEER 13	0.69	13	8.97	7.99	6.23
5- Ton Package SEER 14	0.67	14	9.38	8.80	7.35
5-Ton Package HDAC	0.69	16.7	11.56	10.19	8.6

Source: Proctor Engineering Group

Adjusting Manufacturers' Published Data to Field Conditions

Manufacturers' expanded capacity tables list cooling capacities and unit powers at various indoor conditions, outdoor conditions and evaporator airflow. Manufacturers do not have a uniform way of listing the total and sensible capacities and unit power consumption. Some manufacturers provide the gross ratings (no evaporator fan power or heat) while others supply net ratings (with evaporator fan power and heat). The unit power is listed in one of three ways: gross (compressor and condenser fan), net (compressor, condenser fan, and evaporator fan), or compressor only.

These data were made consistent as described in Section 2.11.

Life Cycle Cost

The Light Commercial and Residential yearly electrical consumption were determined for the baseline units, the HDAC units, and for other currently manufactured units. The annual energy savings are the differences between the HDAC units and each comparison unit (baseline or other current unit) (Table 1).

Table 29: Average Life Cycle Cost Benefits

	Energy Savings (kWh/hr) kWh/year?	Peak Reduction (kW)	Life Cycle Cost Savings over 18 years
Residential			
SEER 12 Baseline	656	1.53	$905 - $1,457
SEER 13 Baseline	231	0.57	$319 - $509
SEER 14 Baseline	85	0.43	$118 - $213
Commerical			
SEER 12 Baseline	1741	1.95	$2,507 - $3,391
SEER 13 Baseline	1619	1.46	$2,331 - $3,037
SEER 14 Baseline	1157	0.51	$1,666 - $2,051

Source: Proctor Engineering Group

Consumer First Cost

The estimated difference in initial costs between SEER 13 and HDAC systems are $246 for the Residential Unit and $67 for the Commercial unit. These estimated were derived using the DOE methodology as used in federal rule setting. The results are similar to estimated produced by LBNL for the CEC (Rosenfeld, Rosenquist, and Rice 2005).

3.3. Air Conditioner Performance Comparison

A goal of the project was to design and build HDAC units that showed a 15% to 25% reduction in peak energy consumption compared to the baseline. The peak energy consumption on hot dry climates is determined by the PEERs (Sensible Heat Ratio * EER) of the machine. Both the split HDAC and the package HDAC meet the design criteria. Of greater importance, the HDAC units also showed significant peak reduction improvements over SEER 13 units (units that meet the new federal standard for 2006).

The hot dry peak performance (Sensible EER) of the split system compared to the SEER 12 and SEER 13 baselines is shown in Figure 54.

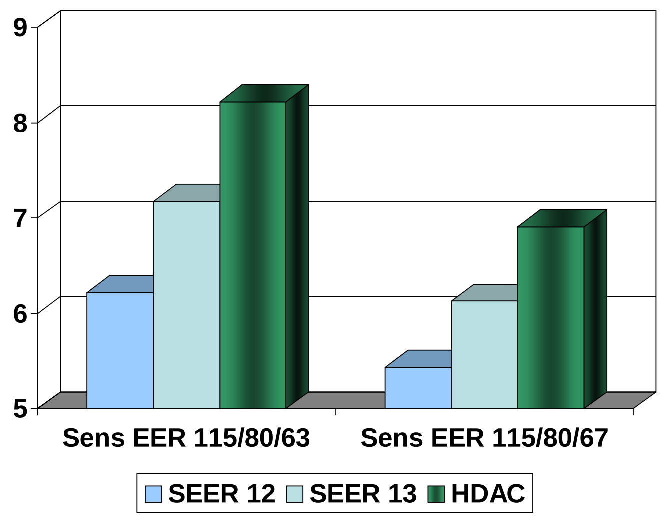

Figure 54. Split HDAC Performance

Source: Pacific Gas and Electric Company

The hot dry peak performance of the package HDAC unit substantially exceeded expectations. The package unit hot dry peak performance is shown in Figure 55.

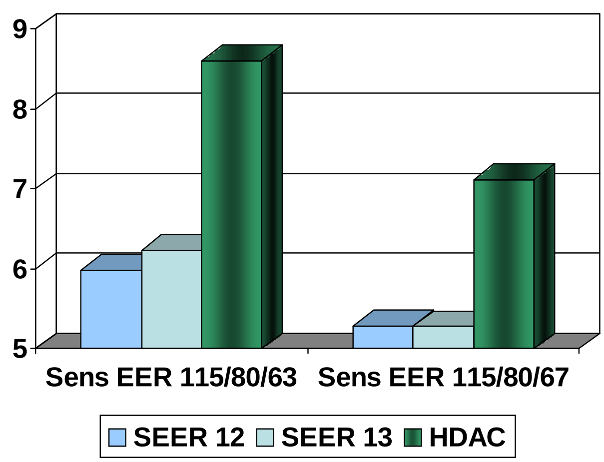

Figure 55. Package HDAC Performance

Source: Southern California Edison

Comparison to Available Equipment

One of the questions in the development of the HDAC specification is whether existing equipment with small or no modification can be combined in a way that produces the desired performance with little or no change in the procedures at the manufacturing plant. To answer this question the team compared manufacture's published AC performance to the performance of the HDACs.

Figure 56 shows that existing split system equipment carefully chosen for high sensible efficiency and relatively low evaporator fan watt draw compares favorably with the split

HDAC. This figure compares a Lennox HS27-036 (a R-22 unit) outdoor unit with combinations of two different evaporator coils (33-38 and 26-51/65) and two furnaces (an ECM furnace and a flow efficient standard furnace).

It is important to note that as new 410A machines come on the market that their PEERs performance will be less than R-22 machines of the same SEER.

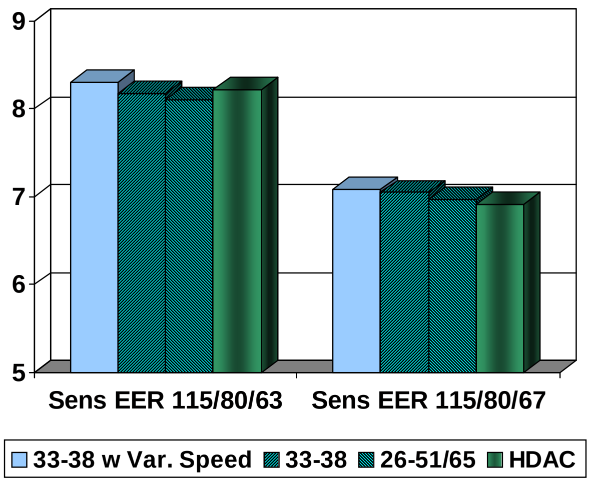

Figure 56. Split System Performance Comparison – Available R-22 Production Units to R-410A HDAC Unit

Source: Proctor Engineering Group

Figure 57 shows a comparison of the HDAC package unit with existing package units. The initial investigation found no current units that performed as well as the HDAC package unit. It is important to note however that the HDAC unit was a modified version of an existing package unit. The HDAC kept the same evaporator coil, furnace heat exchanger, evaporator fan and motor, and compressor.

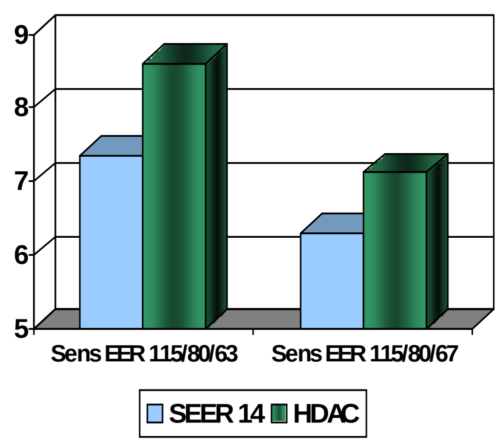

Figure 57. Package System Performance Comparison – Available Production Unit to HDAC Unit

Source: Proctor Engineering Group

4.0 Field Testing

4.1. Overview

The purpose of the HDAC field test was to determine the field performance of air conditioners selected to meet (or approach) the draft HDAC specifications. The design of the PIER project anticipated a number of the differences between standard laboratory tests and field conditions. The PIER project tested the proof of concept HDACs at the duct airflow restrictions common to the field, at temperatures approached or achieved at peak conditions, and under both moderate and dry indoor conditions. Nevertheless laboratory testing does not cover the full range of conditions experienced in the field, including occupant behavior, duct system performance, thermostat effects, and most importantly – air conditioner cycling.

