Public Interest Energy Research (PIER) Program
FINAL PROJECT REPORT

EFFICIENCY CHARACTERISTICS AND OPPORTUNITIES FOR NEW CALIFORNIA HOMES (ECO)

Prepared for: California Energy Commission
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Rick Chitwood of Chitwood Energy Management, Inc. performed the field research for the Efficiency Characteristics and Opportunities for New California Homes project. John Proctor of Proctor Engineering Group, Ltd., analyzed the data obtained from the field research and developed this report. Bruce Wilcox served as project manager. The research team acknowledges the support of the California Energy Commission PIER Program, as well as the significant support provided by Pacific Gas and Electric Company, Southern California Edison, and Sempra Utilities to carry out additional research as part of this project. The research team would also like to thank the homeowners and residents who participated in this study.

PREFACE

The California Energy Commission 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
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- Environmentally Preferred Advanced Generation
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Transportation

Efficiency Characteristics and Opportunities for New California Homes is the final report for the Efficiency Characteristics and Opportunities for New California Homes project (Contract Number PIR-08-019) conducted by Bruce A. Wilcox, P.E. , Rick Chitwood of Chitwood Energy Management, Inc., and John Proctor, P.E. of Proctor Engineering Group, Ltd. The information from this project contributes to PIER's Buildings End-Use Energy Efficiency Program.

When the source of a table, figure or photo is not otherwise credited, it is the work of the author of the report.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-327-1551.

ABSTRACT

Efficiency Characteristics and Opportunities for New California Homes was a research project of the California Energy Commission's Public Interest Energy Research (PIER) Program. Pacific Gas and Electric Company (PG&E), Southern California Edison, and Sempra Utilities were also major contributors to this study.

The project goal was to reduce end-use energy consumption and peak electrical demand in California by improving the California Building Energy Efficiency Standards for Residential Buildings (Title 24). Phase One focused on gathering data, taking measurements, and testing performance to develop a baseline data set that would support more accurate lifecycle cost and energy savings calculations. During field visits to 80 newly constructed single-family and multifamily California homes, lighting type information was collected, and a series of tests and measurements including heating and cooling system (HVAC) performance, and building air leakage were performed. During Phase Two additional cooling system tests were performed and the impact of making simple upgrades to nine of the field study homes was analyzed.

Phase One results showed that 78 percent of the lighting wattage in single-family and multifamily homes was incandescent, and that average air conditioner performed well below expectations. HVAC tests revealed multiple problems, including low sensible capacity and efficiency, high return static pressures, refrigerant charge and thermostatic expansion valve problems, and potential problems with non-condensables. These problems were particularly severe in zoned systems and combined hydronic systems. Single-family homes were found to be reasonably airtight, but 51 percent of the leakage area was between the attic and occupied space for residences with attached garages and accessible attics.

Phase Two upgrades on nine HVAC units resulted in an average efficiency improvement of 24 percent. The project produced 16 recommendations for improvement to Title 24 Standards.

Keywords: California Energy Commission, HVAC, duct leakage, Title 24, zoned systems, refrigerant non-condensables, house air leakage, blower door test procedures, furnace fans, lighting, California new construction, leakage from attic, leakage from garage, evaporator airflow

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EXECUTIVE SUMMARY

Introduction

The Efficiency Characteristics and Opportunities for New California Homes (ECO) project surveyed selected energy efficiency measures in 80 single-family and multifamily homes built under the 2005 Building Energy Efficiency Standards for Residential Buildings. The project developed a baseline data set to support more accurate life cycle cost and energy savings calculations for new and existing efficiency measures. The goal of the project was to reduce end-use energy consumption and peak electrical demand in California by improving the standards.

The project built upon previous 2006 research for the 2008 Standards (Contract 500-04-006) that carried out field research on heating, ventilation, and air-conditioning (HVAC) system characteristics. That research led to the furnace fan watt draw and airflow requirements that were adopted in the 2008 Standards. This study provides additional depth to the 2006 research and provides random sample results and includes multifamily buildings in addition to single-family houses.

Along with the California Energy Commission, Pacific Gas and Electric Company (PG&E), Southern California Edison, and Sempra Utilities were major contributors to this study. Through their 2011 Title 24 Codes and Standards Enhancement (CASE) project, the three utilities expanded the field survey activities in the following areas:

Residential CASE 1	Air leakage testing and fireplace air leakage testing.
Residential CASE 7	Zoned air conditioner (AC) efficiency including measured efficiency changes between different methods of zoning.
Residential CASE 12	Duct testing including leakage and component-by-component duct pressure drops.

Approach

At the outset, the ECO project produced a detailed survey instrument and pilot tested it in two homes. Data were measured using applicable Alternative Calculation Method (ACM) procedures and included lighting, HVAC characteristics and performance, simple, low-cost HVAC improvements, fireplace air leakage, and house air leakage.

In Phase One of the ECO project, 80 newly constructed homes were recruited from the electricity customers of Pacific Gas and Electric, Southern California Edison, and San Diego Gas & Electric. Forty single-family and 40 multifamily homes first connected to the electric grid in 2007 were randomly recruited. The numbers of participants were proportional to the number of new units in the utility customer lists stratified by three digit ZIP code. Field visits were conducted including a census of lighting types, multiple tests on the HVAC system, and multiple tests of building air leakage.

In Phase Two, additional cooling system tests and simple HVAC upgrades were performed on 10 of the single- family homes.

Findings

Lighting

Seventy-eight percent of the lighting wattage in single-family homes and town houses were incandescent. In apartments, 68 percent of the wattage was in incandescent lamps. The majority of the lamp wattage was controlled by switches while dimmers controlled 10 percent of the wattage in apartments and 33 percent of the wattage in single-family homes.

HVAC Phase One

The predominant heating and cooling system (HVAC) in apartments was a combined hydronic coil from the water heater and an evaporator coil from a split air conditioner. The predominant system in single-family homes and townhomes was a split system air conditioner with a gas furnace.

The average air conditioner performed well below expectations with low airflow across the indoor coils averaging 322 Cubic Feet per Minute (CFM) per ton of cooling capacity. The 10 combined hydronic units had an average airflow of only 280 CFM per ton. Airflow across the indoor coil is a statistically significant predictor of the sensible efficiency of air conditioning systems. On the units in this sample, an increase of 100 CFM per ton would translate to a 14 percent increase in sensible cooling capacity.

The split system air conditioner evaporator blowers drew an average 650 watts per 1000 CFM of airflow, which is 178 percent of the watt draw assumed in the Seasonal Energy Efficiency Ratio (SEER) test procedure. Underestimating fan energy use is a clear deficiency of the federal SEER procedure. Zoned HVAC systems were the largest offenders, drawing 206 percent of the SEER assumed fan wattage with all dampers open and 233 percent of the assumed fan wattage with only the main zone operating.

Only 28 percent of the systems tested met the 2008 California Title 24 Standards for cooling airflow and fan power. The predominant cause of low airflow in these units was excessively high return system static pressure (including the filter).

Low airflow was also a problem in the heating mode. Low furnace airflow can also cause the furnace to cycle off and on due to the high temperature limit switch, potentially increasing heat exchanger fatigue and corrosion.

Thermostatic expansion valves are used to control the flow of refrigerant in an air conditioner, providing a nearly constant temperature difference between the refrigerant entering the indoor coil and the refrigerant exiting the coil. Title 24 provides a liberal requirement that this temperature difference be between 4°F and 25°F. Thirty-one percent of the units tested failed this criterion, indicating problems with the Thermostatic eXpansion Valves (TXV) and/or refrigerant charge or flow restrictions. Additional testing on units which passed the above test indicated at least another 12 percent of the units indicated too little or too much refrigerant.

Seventy-eight percent of the ducted systems had some or all of the ducts in the attic. This location provides the most severe case for conduction losses and return leakage problems. The median duct leakage for single-family homes met the Title 24 prescriptive standard, but townhomes and apartments showed higher leakage rates to outside the housing units.

Duct leakage causes three problems: conditioned supply air loss, return air dilution (often with attic air), and additional house infiltration. The effect of this infiltration, caused by a difference in leakage between supply and return ducts, has been underestimated in past Title 24 calculations. A new duct leakage imbalance value is recommended based on this study.

Since the Federal Test Standard classifies the cabinet around the furnace blower as part of the duct system (not part of the furnace), and this is not required to be insulated by federal standards, the majority of the blower cabinets are not insulated, causing excessive heat gain in the summer and heat loss in the winter. It should be clear in the Title 24 Standards that this part of the system is required to be insulated by the installer or manufacturer.

HVAC Phase Two

Repairs/upgrades on the nine initially operating units in Phase Two resulted in an average efficiency improvement of 24 percent.

The most common repair was reducing the flow resistance of the return duct system between the house and the furnace/air conditioner. This repair, which often involved increasing the size of the return duct, adding additional return ducts, increasing the size of the filter box, and replacing restrictive filters with less restrictive filters, had the greatest effect on improving efficiency.

When the refrigerant was removed and replaced with clean, pure refrigerant, the efficiency of one unit increased by 19 percent.. This efficiency improvement probably indicates that non-condensables, most likely air, were contaminating the refrigerant. The efficiency of another unit increased by 35 percent when the existing refrigerant was removed and proper refrigerant volume installed. Two out of seven of the units in this sample (29 percent) are judged to have had contaminated refrigerant.

The efficiency of the zoned unit in this sample was increased by 17 percent when the zoning bypass was eliminated.

In one case replacing a permanent split capacitor (PSC) fan motor with a more efficient brushless permanent magnet (BPM) fan motor adjusted to the same airflow dropped power usage by 102 watts and increased efficiency by 4 percent.

Fireplace Air Leakage

Fireplaces in single-family homes produced a range of air leakage to outside between less than 2 percent of the house leakage to 18 percent of house leakage. There were 23 fireplaces in the single-family homes; almost half of those were responsible for between 7 percent and 18 percent of the house's total leakage area.

House Air Leakage

A variety of house leakage test methods were compared. The study concludes that measuring leakage at a single point with the house pressurized to 50 pascals using a blower door provides results within 5 percent of the other methods.

The median of single-family homes were found to be reasonably tight at 4.66 Air Changes per Hour at a pressurization of 50 pascals (4.66 ACH50). The leakage to outside the units for apartments and townhomes was significantly higher (apartment median 6.02 ACH50, town house median 6.42 ACH50).

The residences in this study that have both attached garages and accessible attics, on average have 51 percent of the leakage area between the conditioned space and the attic. These residences also have an average of 11 percent leakage between the garage and conditioned space.

Recommendations

The study team has developed the following recommendations as a result of this study:

1. The 2013 revision of the Title 24, part 6, residential efficiency standard (Title 24–2013) should make mandatory a confirmed airflow greater than or equal to 400 CFM per ton and a fan watt draw less than or equal to 0.510 watts per CFM; with an acceptable alternative of the return system sizes specified in Table 25, as verified by the building inspector.

2. Title 24–2013 should mandate labeling HVAC return locations with the size and maximum clean filter pressure drop at 400 CFM per ton clean filter airflow.

3. Title 24–2013 should mandate that all HVAC filters sold in California be labeled with a standardized clean filter pressure drop and clean filter airflow table.

4. Title 24–2013 should mandate a confirmed total duct leakage of less than or equal to 24 Cubic Feet per Minute at a pressurization of 25 pascals (CFM25) per ton for single-family homes and town homes.

5. Title 24–2013 should mandate a confirmed total duct leakage of less than or equal to 48 CFM25 per ton for apartments regardless of the location of the duct systems.

6. Title 24–2013 ACM should calculate energy consumption based on 17 percent duct leakage imbalance.

7. Title 24–2013 ACM should calculate energy consumption based on 51 percent of the house air leakage area between the occupied space and the attic.
8. Title 24–2013 should clearly define the fan cabinet and return plenum on furnaces as part of the duct system and specify that it must be insulated to the levels specified for duct systems in the space in which they are located.
9. Title 24–2013 should revise the acceptable limits for California Home Energy Rating System (HERS) inspections of Thermostatic Expansion Valves (TXV) in air conditioners. The limits should be greater than 2° F and less than or equal to the manufacturer's target subcooling of 8°F.
10. The California Energy Commission should sponsor additional field research to determine the extent of non-condensables in the refrigerant of newly installed air conditioners.
11. Title 24–2013 should mandate that any zoned system must not have a bypass from the supply to the return and that the airflow in all potential operating modes meet the airflow specification of Recommendation number 1.
12. For single-family buildings and town houses, Title 24–2013 should mandate a confirmed building shell air leakage of less than or equal to 4 ACH at 50 pascals using a single point test.
13. For multifamily buildings, Title 24–2013 should mandate a confirmed unit air leakage of less than or equal to 6 ACH at 50 pascals using a single-point test.
14. Title 24–2013 should mandate that air conditioner condensing units not be placed within 5 ft. of a dryer vent.
15. Title 24–2013 should mandate that there be no obstruction within 5 ft of the condenser coil inlet and condenser coil outlet.
16. Title 24–2013 should mandate that furnace heat rise must not exceed the manufacturer's specification.

CHAPTER 1: Introduction

1.1 Background

The Efficiency Characteristics and Opportunities for New California Homes project (ECO) surveyed selected energy efficiency measures in 80 single and multifamily homes built under the 2005 Building Energy Efficiency Standards for Residential Buildings (Standards) to provide a baseline data set. This data set was developed to support more accurate life cycle cost and energy savings calculations for new and existing efficiency measures. The ultimate goal of the project was to reduce end-use energy consumption and peak electrical demand in California by improving the Standards.

The project built on 2006 research for the 2008 Standards (contract 500-04-006) that carried out field research on heating, ventilation and air conditioning (HVAC) system characteristics. That research led to the furnace fan watt draw and airflow requirements adopted in the 2008 Standards. This study provides additional depth to the 2006 research by using a random sample of single-family and multifamily homes.

1.2 Utility CASE Initiatives

Along with the California Energy Commission, Pacific Gas and Electric Company (PG&E), Southern California Edison, and Sempra Utilities were major contributors to this study. Through their 2011 Title 24 Codes and Standards Enhancement (CASE) project, the three utilities expanded the field survey activities in the following areas:

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1.3 Project Summary

The project produced a detailed onsite survey instrument and pilot tested its application in two homes. Data were measured using applicable Alternative Calculation Method (ACM) procedures and included lighting, HVAC characteristics and performance, simple low-cost HVAC improvements, fireplace air leakage, and house air leakage.

1.3.1 Recruiting

The first phase of the ECO project recruited 80 newly constructed homes from the electricity customers of PG&E, Southern California Edison, and San Diego Gas and Electric Company. Forty single-family and 40 multifamily homes first connected to the electric grid in 2007 were randomly recruited. The numbers of participants were proportional to the number of new units in the utility customer lists stratified by three digit zip code. Field visits were conducted to gather a census of lighting, perform multiple tests on the HVAC system, and conduct multiple tests of building air leakage.

In a second phase of the ECO project, additional cooling system tests were performed and simple HVAC upgrades were made to 10 of the single-family homes. The energy usage impact of these upgrades was evaluated and the resulting data was used to recommend modifications to the Standards.

1.3.2 Lighting

The lighting census provided researchers with information about many previously unknown statistics on actual residential homes. The resulting data set is available to researchers. Seventy-eight percent of the lighting wattage lamps in single-family and town houses were incandescent. In apartments, 68 percent of the wattage was in incandescent lamps. The majority of the lamp wattages were controlled by switches while dimmers controlled 10percent of the wattage in apartments and 33 percent of the wattage in single-family homes.

1.3.4 HVAC Phase One

The predominant heating and cooling system (HVAC) in apartments was a combined hydronic coil from the water heater and an evaporator coil from a split air conditioner. The predominant system in single-family homes and town homes was a split system air conditioner with a gas furnace.

The average air conditioner performed well below expectations with low airflow across the indoor coils averaging 322 cubic feet per minute (CFM) per ton of cooling capacity. The 10 combined hydronic units had an average airflow of only 280 CFM per ton. Airflow across the indoor coil is a statistically significant predictor of the sensible efficiency of air conditioning systems. On the units in this sample, an increase of 100 CFM per ton would translate to a 14percent increase in sensible cooling capacity.

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Thermostatic expansion valves (TXVs) are used to control the flow of refrigerant in an air conditioner, providing a nearly constant temperature difference between the refrigerant entering the indoor coil and the refrigerant exiting the coil. Title 24 provides a liberal requirement that this temperature difference be between 4°F and 25°F. Thirty-one percent of the units tested failed this criterion, indicating problems with the Thermostatic Expansion Valves and/or refrigerant charge or flow restrictions. At least another 12percent of the units indicated too little or too much refrigerant.

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Duct leakage causes three problems: conditioned supply air loss, return air dilution (often with attic air) and additional house infiltration. The effect of this infiltration has been underestimated in past Title 24 calculations. A new duct leakage imbalance value is recommended based on this study.

Since the Federal Test Standard classifies the cabinet around the furnace blower as part of the duct system (not part of the furnace), the majority of the blower cabinets are not insulated, causing excessive heat gain in the summer and heat loss in the winter. This should be rectified in the Title 24 Standards.

1.3.5 HVAC Phase Two

Repairs/upgrades on the nine units in Phase Two resulted in an average efficiency improvement of 24 percent.

The most common and successful repair was reducing the flow resistance of the return duct system between the house and the furnace/air conditioner.

When the refrigerant was removed and replaced with clean, pure refrigerant, the efficiency of one unit increased by 19 percent. This efficiency improvement indicates that non-condensables were probably contaminating the refrigerant. The efficiency of another unit increased by 35 percent when the existing refrigerant was removed and proper refrigerant volume installed. Two out of seven of the units in this sample (29 percent) are judged to have had contaminated refrigerant.

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Replacing a permanent split capacitor (PSC) fan motor with a brushless permanent magnet (BPM) fan motor adjusted to the same airflow dropped power by 102 watts and increased efficiency by 4percent.

1.3.6 Fireplace Air Leakage

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1.3.7 House Air Leakage

A variety of house leakage test methods were compared. The study concludes that a single point method at 50 pascals provides results within 5 percent of the other methods.

The median of single-family homes were found to be reasonably tight (4.66 ACH50). The leakage to outside the units for apartments and townhomes was significantly higher (apartment median 6.02 ACH50, townhouse median 6.42 ACH50).

The residences in this study that had both attached garages and accessible attics, on average had 51 percent of the leakage area between the conditioned space and the attic. These residences also had an average of 11 percent leakage between the garage and conditioned space.

2. CHAPTER 2: Methods

The project began with recruiting participants and scheduling field visits to test the performance of the participants' homes. Field visits included a census of lighting for each home, multiple tests on the heating and air conditioning system, and multiple tests of the building air leakage performance. The data acquisition forms are reproduced in Appendices A and B.

Data for Phase One of this project were collected between 9/25/09 and 1/23/10.

Phase Two of this project gathered additional cooling system data on 10 of the single-family houses during the summer of 2010.

2.1 Recruiting

Lists of residences first connected to the electric grid in 2007 were obtained from the California Investor Owned Utilities. These lists included 5000 residences in Southern California Edison's service area, 5000 residences in Pacific Gas and Electric Company's service area, and 4060 residences in San Diego Gas and Electric's service area. These lists were separated between detached single-family residences and attached multifamily residences. The lists were randomized and target participation was determined for representation proportional to the newly constructed units in each three digit zip code.

Potential participants were mailed invitations and offered three methods of response: prepaid mail, website, and toll-free phone call. Potential participants were offered $100 for their participation. Respondents by those three methods were contacted by phone and the field visits were scheduled. The process maintained the stratification by building type and three digit zip code.

2.2 Lighting

A direct observation census of lamps was taken at each residence. The lamps were classified by illumination type: Incandescent, Fluorescent, Compact Fluorescent, Halogen, or Light Emitting Diode. The lamps were also classified by location in the house, wattage, hardwired vs. portable, and control type: switched, dimmer, and occupancy sensor.