Once the draft specification was produced a number of manufacturers were approached to provide air conditioners that would meet the draft specifications by selection of existing components or modifications to their existing equipment. Three major manufacturers responded with combinations of existing components that, on paper, approached within 3% of the draft specifications[6].

Units from those three manufacturers were installed and monitored in three utility company service areas by a team consisting of Paragon Consulting Services, ADM Associates, and Proctor Engineering Group (the Team).

Sites were selected in seven locations. Four of the locations were in Pacific Gas and Electric (PG&E) Company's service area; two locations were chosen in Southern California Edison's (SCE) service area; and one location was chosen in Nevada Power's (NP) service area.

These results are dependent upon, and borrows heavily from the NP Final Report Hot Dry Climate Air Conditioner (HDAC) Measurement and Verification (M&V) Residential Field Test by Paragon Consulting Services as well as the SCE Data Collection Report for Testing of Optimized Air Conditioner Design in Hot Dry Climates by ADM Associates.

An additional report, the PG&E Hot Dry Climate Air Conditioner Pilot Field Test by Proctor Engineering Group is available for details on the PG&E portion of the field test.

A more extensive description of the field testing can be found in *Hot Dry Climate Air Conditioner (HDAC) Combined Field Test Report*, Proctor Engineering, May 2007.

4.2. HDAC Specification

The HDAC specification developed as a result of the laboratory testing and used in assessing potential manufacturer products was as follows:

[6] Specifications that were lower than the performance of the PIER HDAC units.

Table 30. Hot Dry Air Conditioner Draft Specifications

Condition #1	Hot Dry 115/80/63
Gross Sensible Capacity (sensible btuh)	75% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 8 btu/watthr
Condition #2	Hot Medium 115/80/67
Gross Sensible Capacity (sensible btuh)	65% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible PEER	at least 6.8

Source: Proctor Engineering Group

Table 30 Notes:

1) With the External Static Pressure from the return plenum to the supply plenum downstream of the evaporator coil is defined by $\left(\frac{CFM\ per\ ton}{495\ CFM\ \ per\ ton}\right)^2$ An air conditioner system (furnace, outside unit, and evaporator coil) with a flow of 400 CFM per ton would be tested at 0.653 IWC.

2) Net Sensible PEER is the net sensible capacity divided by the total unit watt draw.

4.3. Site Selection

The Team completed a field test to compare the performance of standard air conditioners to air conditioners selected for hot and dry climates (HDACs). The field test consisted of site and AC selection, installation and replacement, performance monitoring, and data analysis. Standard (baseline) SEER 13 air conditioners were first monitored and then replaced with HDACs. Monitoring of the baseline and HDAC units were completed during the summer of 2006.

The characteristics of the homes and air conditioners used in the project are listed in Tables 31 and 32.

Table 31. Site Characteristics

Site	Bakers-field	Concord	Madera	Yuba	Furnace Creek	Victor-ville	Las Vegas
House Size (square feet)	1200	1400	1650	1600	1230	1600	1225
Year Built	1941	1970s	2002	1991	NA	2004	NA
Air Handler Location	Bedroom Closet	Hall Closet	Attic	Attic	Package Rooftop	Attic	Attic

Source: Proctor Engineering Group

Since the sensible heat ratio is the fraction of the cooling that reduces the indoor temperature, it is evident that these standard units wasted almost a quarter or more of their cooling capacity removing water rather than reducing the temperature. Designs capable of sensible heat ratios of 0.80 or higher are possible and come of the installed HDACs approached this ratio.

The standard ACs were SEER 13 R-SS units either already in place or selected by the contractor and installed for this test.

Table 32. HDAC Air Conditioner Specifications[7]

Site	Bakers-field	Concord	Madera	Yuba	Furnace Creek	Victor-ville	Las Vegas
Rated SEER	13.25	13.5	14	14.2	14	13 to 13.7	15
Rated Sensible EER	8.5	8.8	9.4	9.2	8.25	NA	NA
Rated EER	11.2	11.3	12.3	11.7	11.7	10.7 to 11.2	12.5
Sensible Heat Ratio (temperature reduction fraction)	0.76	0.78	0.77	0.79	0.73	NA	NA
Rated Capacity (Btuh)	36400	44100	50730	35200	59000	52000 to 56000	35600
Nominal Size (Tons of Cooling)	3	3.5	4	3	5	5	3
Nominal Evaporator Coil Capacity (Btuh)	48000	60000	60000	42000	60000	NA	48000
Refrigerant	R-410A	R-410A	R-410A	R-410A	R-410A	R-410A	R-410A
Metering Device	TXV	TXV	TXV	TXV	TXV	NA	TXV
Fan Motor Horsepower	3/4	1/2	1/2	1/2	3/4	NA	3/4
Fan Motor Type	ECM	ECM	ECM	ECM	ECM	ECM	ECM

Source: Proctor Engineering Group

The HDAC air conditioners consisted of components (outside unit, inside coil, and furnace) selected because they approached the draft HDAC performance specification. The selections were based on published performance data on the outside unit and coil combination, the coil pressure drop, and the furnace blower provided by the manufacturers.

[7] With ARI furnace default assumptions and at standard 95/80/67 conditions.

4.4. Results

Results were obtained for seasonal cooling energy consumption (kWh) as well as coincident and non-coincident peak power draw (average kWh per hr). These results are presented for standard operation[8].

The seasonal cooling energy consumption of each unit and the annual energy savings are shown in Table 33.

Table 33. Standard vs. HDAC Performance Summary

Location	Standard Unit Annual Energy Usage (kWh)	HDAC Unit Annual Energy Usage (kWh)	Energy Savings (kWh)	**Annual Energy Savings (%)**
Las Vegas	2770	2291	478	**17%**
Furnace Creek	11086	9232	1854	**17%**
Victorville	3534	2498	1036	**29%**
Madera	1966	1618	348	**18%**
Yuba	1592	1256	336	**21%**
Bakersfield	3059	3262	-203	**-7%**
Concord	420	443	-23	**-5%**

Source: Proctor Engineering Group

Las Vegas, Furnace Creek, Madera, and Yuba all showed substantial Annual Cooling Energy Savings of 17% to 28%. Bakersfield and Concord showed increases in Annual Cooling Energy Use of 7% and 5% respectively. The units in Bakersfield and Concord were intensively monitored and it was determined that they performed well below the manufacturers' published data.

Peak Demand Savings

Peak demand is determined by a combination of air conditioners that are running continuously for the whole hour at peak, air conditioners that are off for the full hour, and air conditioners that are cycling during the hour.

The units monitored in this field test were specifically selected to make sure they were used during peak periods. Their operation at peak was as follows:

- Bakersfield and Furnace Creek were continuous running.
- Concord, Victorville, Las Vegas, and Madera were cycling.
- Yuba used daytime thermostat setups and was cycling on some peak days and continuous running on other peak days (depending on the severity of the thermostat change).

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[8] The PG&E test included additional operating modes designed to obtain higher overall sensible cooling.

Peak demand reductions at system peak depend on the operating conditions of the unit.

1. For units where the capacity is less than the load at peak, the savings will be the difference between the connected loads of the units (Connected load is Capacity/PEERs). For units with equal capacity then, the peak savings is 1 – PEERs std. / PEERs hdac.
2. For units where the capacity exceeds the connected load, the peak savings will again be 1 – PEERs std. / PEERs hdac.

Coincident Peak Demand

The peak demand of major importance occurs on hot afternoons and is driven by the diversified air conditioner demand. The diversified peak demand of air conditioners is generally coincident with the peak demand of the system. The hours from 3PM to 6PM are of particular significance. The coincident peak demand for matched peak days are shown in Table 34.