2.3 HVAC

The heating and air conditioning systems and their air distribution systems were tested. Each system had the following parameter and performance measurements:

- Make, model, and output rating of the furnace, condensing unit, and indoor coil
- Air circulation blower type, watt draw, power factor, and speeds in cooling and heating
- Furnace standby watts, watts with induced draft blower on, and watts with gas valve on

- Heating airflow at the furnace as well as furnace inlet, furnace outlet, and cooling coil outlet static pressures
- Cooling airflow at the furnace as well as furnace inlet, furnace outlet, and cooling coil outlet static pressures
- Constant on fan flow and power consumption
- Air filter type, size and pressure drop
- Air conditioner condenser unit watts, amp draw, voltage, as well as nameplate fan full load amps (FLA) and compressor rated load amps (RLA)
- Air conditioner condenser air entering temperature, saturation temperature (from high side pressure) and liquid line temperature
- Air conditioner evaporator saturation temperature (from low side pressure at the outdoor unit) and suction line temperature at the outdoor unit
- Air conditioner return plenum wet and dry bulb temperatures as well as supply plenum dry bulb temperatures
- Cooling air delivery dry bulb temperature and flow from each supply grille as well as return grille temperatures in the same time period
- Duct leakage at 25 pascals (0.10 IWC) as well as supply and return static pressures with air handler on, supply registers and return grilles blocked (Half Nelson test).

2.3.1 Instrumentation

Table 1displays the instrumentation used for the heating and cooling system measurements.

Table 1: HVAC Instrumentation

Measurement	Device
Watts, Amps, Voltage, Power Factor	Extech 380940 meter
Airflow at Furnace in CFM	Energy Conservatory TrueFlow Plates with DG 700 Meter
Static Pressures	Energy Conservatory DG 700
Air Dry Bulb Temperatures	Low mass type K thermocouples with Fluke 52-II meter
Air Wet Bulb Temperatures	Low mass thermocouples with wetted cotton sleeve
Refrigerant Line Temperatures	Insulated low mass type K thermocouples with JBdigital pressure manifold and temperature gauge
Duct Leakage	Energy Conservatory Duct Blaster
Refrigerant Weight	Mastercool Accucharge II98210A Scale

Source: Data gathered by Rick Chitwood

2.4 Building Shell

The building shells were tested for the following parameters:

- Fireplace leakage to outside using Duct Blaster and Blower Door Method
- Building shell leakage using Single Point Depressurization at 50 pascals
- Building shell leakage with range hoods and fans sealed using Single Point Pressurization at 50 pascals
- Building shell leakage using ASTM E779-03 (automated, both pressurized and depressurized)
- Building shell leakage using ASTM 1827-02 (five tests depressurized) with the following measurements:
    - Building shell leakage
    - Garage pressure
    - House and garage leakage
    - Attic pressure

2.4.1 Instrumentation

Table 2 displays the instrumentation used for the building shell measurements. These instruments were newly calibrated by the manufacturer.

Table 2: Building Shell Leakage Instrumentation

Measurement	Device
Building Shell Leakage	Energy Conservatory Blower Door with DG 700 Meter or APT
Static Pressures	Energy Conservatory DG 700 Meter

CHAPTER 3: Results

3.1 Recruiting and Building Characteristics

3.1.1 Phase One

The project successfully recruited and measured 80 residences that were first occupied in 2007. The breakdown of occupancies by climate zone is shown in Table 3.

Table 3: Building Configuration by Energy Commission Climate Zone

Climate Zone	Apartment	Town House	Single-Family	Total
1	0	0	0	0
2	1	0	1	2
3	3	3	2	8
4	0	2	1	3
5	0	0	0	0
6	0	2	2	4
7	1	2	1	4
8	3	4	1	8
9	2	2	2	6
10	3	2	10	15
11	2	0	3	5
12	2	0	5 + 1*	8
13	1	2	5	8
14	2	1	3	6
15	1	0	1	2

16	0	0	1	1
Total	21	20	38	80

*Single-Family Detached units include one modular home in Climate Zone 12.
Source: Data gathered by Rick Chitwood.

The Building Characteristic data for these units are displayed in Table 4.

Table 4: Building Characteristics by Building Configuration

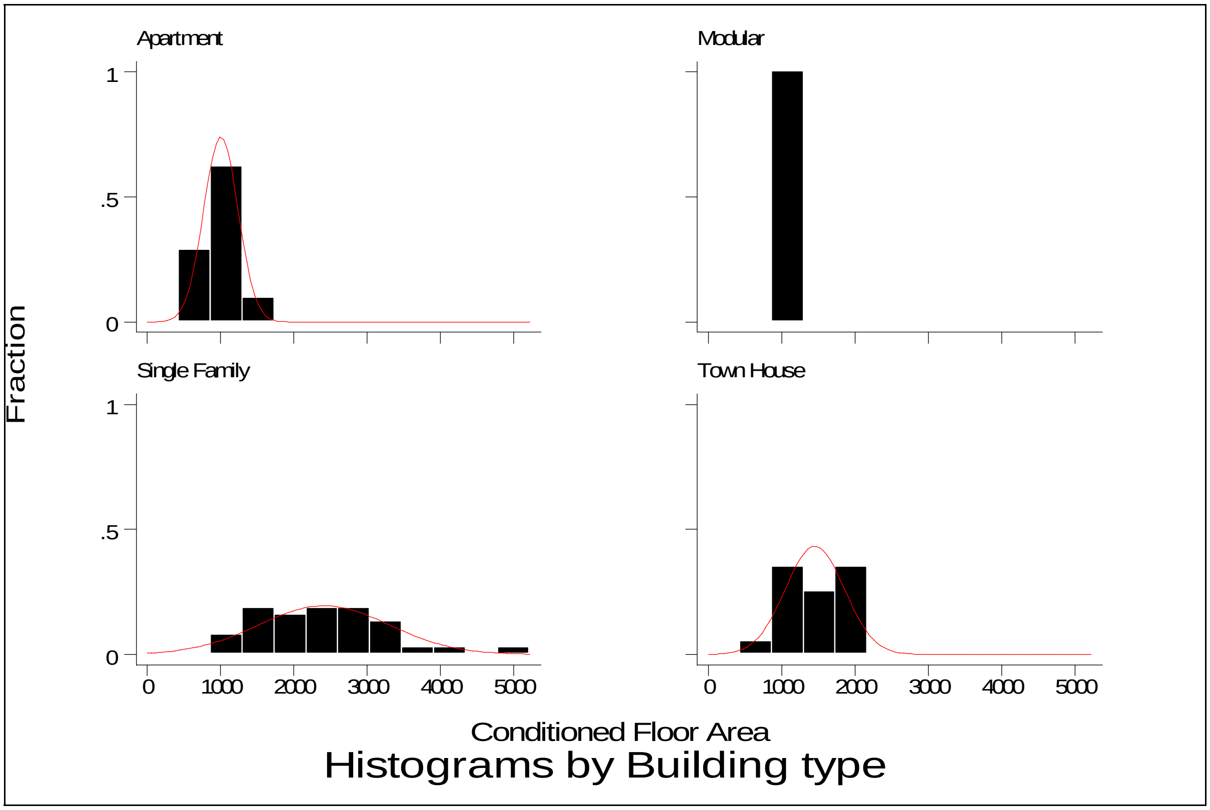
Bedrooms	**Apartment**	**Town House**	**Single-Family**	**Total**
1	3	2	0	5
2	10	6	7 + 1	24
3	6	11	12	29
4	2	1	13	16
5	0	0	5	5
6	0	0	1	1
Stories				
1	18	2	18 + 1	39
2	2	10	18	30
2.2	0	2	0	2
2.5	0	1	0	1
3	1	5	2	8
Exterior				
Board	2	1	4 + 1	8
Masonry / Cinder Block	0	1	0	1
Sheet Siding	1	0	0	1
Stucco	16	15	27	58
Stucco w/ Accent	2	2	3	7
Stucco/Board	0	1	4	5

Source: Data gathered by Rick Chitwood

The town homes and apartment above two stories are units over garages. One of the three-story single-family units has both a garage and living space on the lowest floor; the other only has a garage.

Figure 1and Table 5 show the conditioned floor areas of the project residences. The conditioned floor area of Town House is bimodal with single and two bedroom units averaging 1200 square feet and three bedroom units averaging 1600 square feet.

Figure 1: Conditioned Floor Area by Building Configuration



Source: Data gathered by Rick Chitwood

Table 5: Conditioned Floor Area by Building Configuration

Building Configuration	Mean	Standard Deviation	Frequency
Apartment	1004	235	21
Modular	1248	0	1
Single-Family	2410	890	38
Town House	1450	401	20