Table 34. Standard vs. HDAC 3PM to 6PM Coincident Peak Demand Summary

	Las Vegas	Furnace Creek	Victor-ville	Bakers-field	Concord	Madera	Yuba
Standard Unit 3PM to 4PM Peak Demand (W)	1849	6245	4102	3156	2751	1975	1914
HDAC Unit 3PM to 4PM Peak Demand (W)	1612	6025	3040	na	na	1418	1293
Average 3PM to 4PM Peak Demand Reduction (W)	237	220	1062	0	0	557	621
	13%	4%	26%	0%	0%	28%	32%
Standard Unit 4PM to 5PM Peak Demand (W)	1934	6098	3935	3159	2624	2178	1960
HDAC Unit 4PM to 5PM Peak Demand (W)	1416	5935	2910	na	na	1562	1262
Average 4PM to 5PM Peak Demand Reduction (W)	518	163	1025	0	0	616	698
	27%	3%	26%	0%	0%	28%	36%
Standard Unit 5PM to 6PM Peak Demand (W)	1872	5962	3768	2902	2859	2254	2018
HDAC Unit 5PM to 6PM Peak Demand (W)	1541	5977	2780	na	na	1751	1302
Average 5PM to 6PM Peak Demand Reduction (W)	331	-15	988	est. 0	est. 0	503	716
	18%	0%	26%	0%	0%	22%	35%

na = insufficient higher temperature data with no reason to believe performance is better than standard.

Source: Proctor Engineering Group

The non-coincident peak loads were substantially reduced (between 237 and 1062 Watts) at four of the locations. At Bakersfield and Concord the HDAC units were performing well below the manufacturers' reported performance and no better than the standard units they replaced. At Furnace Creek both the standard and HDAC units were undersized to the load and had the same connected loads. The HDAC had a larger delivered capacity however under peak conditions it was still operating continuously.

Average Peak Demand

The average peak demand is over a much larger period[9], including periods where the watt draw is considerably less. The average peak demand of each unit and the average peak reductions are shown in Table 35.

Table 35. Standard vs. HDAC Average Coincident Peak Demand Summary

	Standard Unit Average Coincident Peak Demand (W)	HDAC Unit Average Coincident Peak Demand (W)	Average Coincident Peak Demand Reduction (W)	Average Coincident Peak Demand Reduction (%)
Las Vegas	1630	1292	339	**21%**
Furnace Creek	4125	3632	493	**12%**
Victorville	2633	1903	729	**28%**
Madera	1080	902	177	**16%**
Yuba	1041	847	194	**19%**
Bakersfield	1888	2073	-186	**-10%**
Concord	297	312	-16	**-5%**

Source: Proctor Engineering Group

The regulatory coincident peak contains many hours when even the undersized air conditioner at Furnace Creek was cycling. As a result all five of the performing units showed significant peak reductions of 12% to 28%.

Occupant Survey

The occupant surveys were performed by interview as described in the Methodology Section.

[9] Noon to 7 p.m. Monday through Friday, June 1, through September 30.

Table 36. Occupant Satisfaction Survey Results

	Comfort	Humidity	Noise	Occupant's Comments
Bakersfield	No Difference	No Difference	HDAC Quieter	HDAC did not cool the house down as fast as the standard unit and ran longer periods. HDAC airflow was noticeably higher.
Concord	No Difference	No Difference	HDAC Quieter	
Madera	No Difference	No Difference	HDAC Quieter	HDAC had less fluctuation in inside temperatures.
Yuba City	No Difference	No Difference	HDAC Quieter	The HDAC provided better cooling in back rooms

Source: Proctor Engineering Group

It is interesting to note the difference between perception and reality. Monitored data from the Bakersfield home shows that the HDAC unit actually ran shorter periods than the Standard unit, and that there was no difference in "pull down time" or indoor temperatures. It is possible that the occupants comments about "running longer" might be translated to "more often" since the number of cycles per hour was greater with the HDAC unit.

5.0 HDAC Market Connection

5.1. Market Connection Strategy and Plan

The Market Connection component of the HDAC Project began with initial planning of outreach activities to be carried out within Task 6.4. Post-project technology transfer activities were also recommended during the project, and are covered in Section 4. Market Connection planning included specific goals, overall strategy, activities and media, schedule, and specific responsibilities.

5.1.1. Market Connection Planning

The focus of the Market Connection planning effort was on mapping a path to commercial success that identified elements such as securing manufacturer commitments, educating key decision-makers and intermediaries, reaching an initial market, demonstrating and documenting value, refining and expanding applications. In addition to resolving state, regional and national codes and standards issues, the plan included beyond-code (i.e., incentive) opportunities, providing aids for equipment specifiers, and assuring continued manufacturer commitment to advancing the market and supply.

Additional tasks considered were to build a utility incentive program, to provide advice for upgrades to California's Title 24 specifications, to recommend testing standards, to develop a DOE procurement program, and to modify DOE's appliance standards and California's Title 20 standards. These tasks were classified as post-project activities after they proved to be beyond the allowable timeframe of the project. Other deferred topics included the general codes/standards/incentives topic area, in contrast to the elements of the plan related to public and professional education, manufacturer business cases, and other non-regulatory supply chain issues.

Because the HDAC project evolved continuously based on results of prototype analysis, utility and manufacturer participation, and field testing, initial Market Connection plans changed in response to the overall project's development such that no formal task report could be prepared before the project term ended. This section describes the Market Connection approach that resulted over the project term.

5.1.2. Assessment of the HDAC Market and Barriers

The Market Connection strategy for the project began with the need to assure that the HDAC would find an adequate market for manufacturers to be interested in the production of a region-specific product instead of the standard specification used nationwide. This required an assessment of the size, location, and other aspects of the potential market for the hot-dry air conditioner. The approach and results of that market assessment are described in Section 4.2. The results indicated that the HDAC market is very large—nearly a million units per year in potential sales. No substantial barriers were identified, apart from the need for market-justifiable prices and energy cost savings. Distributors, dealers, and developers that were interviewed indicated confidence and a willingness to sell and install HDAC units.

The original barriers assessment assumed that the HDAC would be a new product involving improvements in both the condenser and evaporator units, with a unique specification that would be easy to order with no difference in installation or service from conventional units. However, as the project's prototypes were completed, tested, and specifications for commercial HDAC units were derived from their performance, the participating manufacturers found that specific combinations of their existing condenser and evaporator products (either complete furnaces, air handlers, or evaporator coils) appeared able to perform within a few percent of the prototype HDAC's peak-load efficiency and demand. Consequently, a new barrier became evident in the need for more precise specification than is typically done for unit combinations for each installation. At the same time, this situation also represents a more general need for improved specification of installations even without a specific HDAC product.

5.1.3. Development of Alliances

It was clearly understood in the HDAC project that the manufacturing industry would likely be hesitant to introduce a new type of air conditioner product. The industry has long held to a standard national specification of air conditioning products, with strong opposition to regionalization of air conditioner variations. As a result, the market connection strategy included several alliance-building steps:

- Direct outreach to major manufacturers based on demonstration of prototype success and market characterization.
- Representation of key stakeholder groups on the project's formal review committee, including manufacturers, utilities, regulators, and energy efficiency advocacy groups.
- Inclusion of all major Western utilities in order to build and demonstrate support for special incentives for HDAC installations when commercially ready.
- Presentations to key air conditioning industry groups such as the ARI and ASHRAE.

All these activities were implemented effectively through direct contact. Post-project activities should continue to encourage these alliances and others in order to properly support the manufacturers' interest in HDAC concept commercialization.

5.1.4. Strategy

The original Market Connection strategy envisioned a hardware product that the project would endeavor to convince major HVAC manufacturers to build and market. That product would be based on the technical innovations incorporated by Proctor in the split-system and rooftop prototypes that proved successful in lab testing. However, opportunities for two major changes in that strategy soon arose.

The first change was in the realization that each manufacturer would insist on its own approach to the problem rather than simply adopting the prototype innovations. This led to a decision to provide the manufacturers with an HDAC performance specification rather than technical details, allowing them full freedom to apply their own engineering talents and business needs. This approach was strongly supported by the manufacturers, especially since they were, at that time, still implementing the federally mandated SEER 13 minimum equipment upgrade requirement, the largest new investment and engineering change in the industry in decades. It was not realistic to expect that they would be ready to change the newly built production designs and manufacturing procedures that were in the process of being refined.

The second change was brought about by the manufacturers' discovery that their existing product lines included available combinations of condenser and evaporator units that would nearly meet the prototype-derived HDAC performance specification. This led to adopting a new two-phase approach to the project. The first phase involved initially identifying the appropriate combinations and some education and marketing outreach to distributors, dealers, and housing developers in order to properly specify equipment for both new and retrofit applications. No manufacturing changes would be required in this phase. The major Market Connection activity required would be to publicize the project's later successful field test results of such existing equipment combinations and work with the manufacturers and others to encourage the necessary field personnel education in proper specification and the marketing advantages of that effort.