Source: Data gathered by Rick Chitwood

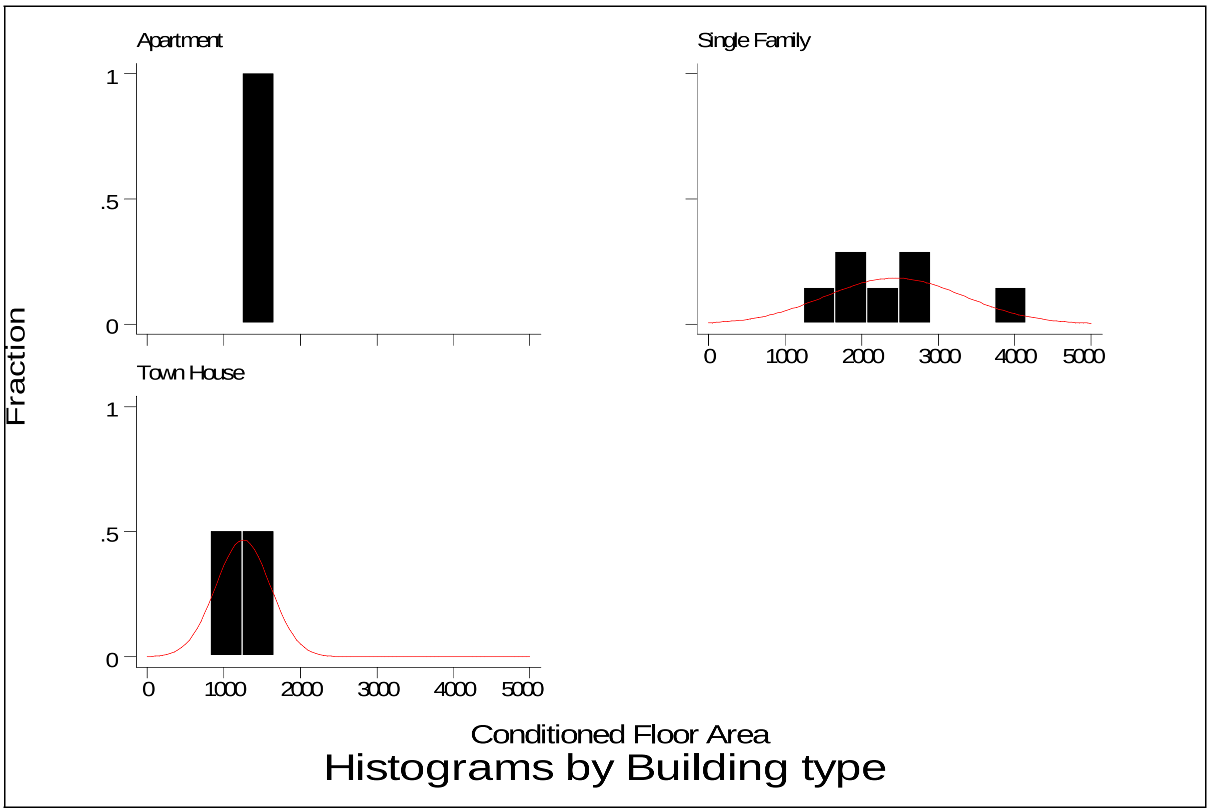
3.1.2 Phase Two

Phase Two of the project returned in the summer to 10 of the residences and performed summer condition measurements and where practical, made repairs to the HVAC systems. These units were sampled as representing the range of units found in the 40 unit survey, with particular focus on single-family units.

There were seven single-family dwellings in this group, one apartment and two town houses.

The breakdown of conditioned floor area by building configuration is shown in Figure 2 and Table 6.

Figure 2: Conditioned Floor Area by Building Configuration



Source: Data gathered by Rick Chitwood

Table 6: Conditioned Floor Area by Building Configuration

Building Configuration	Mean	Standard Deviation	Frequency
Apartment	1300	0	1
Single-Family	2434	906	7
Town House	1251	356	2

Source: Data gathered by Rick Chitwood

3.2 Lighting

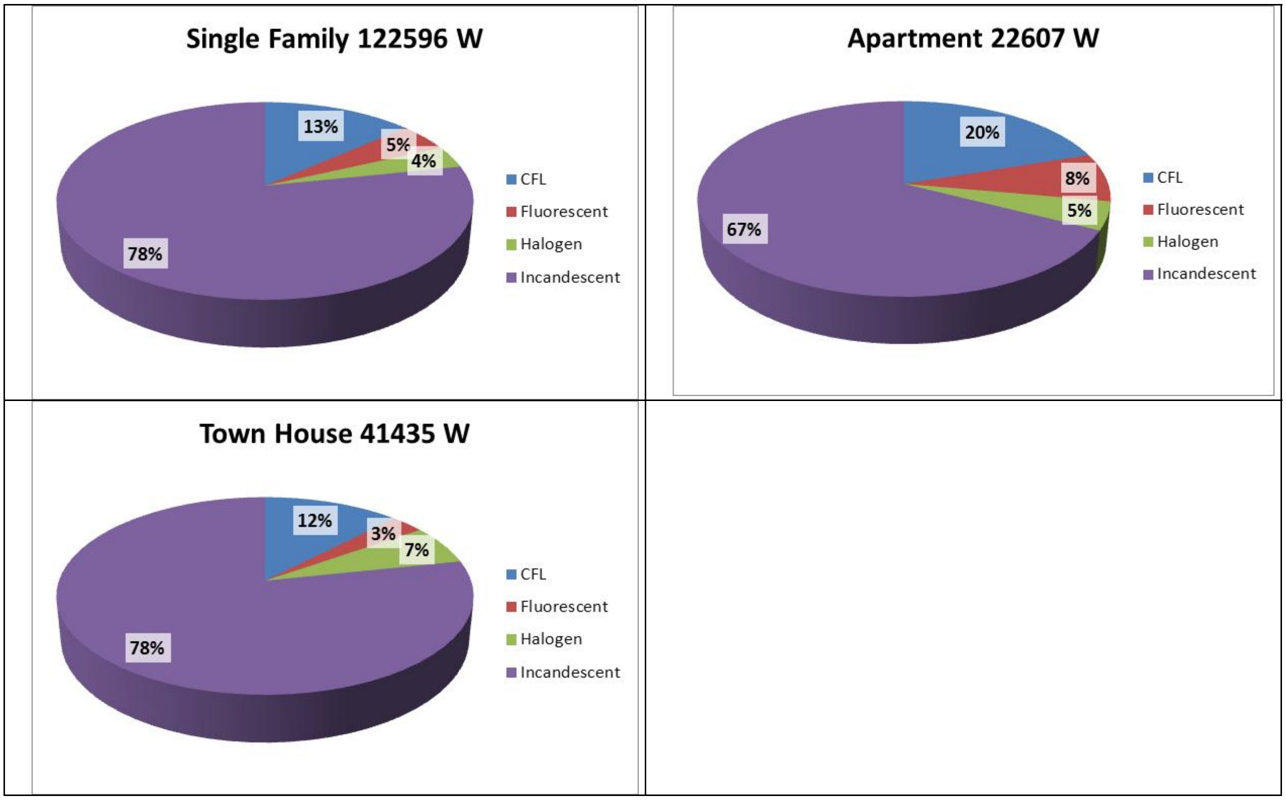
The lighting census portion of this investigation found 4,244 indoor lamps. The breakdown of lamp types is displayed in Table 7. The total wattage breakdown by lamp type and building type are displayed in Figure 3.

Table 7: Indoor Lamp Type by Building Configuration

Lamp Type	Apartment	Modular	Single-Family	Town House	Total
CFL	242	29	790	278	1339
Fluorescent Tube	63	5	221	53	342
Halogen	6	0	53	34	93
Incandescent	276	2	1638	547	2463
LED	0	0	0	7	7
Total	587	36	2702	919	4244

Source: Data gathered by Rick Chitwood

Figure 3: Total Wattage Percentage by Lamp Type by Building Configuration



Source: Data gathered by Rick Chitwood

The means and standard deviations of wattages by lamp type and building configuration are shown in Table 8.

Table 8: Mean Wattage with Standard Deviations for Single Wattage Indoor Lamps

Lamp Type	Apartment	Modular	Single-Family	Town House	Total
CFL (mean W)	19.26	15.25	20.72	18.88	19.84
(std. dev. of W)	8.16	4.67	6.57	8.32	7.40
Fluorescent Tube	27.70	36.00	26.42	24.76	26.57
	6.06	5.66	7.52	6.55	7.06
Halogen	237.50	.	106.92	92.14	114.09
	125.00	.	108.57	93.82	110.50
Incandescent	55.39	60.00	57.98	56.91	57.36
	17.99	0.00	14.77	14.75	15.28
LED	.	.	.	10.00	10.00
	.	.	.	5.77	5.77
Total	38.30	19.79	43.90	41.01	41.94
	34.30	12.54	29.67	29.51	30.52

Source: Data gathered by Rick Chitwood

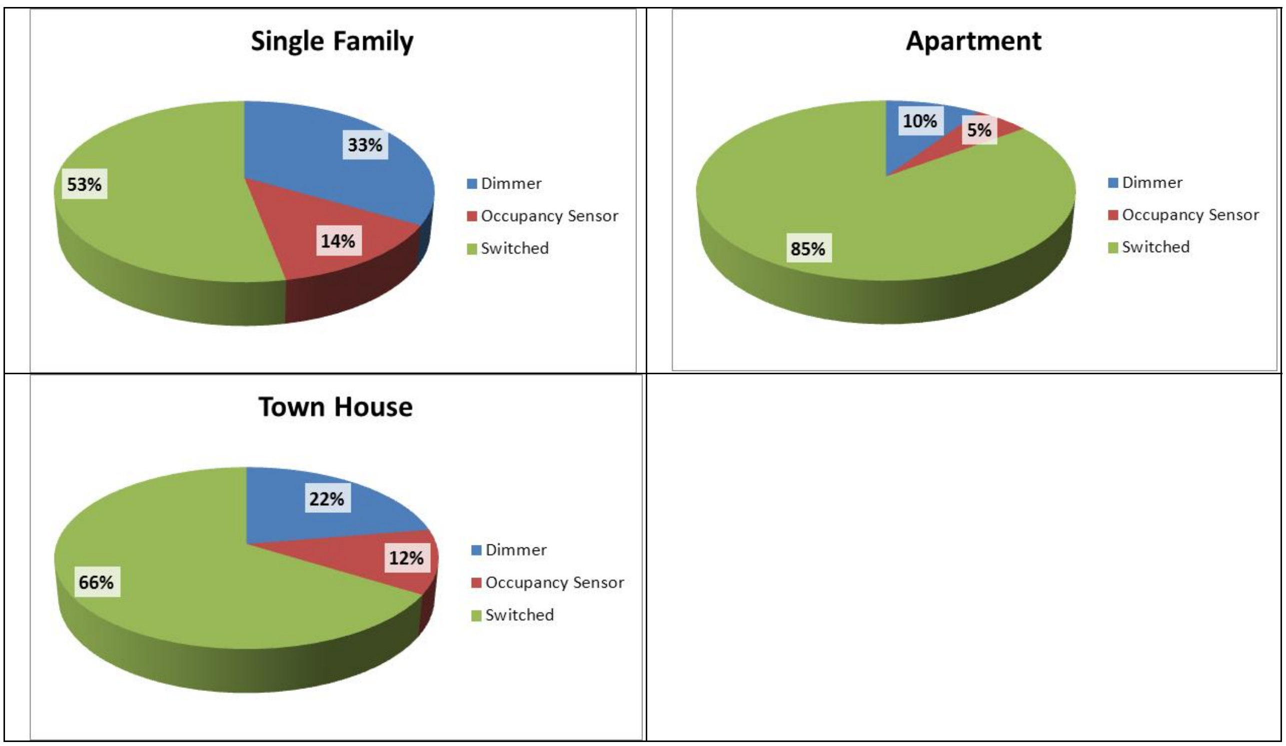
The lamps were predominantly controlled by switches, with a significant number of single-family and town homes using dimmers. The breakdown of controls by building configuration is shown in Table 9 and Figure 4.