5.2. Market Characterization

5.2.1. Western HVAC Sales Volumes

A separate task report was delivered by the project's Market Connection team that presented a market characterization based on extensive analysis of available statistical data, additional insight gained through the development of the proof-of-concept units, and interviews with various market players. This market assessment estimated the air conditioner/heat pump markets in the hot-dry regions of the West, including sourced estimates of the installed heat pump and air conditioning equipment populations by location, unit size, and age.

Because the intent of the market study was to indicate the total potential HDAC market in the United States, it was broadly inclusive of all the hot-dry climate areas of the West—from West Texas through the Mountain States (Arizona, New Mexico, Colorado, Utah, Nevada) to California. Further, low-population hot/dry areas of more northern states, such as eastern Oregon and Idaho, were not included due to lack of data.

Data Sources Used

1. Briggs, Lucas and Taylor, Climate Classification for Building Energy Codes and Standards, ASHRAE Technical Paper, 2002.

2. Current Industrial Reports, U.S. Bureau of the Census, http://www.census.gov/industry/1/ma333m04.pdf

3. Residential Energy Consumption Survey 1997 and 2001; Energy Information Administration (2005 update is not yet completed)

4. Housing Unit Estimates 2000-2004 by State, U.S. Census Bureau

5. American Housing Survey 2003, U.S. Census Bureau

6. Phone conversations with industry experts Joseph Pietsch and Matt Tyler
7. Informal interviews with HVAC distributors and installers in hot/dry areas

Conclusions

- The 3-ton split and 5-ton rooftop units are reasonable choices for prototype models, due to their high reported sales figures as compared to those for other standard sizes in the 5-ton and under category.
- Using 2000 CDD (cooling degree days per Briggs et al as listed above) as the standard, the Mountain region has about the expected number of central air conditioners installed for the population in the 2000+ CDD areas but California has several times the level of air conditioning expected by that measure. This suggests a high level of air conditioning use in parts of the state that are dry but mild in climate.
- The approximate projected 2007 yearly sales estimates, which includes all sizes 5 tons and under for the two types of air conditioners in the hot/dry region for this project are:

Air Conditioner Size Category	Estimated Annual Sales Volume
All splits (assumed primarily residential)	**830,000 units per year**
All unitary (assumed primarily commercial)	**100,000 units per year**
Total all sizes 5 ton and below	**~930,000 units per year**

- Total annual air conditioner sales in hot/dry regions for all sizes 5 tons and below was estimated to be approximately 930,000 units.
- The residential air conditioner market in the hot-dry target area is 47% new construction, an unusually high proportion due to the heavy home construction activity in these states in the past few years. As the new construction volume has declined since the completion of that market study, the existing residential market will become more dominant.
- Residential unit sales vastly outnumber small commercial sales, accounting for approximately 90% of total units sold in sizes 5 tons and below.

A variety of data sources, often not directly consistent or complete with all desired details for the purposes of this study were used in the market assessment. Several topics could be addressed only through assumptions, which are documented in the task report. Because of the data imperfections, the results reported here should be seen as approximate and subject to change under other data, interpretations, and assumptions. In particular, assumptions regarding the extent of the hot-dry sales region are subject to differing interpretations, as are the assumptions regarding the allocation of split-system as compared to unitary/rooftop units between the residential and commercial markets. However, these results clearly indicate that the potential HDAC annual market is a substantial proportion of national air conditioner sales, and represents a major source of future energy and peak demand savings. At a conservative peak demand savings of 15% compared to SEER 13 units as currently specified and an estimated 3kW marginal peak demand per conventional unit with no cycling at the peak (implying proper unit sizing), the annual technical potential *added year-over-year incremental reduction* in peak demand for this annual market size is approximately 400 MW. This implies avoidance of building one or two large power plants for the West *each year*. Even with conventional tendencies in unit oversizing, leading to unit cycling even at peak conditions, full saturation of HDAC units could yield as much as 150 MW per year in incremental peak savings. Similarly, if the average HDAC unit were to save only that 15% for 300 hours per year, total technical potential savings for the population of units replaced or added each year would be about 500,000 MWh.

Result: Choosing the Appropriate Air Conditioner Sizes to Prototype

One purpose of the market study was to use basic recent unit sales data to confirm the appropriateness of the project's test unit sizes, based on their sales relative to other sizes. The 2004 US Census Bureau's Current Industrial Reports data, as cited earlier, indicated several key points regarding the unit types for this project, test unit sizing, and the residential versus commercial sales volumes. This data is aggregated at the national level; no regional or state-specific data were found.

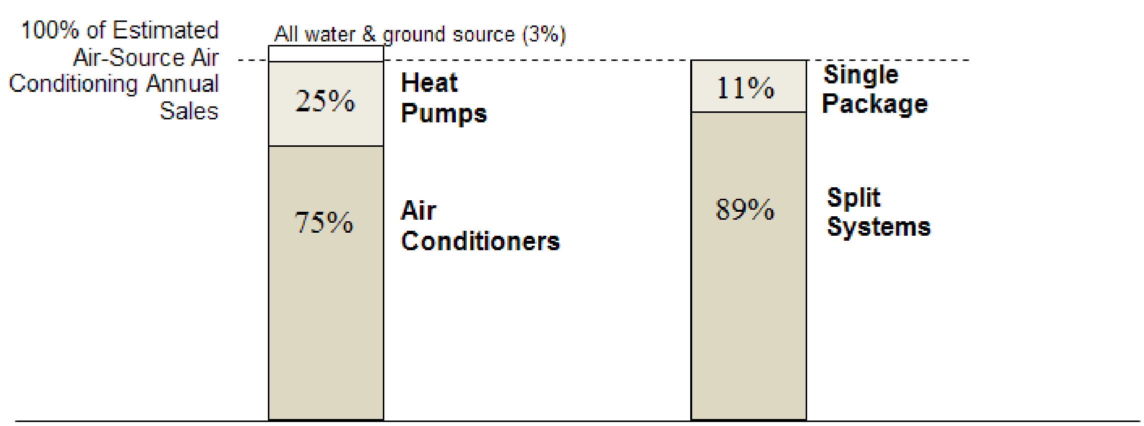

Figure 58: Composition of National Air Conditioning Sales by Major Characteristics

Source: US Census Bureau

- In 2004 air-source heat pumps of all types accounted for about 25% of the total air-source units sold nationally versus 75% for air conditioners.
- Water-source air conditioners and heat pumps accounted for an additional 3% beyond the air-source total (excluded from this study).
- Overall, split systems (assumed primarily residential) account for 89% of the total, while the single-package systems, including both a/c-only and year-round (gas pack) a/c units, assumed to be predominantly commercial, represent 11% of national sales.

- For single-package and year-round a/c units, the proposed 5-ton size is similar to other sizes in sales volume and is therefore a valid representative of this configuration for the prototype development for this project.
- For split systems, the proposed 3-ton size accounts for the greatest numbers of units sold compared to all other sizes. This confirms that a 3-ton model is a realistic choice for this project's prototype development.

Saturation of Air Conditioning Market

A comparison between 2000+ CDD households to households with central air installed suggests that California is oversaturated with central air conditioning considering it's climate conditions. California has far less of it's population in the warmer areas yet, more than three times the expected concentration of air conditioners when using the 2000 CDD standard. The Mountain region is almost perfectly in line with expectations if we consider the 2000 CDD to be the appropriate dividing line.

These findings suggest that the populations of the Mountain states have a much higher exposure to hot-dry weather (27.9%) than does the California population (8.9%). Further, Californians in cooler areas (CDD<2000) of that state account for the majority of the state's residential air conditioners. The California air conditioner market is clearly much broader than just the nominally "hot" areas. We considered deleting air conditioners sold in those less-hot (but still dry) areas, based on the apparent oversaturation, but decided that since most of the air conditioners located in those less-hot areas still experience some very hot days and normally have low humidity in the cooling season, all California air conditioners should be included in the hot-dry a/c market estimate.

Saturation is much lower in the Mountain states, where the population using air conditioning is much closer to the proportion living in the 2000+ CDD areas. As in California, we therefore make no correction for saturation. We conclude that using the available Mountain States plus California data, adjusted only for the addition of West Texas, needs no further adjustment for this study's market estimations.

Assessing Air Conditioner Age in Existing Homes

The saturation data above indicate that the entire hot-dry region has reached technical market saturation in existing homes—that is, the percentages of homes with air conditioners in both California and the Mountain states is at least as high as the percentages of homes in the hot portions (CDD≥2000) of those states. That data also suggests that central air conditioning has not reached true saturation in most of the West if the technically oversaturated California situation can be taken as an indication of future market expansion potential in the other hot-dry states. Perhaps this might be driven by future advances in average income and living standards, but for the purposes of this study, we must consider that upside potential to be only speculative due to the lack of more definitive market indicators. We assume conservatively, therefore, that few future air conditioner sales will be in existing homes currently without air conditioning—that is, other than replacements or new construction.

To estimate replacement volume, we can consult the EIA report as shown in the following table.

Table 37. Central Air Conditioner Age (MM units)

	Total U.S	California	Mountain
< 2 years	6.1	0.3	0.3
2 to 4 years	10.1	0.6	0.74
5 to 9 years	15.3	0.7	0.5
10 to 19 years	15	1.3	0.4
20 years or More	5.6	0.6	0.3
Don't Know	3.7	0.4	0.2

Source: EIA Residential Energy Consumption Survey, 2001

Over half the air conditioners in California were 10 years old or older in 2001, the latest year available with such data. Therefore we should expect many if not most of those air conditioners (with an assumed typical life of 15 years) to be up for replacement now through the next few years. In contrast, almost half of the total air conditioners in the Mountain states had been sold in the preceding four years. This suggests relatively moderate replacement demand in the near future in those states. California has shown that many people will install air conditioners in milder climates, so the Mountain states may follow that lead in the coming years as the standard of living rises in those areas. However, we conclude that replacement sales rates in the Mountain states for the next several years are likely to be lower than in California—although by 2006 those same units now average 9 years of age and will enter the heavy replacement period within this decade. Consequently, we make no correction for replacement timing.

Combining Data to Determine Sales Figures

As outlined in Section 2 above, the Current Industrial Report data for 2004 provides estimates of HVAC sales by type and size but only for the United States as a whole. In this study we estimate annual sales in the hot-dry region by multiplying the CIR national estimates by an adjustment factor based on the proportions of total units in place in the hot-dry region as compared to the national totals, using the latest (2001) EIR housing characteristics data. This requires combining the EIR's Mountain region and California data. To yield estimates of sales only in the hot-dry areas, the California data must be adjusted to remove the "oversaturation" (proportion of sales in parts of the state in climate zones other than hot-dry). Conversely, in estimating unit sales in the Mountain states, where we have shown undersaturation (less than full adoption of air conditioning), we make no firm correction but suggest that this situation may lead to additional unit sales for existing homes in that region by some unknown time in the future. That added sales volume could be significant.

The Current Industrial Report data presented earlier for the year 2004 listed approximately 7.5 million air conditioning and heat pump units sold in capacities of 5 tons and under. About 6.6 million of those (~90%) were split systems (assumed predominantly residential) and the remaining 10% were single-package units primarily used in commercial buildings. Researchers can use other Census data on annual housing stocks by year, to estimate the relative percentage of each region's installed units of each type and size. Assuming no major changes in regional shares since the 2001 EIR data, these percentages can then be used to calculate the relative sales per region for the CIR figures. The authors confirm those changes in the table below (note that West Texas is excluded due to lack of data).

Table 38. Housing Unit Increases by State, 2001-2004

	July 1, 2004 (A)	July 1, 2001 (B)	Δ% (A/B-1.00)
United States	122,671,734	117,871,387	**4.1%**
Arizona	2,458,231	2,265,420	8.5%
Colorado	2,010,806	1,875,162	7.2%
Idaho	578,774	541,368	6.9%
Montana	423,262	415,604	1.8%
Nevada	976,446	866,265	2.7%
New Mexico	825,540	794,394	3.9%
Utah	848,737	790,085	7.4%
Wyoming	232,637	225,961	3.0%
Mountain total	8,354,433	7,774,259	**7.5%**
California	12,804,702	12,369,318	**3.5%**
Ratio of Mtn/US increase			**1.033**
Ratio of CA/US increase			**0.994**

Source: http://www.census.gov/popest/housing/HU-EST2004.html

This data indicates that the 2001-2004 increase in housing units is about 1 million homes or 5% for the combined California/Mountain states. This rate of increase is very similar (5% compared to 4.1%) between those states and the U.S. as a whole. Some Mountain states increased in housing units much faster than the national average, but for all the Mountain states and California together, the difference from the national rate is very small and no adjustment is needed.

An adjustment factor is needed to represent the current trend from 2004 to 2007, the earliest year in which the HVAC technology could be commercialized. We note that in early 2006, HVAC&R News reported 2005 air conditioner factory shipments—not sales —at a record level of 8.6 million units (a 16% increase over 2004, the previous record), suggesting that this study's market potential estimates should be factored upward if that trend were expected to continue. However, this is unlikely because there was little change in housing starts between 2004 and 2005, and starts are slowing in 2006 due to rising interest rates. At least some of that record 2005 production is likely to have been due to distributor and dealer stockpiling of SEER 10-12 units in anticipation of the SEER 13 production mandate that went into effect in January 2006, and the likelihood that SEER 13 production would begin slowly, leading to a temporary shortage in early 2006. Such stockpiling suggests that some <SEER 13 units shipped in 2005 would actually reflect a part of the 2006 installations. The authors therefore reject a special

adjustment for 2005 and instead continue the previous slower upward trend in annual sales (annualized 2001-04 three-year increase).

Together these adjustment factors combine to adjust the available data as follows:

Table 39. Adjustment Factors and Resulting Estimated Annual Unit Sales by Region, 2005

	Source	U.S. Total (MM)	California	Mountain +WTexas
2001 Existing Residential a/c + hp units	EIA	57.5	3.9	2.6+0.3
2001-04 Adjustment Factor	Census	1.041	1.035	1.075
2004 Installed Residential a/c + hp		**59.9**	**4.0**	**3.12**
2004-07 Est. Adjustment Factor	extrap	1.041	1.035	1.075
2007 Est. Installed Residential Units		**62.4**	**4.18**	**3.35**
Regional percentages per above		100%	6.7%	5.4%
2004 sales of residential units, a/c + hp (national only)	CIR	**6.56**		
2004-07 Est. Adjustment Factor	extrap	1.041		
Estimated sales of residential 2007 units, a/c + hp (national only)		**6.83**		
Derived residential sales by region	proport.		**0.46**	**0.37**
3 ton units (AC + HP)		**1.50**		
By region (w/percentages above)			**0.10**	**0.08**

Source: BKI

The authors conclude that total 2007 hot-dry region sales potential for residential central air conditioners and heat pumps is approximately 460,000 + 370,000 = 830,000 units. Of that total, 3-ton units account for an estimated 100,000 + 80,000 = 180,000 units or about 23%.

Estimation of Commercial Sector HDAC Sales Potential

For a rough estimate of commercial-sector sales potential, the authors can refer to the same hot-dry region market percentage estimates gathered for the residential market and use them with the CIR sales data for the commercial sector, as in the following table.

Table 40. Estimated Annual Commercial Units Sales by Region

	Source	U.S. Total, MM	California	Mountain & WTexas
2004 sales of commercial units, a/c + hp (national only)	CIR	0.816		
2004-07 Est. Adjustment Factor	extrap	1.041		
Estimated sales of commercial 2007 units, a/c + hp		**0.849**		
Regional percentages per above table		100%	6.7%	5.4%
Derived commercial sales by region	proport.		**0.057**	**0.046**
5 ton units (AC + HP)	CIR	**0.190**		
By region			**0.013**	**0.010**

Source: BKI

The authors conclude that total 2007 hot-dry region sales potential for commercial central air conditioners and heat pumps is approximately 57,000 + 46,000 units = ~ 100,000 units. Of that total, 5-ton units account for an estimated 13,000 + 10,000 units or about 23%.

Retrofit versus New Construction

An estimate of air conditioning installations in new residential construction can be derived from available data. The principal data source is the Census Bureau's "Characteristics of New Housing for 2005" which can be found in the official website at (http://www.census.gov/const/www/charindex.html)

That source, however, considers the entire Western region as a single unit (including the hot-dry areas as well as the coastal states, Alaska, and Hawaii, but excluding Texas). The authors use other sources to disaggregate the 2005 housing data.