Table 9: Indoor Lamp Controls by Building Configuration

Control	Apartment	Modular	Single-Family	Town House	Total
Dimmer	13	0	199	50	262
Occupancy Sensor	7	0	71	21	99
Switched	278	19	690	325	1312
Total	298	19	960	396	1673

Source: Data gathered by Rick Chitwood

Figure 4: Wattage Distribution by Lamp Control by Building Configuration



Source: Data gathered by Rick Chitwood

The means and standard deviations of wattages by lamp control and building configuration are shown in Table 10.

Table 10: Mean Wattage With Standard Deviations for Single Wattage Indoor Lamp Controls

Control	Apartment	Modular	Single-Family	Town House	Total
Dimmer	58.69	.	60.98	57.64	60.23
	18.25	.	33.20	10.82	29.59
Occupancy Sensor	35.14	.	54.66	57.43	53.87
	8.30	.	17.60	8.42	16.37
Switched	39.51	19.79	39.16	38.52	38.80
	37.76	12.54	29.70	32.89	32.24
Total	40.25	19.79	44.84	41.95	43.05
	36.88	12.54	31.12	30.97	32.17

Source: Data gathered by Rick Chitwood

The counts of fixture types are shown in Table 11. Eighty percent of the fixtures were hardwired.

Table 11: Fixture Types

Fixture Type	Number
Can Light	407
Ceiling Fan	122
Ceiling Fixture	292
Counter/Cabinet	42
Fan Light	68
Plug Lamp	335
Range Hood	64
Surface	34
Suspended	68
Track	4
Vanity	170
Wall Fixture	68
Total	1674

Source: Data gathered by Rick Chitwood

3.3 HVAC

3.3.1 Phase One

Phase One of the ECO project investigated 88 HVAC systems in the 80 residences. Two systems were not measured in a three-system house because of time considerations.

- Seven systems had no duct system.
- Eight systems had no cooling.
- Ten systems were zoned.

The breakdown of system types by configuration are displayed in Table 12.

Table 12: HVAC System Type by Building Configuration

System Type	Apartment	Modular	Single-Family	Town House	Total
Combined Hydronic	10	0	0	1	11
Electric Resistance W	1	0	0	0	1
Furnace Only	0	1	1	1	3
Heat Pump	4	0	0	2	6
Hydronic Floor No Cooling	1	0	2	0	3
Multiple Splits w Furnaces	0	0	17	0	17
Package Rooftop Heat	0	0	1	0	1
Package Rooftop Unit	0	0	2	3	5
Packaged Terminal AC	2	0	0	0	2
Single Split AC w/ Furnace	4	0	23	11	38
Wall Furnace No Cooling	1	0	0	0	1
Total	23	1	46	18	88

Source: Data gathered by Rick Chitwood

3.3.1.1 AC Sensible EER (EER_S)

The metric of interest in cooling for most of California is the Sensible Energy Efficiency Ratio. This metric is the result of dividing the sensible capacity by the total watt draw of the air conditioner. The sensible capacity was measured for the air conditioning systems in two locations: at the air conditioner (the sensible heat removed through the unit) and at the delivery system terminals (the sensible heat removed between the return grille or grilles and the supply grilles).

In both locations the sensible capacity is computed as:

$$SensCap_{net} = \sum_{r=1}^{n} 1.08 \times CFM_r \times DB\ Temp\,°F_r - \sum_{s=1}^{n} 1.08 \times CFM_s \times DB\ Temp\,°F_s - FanHeat$$

Where:

Subscript r identifies each return
Subscript s identifies each supply
$FanHeat(BTU/hr) = FanWatts \times 3.412$

In both locations the total watt draw is computed as:

$$Watts_{tot} = FanWatts + CompressorWatts + CondenserFanWatts$$

The sensible EER then is:

$$EER_{Sensible} = \frac{SensCap_{net}}{Watts_{tot}}$$

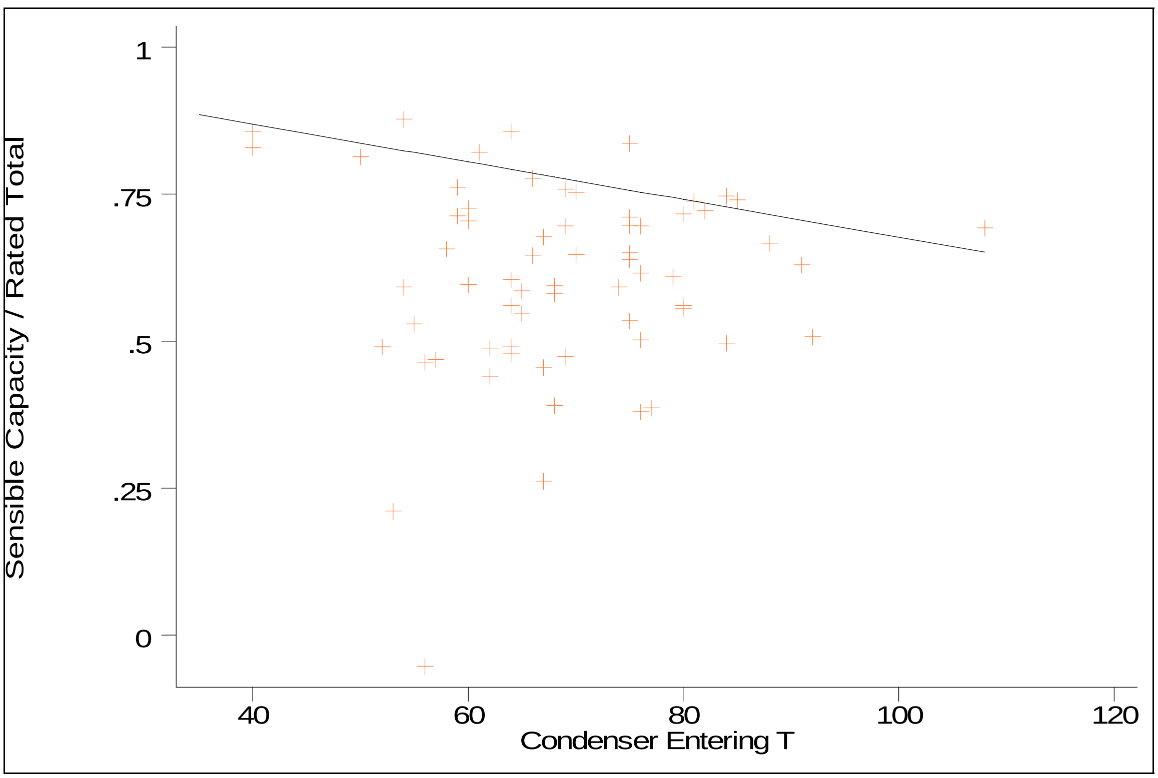
The sensible capacity is known to increase with higher airflow (CFM per ton) and with lower temperatures, as well as other environmental, installation, maintenance, and design factors.

3.3.1.2 AC Sensible Capacity

In Phase One, most of the air conditioning performance was measured at low outdoor temperatures averaging 67°F and ranging from 35°F to 108°F. As a result the sensible capacity of most these units should become a greater fraction of their rated total capacity at 95°F.

Figure 5 shows the sensible capacities measured at the unit vs. the outdoor temperature. An expected generic plot of sensible capacity against outdoor temperature for a unit operating with an 80°F return plenum temperature and 50 percent return relative humidity is also displayed for comparison.

Figure 5: Sensible Capacity vs. Outdoor Temperature



Source: Data gathered by Rick Chitwood

The wide variation in sensible capacity and the large differences from expectations are a result of factors other than the outdoor temperature. The dominant explanatory variable is the rate of airflow through the system. Units operating with THERMOSTATIC EXPANSION VALVES superheat in excess of 25°F are also significantly less efficient. The indoor condition as defined by the wet bulb depression (a measure of humidity with larger depressions indicative of drier indoor air) is a significant performance factor. A linear regression of the measured sensible capacity at the unit against measured parameters explains 66 percent of the variability in the sensible capacity.

The parameters, their coefficients, t values and 95 percent Confidence Intervals are displayed in Table 13. One unit with no capacity (0 BTU/hr) was excluded from the regression.