Per the 2005 Census report on new housing, the total single-family home construction total for the West was 437,000 in 2005. About 317,000 of those had air conditioning (73%-in contrast with the West's existing housing stock with only 41% with air conditioning). For multi-unit buildings, in 2005, the Western States new construction total was approximately 69,000 housing units, of which 55,000 had air conditioning (80. Of the 69,000 total units built, only about 8,000 units were in buildings of under 5 units, which is our cutoff for assumed use of conventional split system air conditioners and heat pumps. Researchers can assume that the same 73% air conditioning of single-family homes applies to those small multi-unit buildings, yielding about 6,000 new units with air conditioning. Summing these and the single-family total, we estimate approximately 323,000 new units built with air conditioning out of 445,000 built in 2005.

Table 33. Derivation of Residential New Construction Market Size

	Source	U.S. Total, (000)	West	California	Mountain & WTexas
2005 residential new construction, units In single family homes	Census	1636	437		
In 2-4 unit buildings		37	8		
With air conditioning, per cent		89%	73%		
Housing stock shares by target region			100%	53.7%	31.1%
Est. 2005 residential new const w/ac				**235**	**~144***
2005-07 adjustment factor				1.023	1.050
Est. 2007 residential new const w/ac				**240**	**151**

*136 Mountain plus West Texas estimated at 1% of stock = 8 (000)

Source: BKI

This analysis results in a hot-dry area new construction estimate of 240,000 + 151,000 = ~390,000 new homes with air conditioning built in 2005. This compares with the 830,000 units estimated earlier as the total residential central air conditioner sales for 2007. The authors conclude that despite this heavy activity in new home construction in recent years, the air conditioner market in the hot/dry target area is driven about equally by replacement versus new construction installations.

Contractor Opinion on Prospective HDAC Acceptance

A small sample of HVAC equipment distributors and contractors whose service territory includes hot/dry climate regions were interviewed to gain additional insight into the market forces that could influence the introduction of HDAC units to the residential and small commercial air conditioner market. While the information was largely subjective based on the experiences of the representatives of the contractors and distributors, some valuable implications could be drawn from their input:

- Respondents felt that most residential customers' decisions are driven largely by cost concerns. Although hard data are lacking, the comments received from dealers and distributors suggests that most buyers choose the least expensive up-front option for their home's needs. The vast majority of units sold and installed for all settings have been models with the minimum cost and efficiency allowed by pre-2006 standards.
- Contractors, however, indicated that a select segment of customers are driven by the desire to buy "better" or "best" units. Respondents generally estimated this segment at about 10%, although it may be increasing due to environmental concerns. In future years, the growing consumer awareness and concerns about indoor air quality and other issues with overall home performance may also increase the market share of premium units.

- There was general agreement that small commercial customers' purchasing decisions are driven almost exclusively by cost considerations, but they are more receptive than residential buyers to potential savings over the life of installed equipment.
- Contractors interviewed agreed that there are no significant barriers to their support for introduction of an HDAC unit provided it delivers promised energy savings, it performs as well or better than standard units at providing comfort and cooling in normal conditions, it is no more difficult to specify and obtain, and it is sold and supported by major manufacturers just like conventional units.
- Installed costs of units vary significantly among retailers, and are driven by factors other than unit cost from the distributor. A growing number of HVAC contractors, particularly among those serving residential customers, are beginning to offer duct sealing and other "quality installation" measures as value-added services. These installations increase the effective efficiency of units, but drive up the installation costs significantly. As energy costs continue to rise, these energy-saving features—such as the HDAC configuration—are likely to become more and more accepted in the marketplace.
- Respondents agreed that the most appropriate position for HDAC would be as an alternative to the minimum SEER-13 units. As these minimum-efficiency units will continue to make up the majority of total air conditioner sales, positioning HDAC in this segment would not limit the potential market through the inherently small potential pool of sales represented by higher-efficiency premium units. It would also take full advantage of the unit's favorable performance and deliver energy savings in line with what has been demonstrated in the initial stages of this project.

Potential Market-Wide HDAC Impacts

As illustrated in this market assessment, the market for residential and small commercial HVAC units in hot/dry regions is substantial. If positioned appropriately against standard SEER-13 air conditioners in the hot/dry regions of the western United States, HDACs could have significant market penetration. If unit costs, pricing, energy savings, and the resulting life-cycle cost advantages are as estimated in this project's initial prototype studies (by Proctor Engineering Group), residential and small commercial customers could see a positive economic savings over the life of both the split and rooftop unit types. Appropriate utility-based incentives could increase that advantage and yield even more market success.

5.2.2. Potential Future Market Characteristics

New construction sales represent significant opportunities for larger scale tract home purchases (and resulting energy impacts) given appropriate utility incentives. The authors note that with the high rates of home building in recent years, new construction has been growing faster than retrofits in these Western states, and despite the current downturn, this trend will likely be re-established because of demographic trends. More people are moving to these areas and more housing will be needed.

The replacement market must be seen as an equally, if not greater, potential source of future HDAC sales and energy savings. Even in the construction boom of the 2001-2005 period, annual air conditioner and heat pump replacements were estimated to be greater than new home installations. If the current major reduction in home building continues, the replacement market will become dominant.

The commercial construction market's split between replacement and new construction may be similar, largely due to the commercial infrastructure being expanded to meet the needs of the region's rapidly growing population and housing. However, the authors find no reliable data to support that supposition. A more common expectation is two-thirds replacement, one-third new construction. This suggests a range of 33,000-47,000 units per year in new construction by 2007, of the total estimated 100,000 units to be sold, with replacement installations accounting for 53,000-67,000 units that year. Such new construction sales rates of course depend on continued population growth and local economic conditions.

Finally, the authors note that differences in market saturation and cooling degree days across the hot-dry region are likely to affect the relative marketability of HDAC units in different states. In the desert and mountain states, temperatures are higher and current market saturation is lower than in most of California, so potential energy savings and hence cost-effectiveness are likely to be greater in those areas.

5.3. Market-Related Activities Undertaken

5.3.1. Original Plans and Final Activities

At the start of this project, tentatively anticipated activities included:

- Targeted distribution of press releases and information packages to trade journals and professional groups,
- Displays and/or papers at one or more trade or professional conferences,
- Consultations and submittals to key codes and standards bodies such as Title 24/2008
- Direct in-person educational outreach to other selected technology adoption intermediaries and decision-makers such as the DOE Emerging Technologies and Appliances groups, California utilities, International Facilities Management Association, Southwest Energy Efficiency Alliance (SWEEP), Building Codes Assistance Project (BECAP), US Green Building Council, Consortium for Energy Efficiency (CEE), Emerging Technology Coordinating Council, American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE), Air Conditioning Contractors Association, American Council for an Energy Efficient Economy (ACEEE), and major potential users such as commercial property managers and homebuilders and their appropriate trade associations and professional groups.

Examples were to include building regional market demand through voluntary efficiency program requirements, national and regional market transformation initiatives, publishing and presentations to key groups such as CEE (e.g., supporting market transformation) and US DOE (e.g., recommending appliance standards changes pertinent to the HDAC), and working with Air Conditioning and Refrigeration Institute (ARI), Edison Electric Institute , BCAP and SWEEP to develop a regional initiative for the HDAC product. The information to be conveyed would include publication and presentation of project findings, market research, impact assessment and suggested rating standards and efficiency levels to utilities, market transformation organizations, and state agencies as prioritized in consultation with the Commission's contract manager.

5.3.2. Western Utilities Support Development

Once the decision was made by the project team to conduct HDAC field tests with western utilities, the Market Connection team initiated development of two communications for utility field test participation recruiting. The first was a business case proposition to the utilities that described the reasons for participation in the field tests, the HDAC specification, the potential energy and demand savings benefits, and customer benefits. A key feature of participation was that participating companies would have to pay for the entire research themselves. In addition to the business case piece, the team also developed an HDAC project summary that included: 1) a suggested memorandum of understanding and participation agreement that outlined the commitments and responsibilities of both the HDAC project team and the participating utilities, and 2) a suggested field test monitoring protocol that was modeled from the protocol recommended by John Proctor for his HDAC contract with PG&E.

Through existing relationships and several cold calls, contact was initiated by the Market Connection team with the following utilities for the purpose of inviting field test participation:

- Arizona Public Service (AZ)
- Avista Utilities (WA)
- Bonneville Power Administration (ID, MT, OR, WA)
- Idaho Power (ID, OR)
- NW Energy Efficiency Alliance (ID, MT, OR, WA)
- Nevada Power Company (NV)
- Pacific Power/Utah-Rocky Mountain Power (CA, OR, UT, WA)
- Public Service Company of New Mexico (NM)
- Roseville Municipal (CA)
- Sacramento Municipal Utility District (CA)
- Salt River Project (AZ)
- UniSource Energy/Tucson Electric Power (AZ)
- Xcel Energy (CO, NM)

Ultimately, in addition to PG&E and SCE, only Nevada Power (now integrated with Sierra Pacific) was able to participate in the field tests although all utilities solicited expressed intentions to offer special incentives and marketing to support commercial introduction of the HDAC. Nevada was already engaged in testing a number of air conditioning systems. Its R&D contractor, Paragon Consulting, was able to fit the HDAC unit testing into the existing test program. Each of the other potential utility participants had reasons for not joining in the HDAC field testing process. Examples include the following:

- Arizona Public Service and UniSource Energy had already allocated their program and R&D budgets for 2006.
- Sacramento Municipal Utility District was undergoing a personnel shortage in the R&D area.
- Roseville could only participate if Sacramento took the lead on project management and shared the direct costs.
- Salt River Project identified a commercial test site, but both RTU manufacturing partners (Lennox and Trane) were unable to provide the required three-phase unit in time.
- Bonneville and the Alliance staffs recommended working directly through their utility customers in hot dry areas east of the Cascade mountains. There were no small publicly owned utilities with the technical or financial capacity to participate at this stage.
- Idaho Power was not convinced of the benefits versus the cost of participation. The authors tried unsuccessfully to put Idaho Power and Rocky Mountain Power together for a field test in their contiguous service area.
- Avista determined that its hot, dry service area was too sparsely populated to make it worthwhile.
- Rocky Mountain Power (formerly Utah Power) was going through a buyout. New programs were not being initiated.
- Public Service of New Mexico was just gearing up for a new DSM push and was occupied with other activities at the time.
- Xcel Energy was not responsive to communications from the team.

5.3.3. Manufacturing Industry-Wide Outreach

In parallel with the manufacturer and utility field test recruiting effort, the Market Connection team approached the Air Conditioning and Refrigeration Institute (ARI), the national industry membership organization, team to present a webcast on the HDAC project (and other PIER HVAC work). ARI agreed to host the webcast for its Unitary Large and Unitary Small Equipment Committees. Fifteen of the committee members attended the hour-long webcast in December 2005. The presentation was part of the initial manufacturer recruiting process. There was little follow up from committee members following the webcast. One manufacturer, American Standard, participated in the webcast and in the field test. All participating manufacturers were contacted as part of the field test recruiting.

Based on advice received from ARI staff, the Market Connections team refocused its attention to the individual manufacturers. ARI was not going take an active role in the project on behalf of its members.

In addition to the manufacturers and ARI, the project team contacted the Air Conditioner Contractors of America (ACCA) to inform them about the project and request assistance in supporting the project goals among its membership and with the manufacturers. Although ACCA staff were not able to provide specific support, they were able to generally spread the word about the project through their national network of contractors.

5.3.4. Manufacturer Business Case Coordination

Manufacturer Business Case Development

The project team developed a recruiting packet to be sent to manufacturers. The packet included the 12/05 HDAC Component and Initial Modeling Summary Report by Proctor Engineering Group and a business case document with a similar look and feel to the utility business case document. Both are included in the Appendices to this report.

This business case was developed early in the project and focused on the original new-hardware strategy. It demonstrated market strength and benefit/cost advantages of the HDAC approach, making the case for adoption of technology improvements. As noted earlier in this report's outline of the Market Connection "Phase 1" strategy, this approach quickly evolved into a focus on field personnel training in proper hot-dry climate system specification. Manufacturers were more favorably disposed to this approach, and it led to active participation in the field testing stage by several major manufacturers.

The following HVAC manufacturers were initially contacted for potential interest in participating in the HDAC field test:

- AAON
- Carrier Corporation
- Goettl Air Conditioning
- Goodman Manufacturing Company
- Lennox Industries
- National Comfort Products
- Nordyne Inc.
- Rheem
- Trane/American Standard

- York

Initially, the program recruited Goettl Air Conditioning, a small manufacturer based in Tempe, Arizona. Goettl had a reputation for innovation and was familiar with optimizing cooling systems for the hot, dry climate of Arizona. As the project progressed, the project team determined that the project should try to recruit larger manufacturers if we were to potentially impact a much larger share of the hot dry market.

AAON determined that their equipment already met the HDAC specification. AAON tests its units at 1” of static pressure compared with the HDAC specification at 0.5” and had no compelling reason to join the field tests. AAON considers itself a “manufacturer’s manufacturer” and prefers to pursue the market for larger units in the 20-50 ton range.

After attempted repeated contact with the other manufacturers noted above, only Lennox and Trane chose to formally participate. It was our understanding that Trane’s primary interest was in the potential for increasing market share in California. Lennox was interested in participating to see if its products could meet the specification and provide more efficient cooling.

Lennox was able to provide a 3-ton split system and a 5-ton package rooftop unit with single-phase wiring. Trane provided two 3-ton split systems. Trane personnel noted that the company expected to come to market with a SEER 14 RTU later and preferred to wait, anticipating that this unit might come close to the HDAC specification.

Project team members had extensive contact with Lennox through its Utility Marketing Manager, Julie Humes. Her assistance was invaluable in getting through the various stages of the field test process including administrative components, dealer recruitment, and technical communications with Jim Mullen, the (now retired) government relations director for Lennox. Julie was able to quickly to resolve several challenges that arose with equipment ordering and with one local Lennox dealer.

5.3.5. Market Connections Outreach

The Market Connection team contacted the Southwest Energy Efficiency Project (SWEEP) that works with state governments, regulators, all electric/gas utilities and related efficiency advocates in Arizona, Colorado, Nevada, New Mexico, Utah and Wyoming. SWEEP is known as the leading energy efficiency advocacy group in these states. SWEEP has been effective in influencing regulatory policy in support of energy efficiency and renewables in its six state service area. Market Connection staff made a presentation to the SWEEP annual meeting in Utah in February 2006. Since that time, HDAC team staff have stayed in contact with SWEEP staff to update them on the field test results and to collaborate on strategies for developing support among utilities and regulators. Hot, dry optimized cooling is firmly on SWEEP’s agenda. They will provide support for hot, dry cooling equipment standards. Once the program’s final report is available, it will be distributed by SWEEP its constituents.

As the project moved toward the end of its term, Market Connection staff brought the recently created UC Davis (CA) Western Cooling Efficiency Center (WCEC) into the strategic collaboration on climate optimized efficiency standards. WCEC's mission is to catalog and support a range of cooling strategies that in concert can significantly and cost-effectively reduce the impact of cooling systems on California's electricity grid and in the adjacent hot dry areas of the Southwestern states. WCEC staff will be working on post-project climate optimized cooling strategies, research, and supporting utility HDAC pilot program development. WCEC is the natural institutional home base from which to continue advocating for standards both in California and the southwest.

In August 2007, Market Connection staff participated in an initiative by the California Public Utilities Commission (CPUC) to generate "Big Bold" energy efficiency ideas.

The concept of climate optimized cooling was introduced. Since that meeting, the CPUC issued a proposed order that would encourage collaborative efforts at the state, regional and national levels to establish climate optimize efficiency standards, including one for hot dry climates.

Market Connection staff introduced the climate optimized standards concept in an Emerging Technologies Brainstorm in April 2006 and did the same related in an April 2007, 2011 Title 24 brainstorming session. Both sessions were hosted by Southern California Edison.

In May 2007, staff participated in the California Energy Commission 2011 Title 24 Standards development workshop to support climate optimized standards for consideration in upcoming Title 24 revisions. Other workshop participants strongly supported this approach.