Table 13: Regression Values – Sensible Capacity vs. Measured Parameters

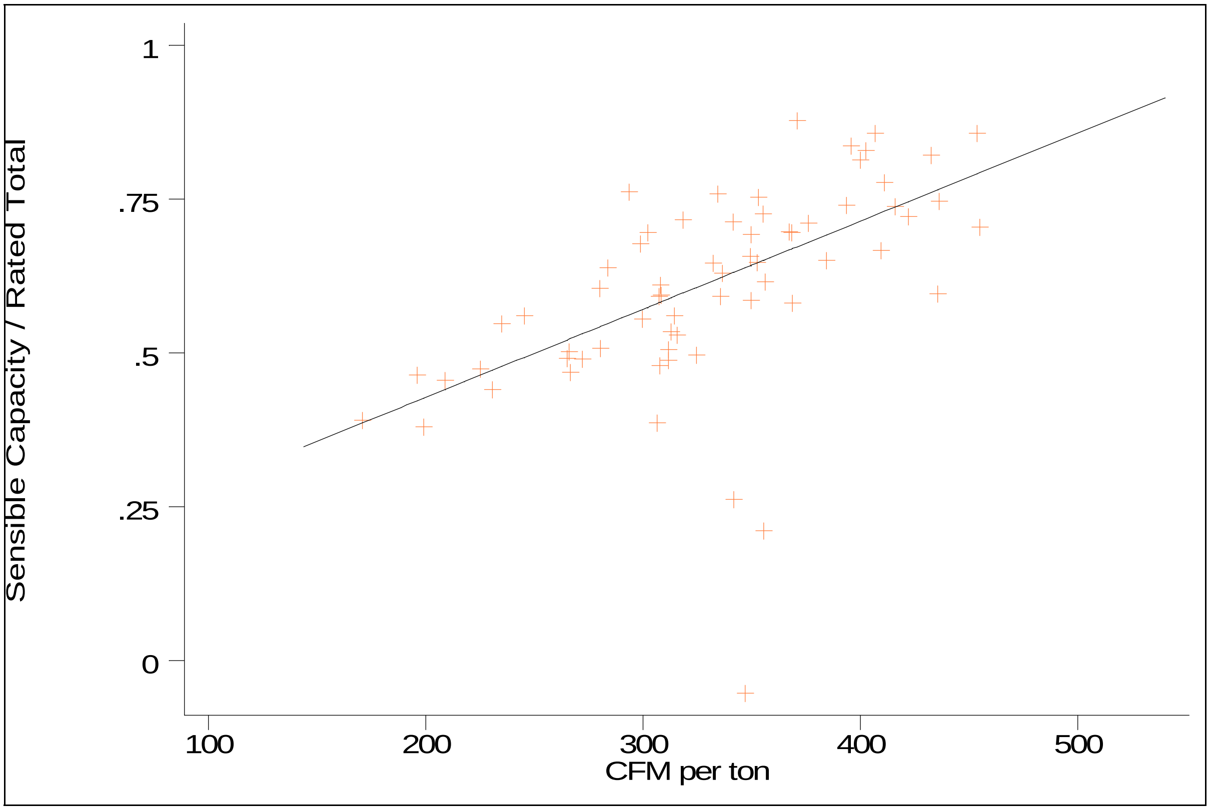
Source	**SS**	**Degrees of Freedom**	**MS**	**N**	61
				F(4, 56)	30.61
Model	0.765728	4	0.191432	Prob > F	0
Residual	0.350193	56	0.006253	R-squared	0.6862
			Adj R-squared		0.6638
Total	1.115921	60	0.018599	Root MSE	0.07908
Sensible Capacity per Rated (95°F) Total Capacity	**Coefficient**	**t**	**P>t**	**[95% Conf.Interval]**	
Condenser Air Entering Temperature(°F)	-0.00187	-2.243	0.029	-0.00354	-0.0002
Wet Bulb depression (°F)	0.017147	4.255	0	0.009074	0.02522
CFM per ton	0.001383	8.687	0	0.001064	0.001702
Thermostatic expansion valves superheat > 25°F (0/1 value)	0.044466	1.957	0.055	-0.00105	0.089977
Constant	0.052064	0.612	0.543	-0.11824	0.222367

Source: Data – Rick Chitwood; Calculations – Proctor Engineering Group, Ltd.

3.3.1.3 AC System Airflow

The effect of system airflow on sensible capacity is statistically significant at the .001 level. The relationship is shown graphically in Figure 6.

Figure 6: Sensible Capacity vs. System Airflow



Source: Data gathered by Rick Chitwood

The breakdown in system airflow in CFM per ton and watts per CFM are displayed in Table 14.

Table 14: Mean System Airflow (CFM/ton) by System Type and Duct (Zoned/Not Zoned) System

System Type	Mean CFM per ton	Mean W per CFM	N
Combined Hydronic	280	0.579	10
Heat Pump	262	0.495	5
Multiple Splits w Furnaces	363	0.515	17
Package Rooftop Heat Pump	393	0.356	1
Package Rooftop Unit	333	0.545	1
Single Split AC w Furnace	321	0.650	45
Total	322	0.597	79

Source: Data gathered by Rick Chitwood

The presence of dampered zones has a significant effect on the airflow and watt draw per CFM of the air conditioner. The breakdown of cooling system airflow and watt draw vs. the presence of automatically dampered zones is displayed in Table 15.

Table 15: Cooling Airflow and Watts per CFM by Duct Zoning Type

Zoning	Mean CFM per ton	Mean W per CFM	N
No Powered Dampers	352	0.530711	51
All Dampers Powered Open	292	0.750924	10
Main Zone Only Damper Open	253	0.849678	8

Source: Data gathered by Rick Chitwood

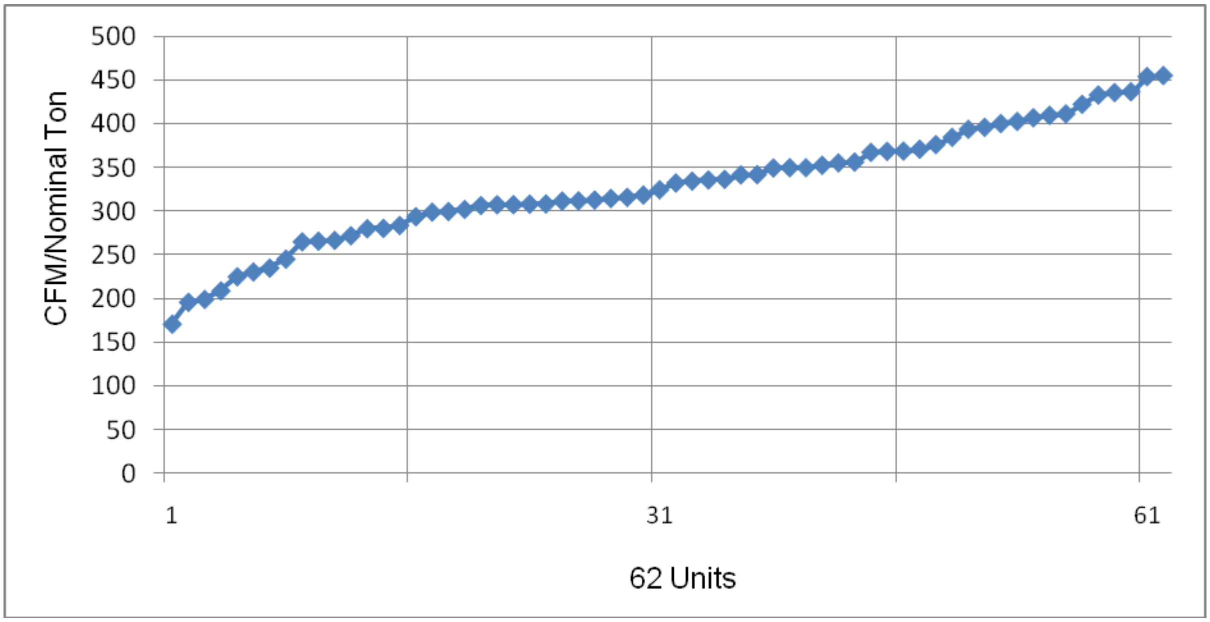
The mean system airflow of the systems without automatic zone dampers was 352 CFM per ton.[1] This is similar to the results from the 2006 study and other field studies (Proctor and Parker, "Hidden Power Drains: Residential Heating & Cooling Fan Power Demand," Proceedings of ACEEE 2000 Conference).

The 10 systems zoned with automatic dampers had significantly lower flow through the system.

The cooling airflow and the power used to achieve that flow through the distribution system are shown in Figure 7 and Figure 8.

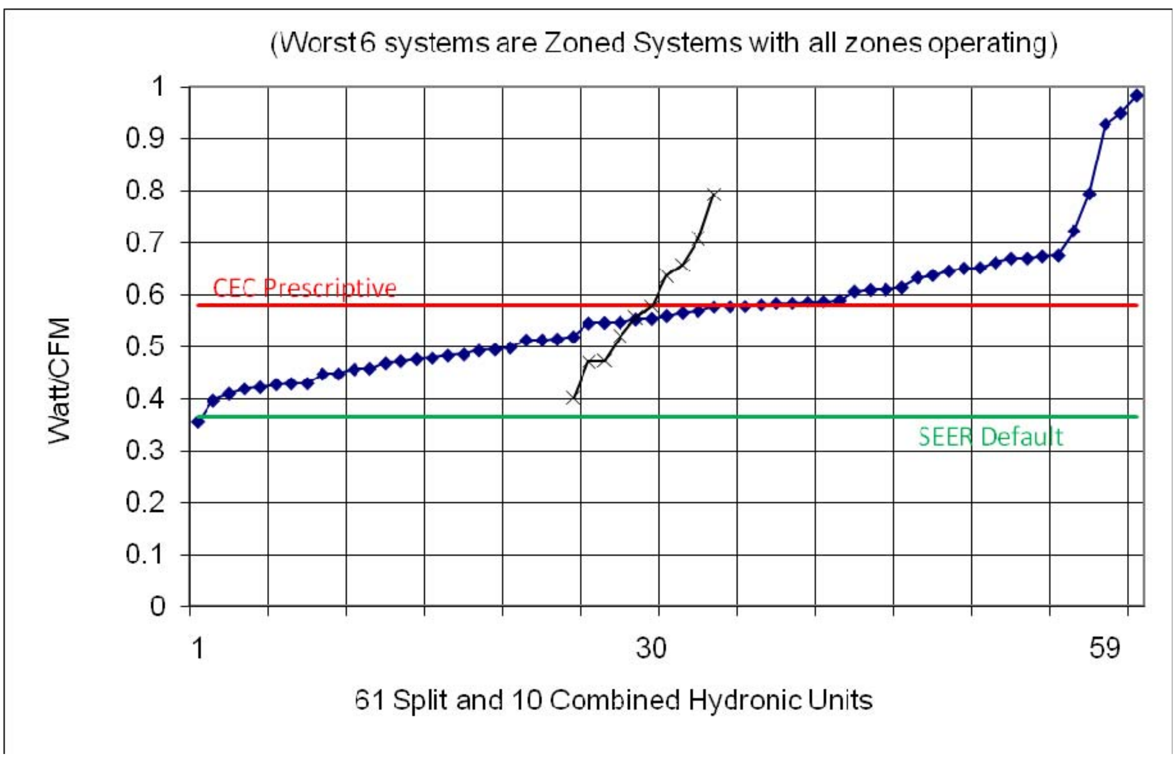
[1]Combined hydronic systems excluded.