In 2007, at the ASHRAE Annual meeting in Quebec, Market Connection team staff proposed a seminar on *Climate Optimized Cooling: Reality and Future Directions* for the January 2008 ASHRAE Winter meeting in New York City. The seminar proposal was brought to ASHRAE Technical Committee 6.3 Central Forced Air Heating and Cooling and Technical Committee 9.5 Residential and Small Commercial Building Applications, for their review and vote of support. The proposal was voted top priority by TC6.3. TC9.5 signed on as a co-sponsor. Team staff organized the seminar, recruited participants and will moderate the seminar whose participants include:

- Hugh Henderson, CDH Energy

  Energy Efficiency for Residential Space Conditioning in Hot Humid Climates

- John Proctor, Proctor Engineering

  Energy Efficiency for Residential and Commercial Space Conditioning in Hot Dry Climates

- Harvey Sachs, American Council for an Energy Efficient Economy (ACEEE)

  Robust Residential Air-Conditioner Standards for the Next Quarter Century

Due to a large number of seminar proposals submitted to the Winter Meeting, ASHRAE has deferred a number of them, including the climate optimized seminar, to the June 2008, Annual Meeting in Salt Lake City. Market Connection team staff participation, as previously planned at the seminar, is one of the recommended post-project activities.

Market Connection staff participated in the Consortium for Energy Efficiency (CEE) Industry Partners meeting in St. Louis in September 2007 and was able to speak about the HDAC project to CEE utility members and HVAC industry government relations and engineering staff.

At the national level, Market Connection staff have collaborated with the staff of ACEEE on federal legislation that would allow the U.S. Department of Energy to establish two or three climate optimized efficiency standards for air conditioners. The legislation is strongly opposed by ARI on behalf of its members. The outcome of the federal legislative effort is uncertain.

5.3.6. Post-Project Activity Plans

As noted above, post-project activities are in planning stages including:

The HDAC project team plans to work with WCEC on moving the climate optimized concept forward in California and in the hot dry southwest states. As now planned, WCEC will lead efforts in California with the CPUC to further expand the development and application of a hot dry cooling standard for the state and outreach to neighboring states. WCEC will be the liaison with SWEEP on promoting the climate optimized concept to its six state utility and regulatory network. WCEC has begun working with the Western Governor's Association on regionwide cooling related issues. WCEC may also play a role in working with US DOE on establishing a hot dry efficiency standard. WCEC is currently assessing the requirements for establishing a utility-sponsored pilot program for HDAC units prior to the 2008 cooling season. The project will require utility funding.

The proposed ASHRAE seminar on climate optimized cooling is expected to be held at the ASHRAE Annual Meeting in June 2008 in Salt Lake City.

Work continues on supporting federal legislation that would allow the U.S. Department of Energy to establish climate optimized standards. The legislation only “allows”, but does not require, U.S. DOE to act. Ongoing support to get DOE to act will be required.

6.0 Conclusions and Recommendations

6.1. Conclusions

Air conditioners manufactured to perform equal to or better than the two proof-of-concept HDAC units would be cost effective based on life cycle costs as long as they are not marked up as premium machines.

The HDAC Project was successful in producing two units that provided the target energy savings and peak reductions.

The project's laboratory tests proved that air conditioners, when designed with hot dry climates in consideration, can exceed the following HDAC specification.

The combination of the furnace, outside unit, and indoor coil meets the following criteria.

Table 42. HDAC Performance Specifications

Condition	Hot Medium 115/80/67
External Static Press. (IWC)	$(\frac{CFM\ per\,ton}{495\,CFM\ \ per\ ton})^2$
Gross Sensible Capacity (sensible btuh)	65% or greater than the gross total capacity at ARI test A (95/80/67)
Net Sensible EER	at least 6.8

Source: Proctor Engineering Group

Notes:

1. Airflow rate across the evaporator coil is determined by the manufacturer.
2. The External Static Pressure from the return plenum to the supply plenum downstream of the evaporator coil is defined by the relationship $(\frac{CFM\ per\,ton}{495\,CFM\ \ per\ ton})^2$. An air conditioner system (furnace, outside unit, and evaporator coil) tested at 400 cfm per ton would be tested at 0.653 IWC.
3. Net Sensible EER is the net sensible capacity divided by the total unit watt draw.

The HDAC Project illustrated that there are very significant additional gains available for dry climates using an extended evaporator fan run time at the end of the compressor cycle.

Given the issues associated with peak electric energy consumption, the installation of a standard SEER 13 unit is a lost opportunity.

Given the degraded performance of 410A machines at high temperatures when compared to R-22 machines, the installation of 410A machines without a specification on their performance at high temperatures is both detrimental compared to R-22 and a lost opportunity.

6.2. Recommendations

Given the energy and peak demand benefits of HDAC units clearly demonstrated as a result of this project, efforts to encourage the adoption of this technology should be made in California and throughout hot dry climate states. The recommendations of this report are as follows:

1) A California Public Utilities Commission decision to focus attention on climate optimized standards at the regional and national level. This public declaration of interest will signal to regulators in the Southwest that this is an issue with region-wide interest that will require region-wide collaboration. In addition, it signals to the U.S. Department of Energy that California regulators and utilities would be willing to cooperate with DOE should it receive the go ahead from Congress to develop regional climate optimized standards.

2) The inclusion of climate optimized efficiency in the development process for the 2011 Title 24 Standards revision process. There was strong support of the need to optimize cooling equipment and designs for a variety of climates in California. Hot dry optimization was the dominant theme. This is the first step for the consideration of climate optimized efficiency standards in Title 24

3) Collaboration with energy efficiency advocates in Washington, D.C. on federal legislation that would allow U.S. DOE to establish climate optimized efficiency standards for two climates, hot dry and hot humid. The very fact that California was investigating this potential both in the laboratory and in field tests, provided sufficient value to substantiate the benefits of proposing the development of federal regional standards.

4) The positive results of the project thus far, along with the additional efficiency potential from a Phase 2 effort, when coupled with the results of other PIER CEC work in HVAC including the Advanced Rooftop Unit and Fault Detection and Diagnostics, and the results of a PIER-sponsored national HVAC Roundtable in 2006, have provided the basis for discussions on the need for a National Cooling Initiative to tackle multiple issues with compressor-based cooling equipment and the potential for further efficiency improvements that would be available across all market segments.

The results of the Market Connection efforts of this project are only partial, due to a variety of evolutionary steps and delays in the overall project. Those led to the ending of the project before some of the key steps could be taken in commercialization and technology advancement. In addition to follow up on the key outcomes noted above, we summarize the following key market connection related steps still needed for full realization of the HDAC concept's potential:

- The Western Cooling Efficiency Center (WCEC) at UC Davis should be supported in its interest in working with HDAC team members to complete the Phase 1 steps needed (listed below) to introduce this innovation.

- A system for qualifying evaporator coils and complete indoor units should be developed, preferably requiring manufacturers to test and submit qualifying data for their products and using an impartial laboratory to review and spot-test submittals.
- Manufacturers should be strongly encouraged by utilities, regulators, and advocacy groups as needed to implement the Phase 1 approach of providing guidance to their distributors and dealers on proper HDAC unit specification. The WCEC may be an appropriate source for this effort, including use of specialists to organize support and help manufacturers to carry out the required skill improvements in both specification and marketing.
- The Emerging Technologies staff groups in the major California utilities, allied with PIER, should consider joint support of the further development of Phase 2 HDAC innovations and their integration into practical units for manufacturing. The needed activities include development, lab testing, and alliances with interested manufacturers to create and field test advanced HDAC units and update the business case for that technology.
- An intensive collaborative effort is recommended to induce the air conditioning industry's adoption of HDAC-related innovations into the next generation of products for home and small commercial uses. This would involve alliances with State and Federal agencies, utilities, regulators, advocacy groups, and manufacturers as well as key industry associations, standards-setting authorities, and possibly key legislative leaders.
- Specific follow up work is required to support the CPUC interest in collaborating regionally and nationally on climate optimized standards. WCEC is in a strong position to conduct this support along with utility backing. Additional information on technology and cost will be needed for the CPUC from both Phase 1 and Phase 2.
- The potential for the Consortium for Energy Efficiency to bring this issue to its membership as an initiative should be explored. Typically, CEE chooses issues that potentially impact its utility members nationally rather than on regional interests or other more specific issues. It is possible, however that its members might see the value in supporting the overall climate optimized standards concept. WCEC may be the appropriate organizational lead on this with the direct backing of CEE utility members from California.

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