Figure 7: Cooling System Airflow



Source: Rick Chitwood

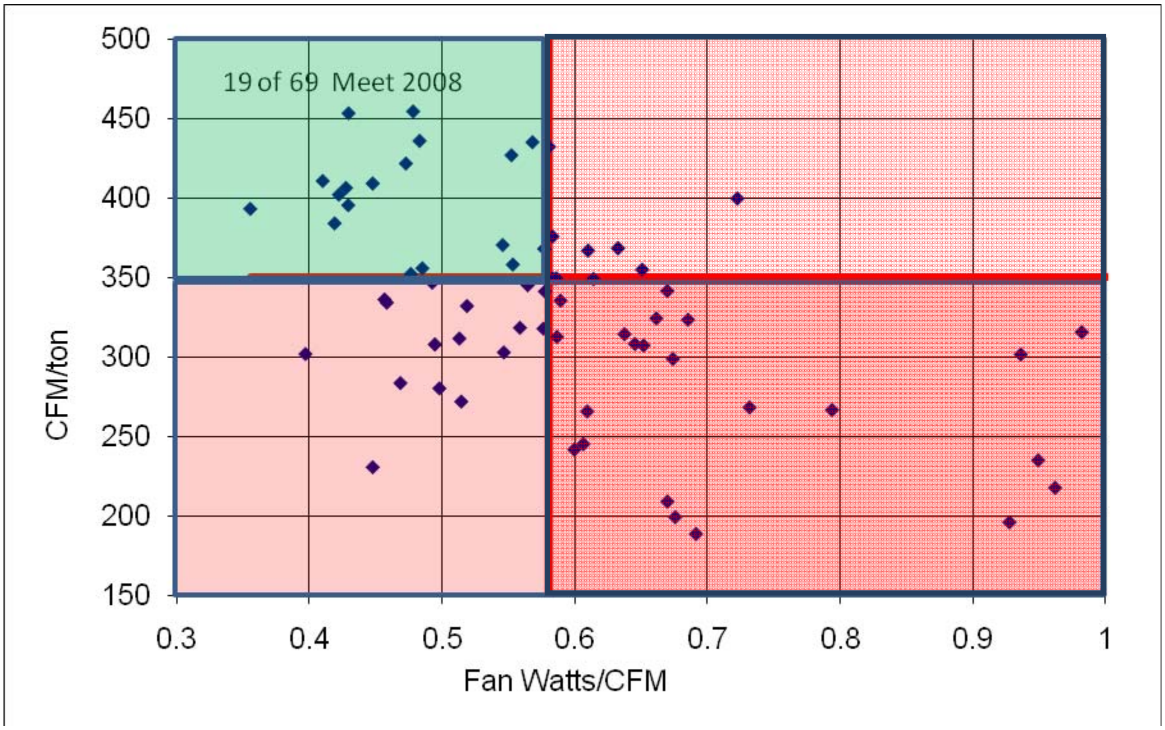
Figure 8: Cooling System Circulating Fan Power



Source: Rick Chitwood

Title 24–2008 provides a prescriptive standard for cooling airflow and fan power. Figure 9 shows that only 19 of 69 systems tested (28 percent) meet this standard. Potential causes of the failure to meet this standard are discussed in the Air Distribution section.

Figure 9: Cooling Airflow and Fan Power

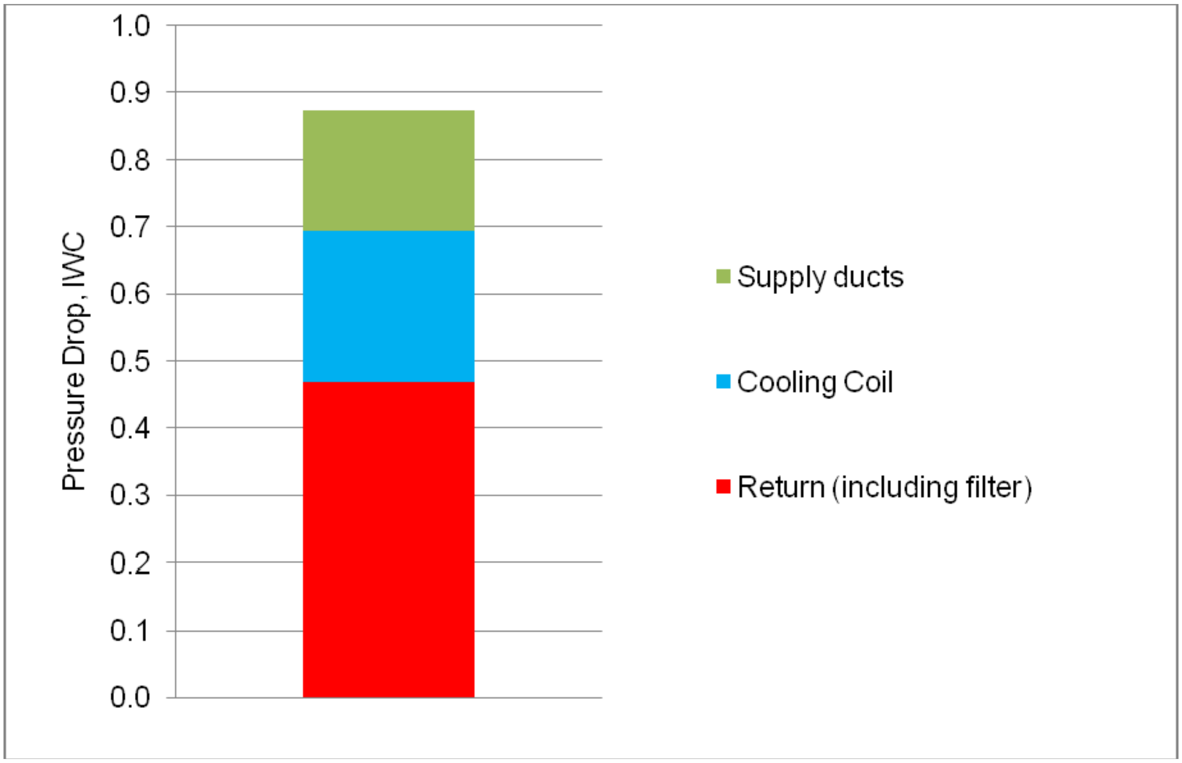


Source: Rick Chitwood

The cooling system airflow is determined by the fan and motor characteristics as well as the air handler/furnace internal design and the system resistance to airflow. The typical circulating air fan motor is a permanent split capacitor motor (PSC). Eighty-seven percent of the PSC units were set on high speed. Four of the units had brushless permanent magnet motors (BPM), which have a different characteristic with respect to watt draw and airflow.

One cause of the low airflows and high watt draws per CFM is the restrictive nature of these duct systems. The average return system pressure drop (including filter), coil pressure drop, and supply pressure drops are shown in Figure 10. Notice that the average return system alone is almost equal to the nameplate certified maximum TOTAL cooling static pressure of 0.50 IWC for all but three of the air conditioners.

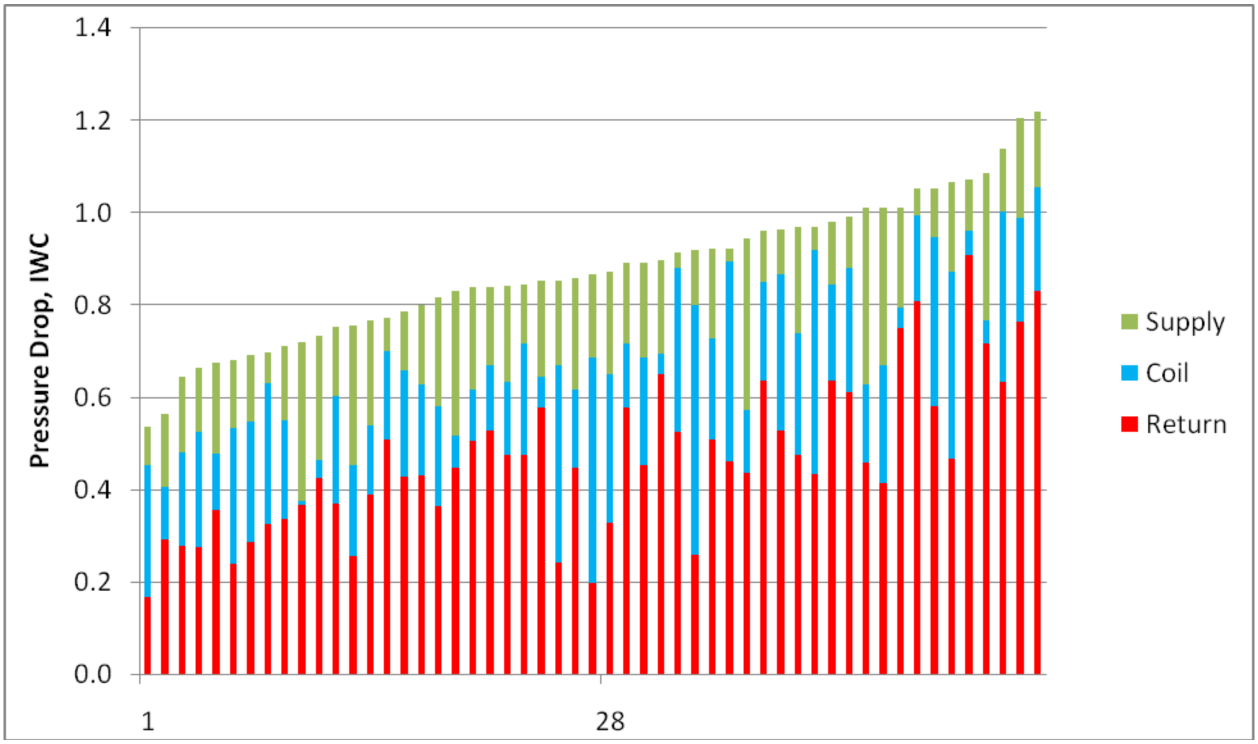
Figure 10: Cooling Airflow Average External Static Pressure



Source: Rick Chitwood

Figure 11 shows the pressure drops in the return, evaporator coil, and supply duct system arranged from the least restrictive system to the most restrictive system. The dominant influence of the return flow resistance is evident in this graph.

Figure 11: Cooling Airflow External Static Pressure by Component



Source: Rick Chitwood

ACCA Manual D, a duct design methodology, suggests that a standard furnace filter will have about a 0.10 IWC pressure drop when it is clean. Unfortunately most of the units do not have standard filters. The mean pressure drop across the filters and for the return systems including the filters are shown in Table 16.

Table 16: Filter and Return System Pressure Drop by AC System Size

Ton	Mean Filter ΔP	Mean Return System ΔP	N
2	0.24	0.45	1
2.5	0.31	0.53	6
3	0.23	0.50	22
3.5	0.39	0.57	9
4	0.21	0.52	8
5	0.40	0.60	8
Total	0.28	0.53	54

Source: Rick Chitwood

The high filter pressure drops are due to a combination of inadequate filter size and the widespread use of 1 inch pleated filters that have high pressure drops even when new. Unit 9 provides an example of the problem, as illustrated by Figure 12 through Figure 14.

Figure 12: Unit 9—HVAC Label

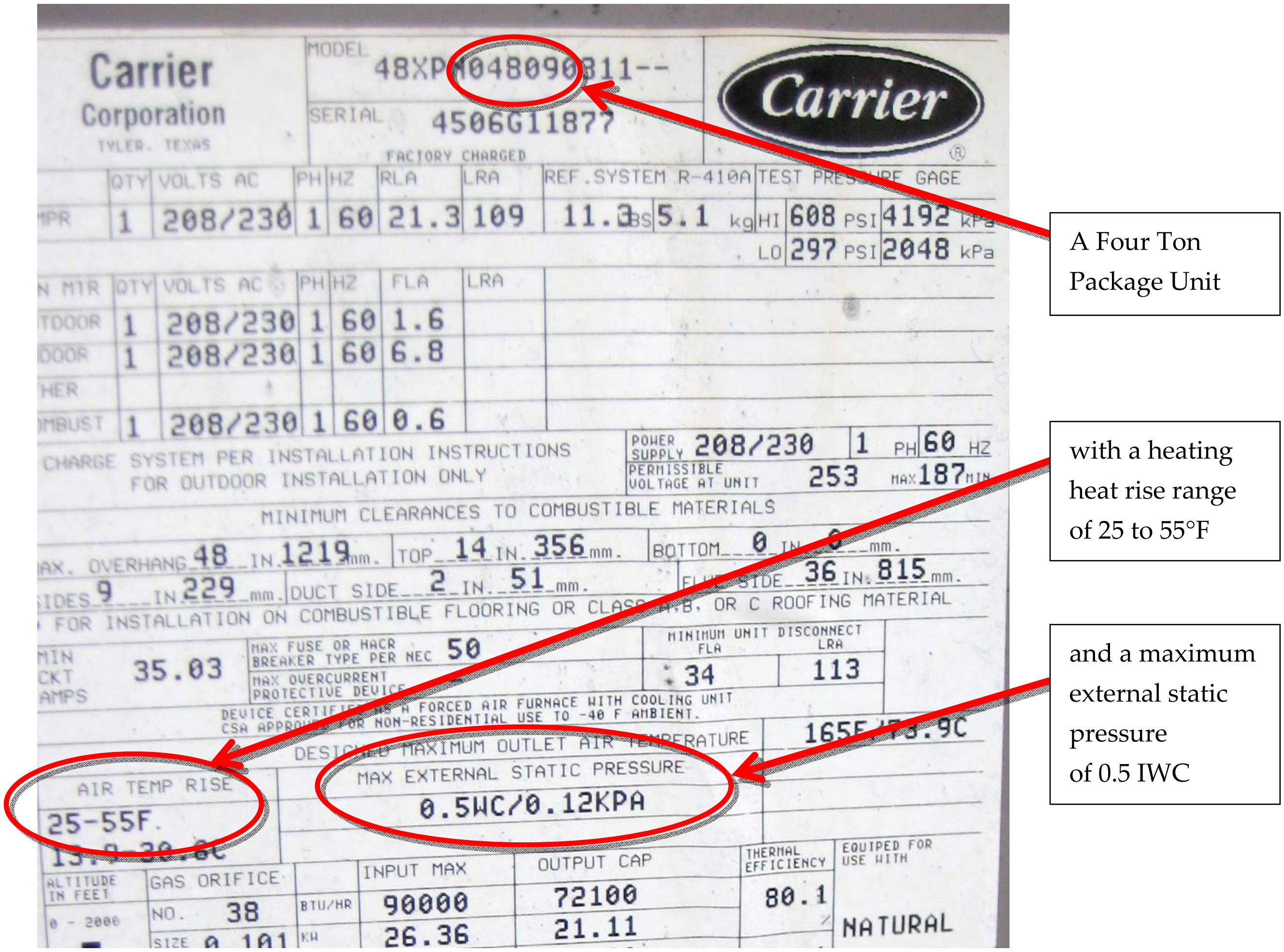


Photo credit: Rick Chitwood

Figure 13: Unit 9—Dirty Filter

Photo Credit: Rick Chitwood

This 14 inch X 30 inch X 1 inch standard filter is so dirty it is sucked into the return duct that is substantially smaller than the return grille. The home owner is sufficiently embarrassed that he offers to go to the store and bring back a new filter. He is instructed specifically to bring back a low efficiency, low cost filter.

Figure 14: Unit 9—Replacement Filter Purchased by Homeowner

Photo Credit: Rick Chitwood

The home owner returns with a 14" X 30" X 1" pleated filter. He is pleased because: "This is 16 times better!"
The result is a record 1.64 IWC pressure drop in the filter alone. This is on a unit that has a stated maximum pressure drop for the whole system of 0.50 IWC.

3.3.1.4 AC Refrigerant Charge and Metering Device Performance

The amount of refrigerant in an air conditioner affects the efficiency of the unit. Both too little refrigerant (Undercharge) and too much refrigerant (Overcharge) result in lower efficiencies. Like most new air conditioners, 90+ percent of these units have thermostatic expansion valves (TXVs). With TXVs the correct level of refrigerant is specified by the manufacturer as determined by the difference in temperature between the refrigerant condensing in the outside coil and the temperature of the liquid refrigerant leaving the outdoor unit (Subcooling). The target subcooling is generally specified by the manufacturer as a single value, such as 10°F. This value varies with make and model, but is often not readily available to technicians who test the units after installation. When the manufacturer's target subcooling is not available, 10°F is often used as a default specification. Title 24 specifies that the installer must adjust the refrigerant charge until the subcooling is within ± 3°F of the target.

Much of the testing in Phase One was in cold weather. In cold weather the pressure in the outdoor coil drops. When there is insufficient pressure to push adequate refrigerant through the TXV to the inside coil, the subcooling is no longer an appropriate indicator of correct refrigerant charge.

The function of the TXV is to meter the refrigerant into the indoor coil such that all of the liquid refrigerant in the inside coil changes to vapor. The TXV performs this function by monitoring the temperature of the refrigerant leaving the indoor coil and adjusting the refrigerant flow so that it always leaves slightly hotter than its boiling temperature. The temperature difference between the boiling temperature and the temperature of the refrigerant vapor leaving the indoor coil is called superheat. To ensure that the TXV is not severely malfunctioning (or other severe faults are caught), Title 24 has specified that the measured superheat[2] not be less than 4°F or greater than 25°F. This test is specified to occur at outdoor temperatures exceeding 55°F.

[2] Measured as the difference between the refrigerant boiling point and the temperature of the refrigerant returning to the outdoor unit.

Table 17 shows 31 percent of the 57 tests above 55°F failed the TXV superheat test, indicating one or more significant faults.

Table 17: Systems Failing the TXV Superheat Criterion

Percentage of Tests	Superheat	Subcooling	Probable Cause
4%	Low	low	TXV problem
2%	Low	ok	TXV problem
5%	low	high	TXV problem with potential overcharge
7%	high	low	Undercharge
9%	high	ok	TXV problem
4%	high	high	Flow restriction or TXV problem
31%	Total		

Source: Data – Rick Chitwood; Analysis – Proctor Engineering Group, Ltd.

High TXV superheat (>25°F) is a clear predictor of low capacity.

Sixty-nine percent of the 57 tests above 55°F passed the TXV superheat test. The units that passed the superheat test were evaluated for refrigerant charge using two different criteria. The first table (Table 18) shows the results using the ± 3°F criterion. This criterion is the Title 24 specification for installers.

Table 19) shows the results if wider criteria are used subcooling between 2°F and 8°F. The wider criteria could be used by HERS raters to ensure the system has the proper refrigerant charge to operate within 5 percent of its rated efficiency.

Table 18: Subcooling Results (± 3°F) for Units Passing the TXV Superheat Criterion

Percentage of Tests	Superheat	Subcooling (Target ± 3°F)	Probable Cause
27%	Ok	low	Undercharge
23%	Ok	high	Overcharge
19%	Ok	ok	Correct Charge
69%	Total		

Source: Data – Rick Chitwood; Analysis – Proctor Engineering Group, Ltd.

Table 19: Subcooling Results (Wider Criteria for HERS Verification) for Units Passing the TXV Superheat Criterion

Percentage of Tests	Superheat	Subcooling (>2°F &<= Target +8°F)	Probable Cause
8%	Ok	low	Undercharge
4%	Ok	high	Overcharge
58%	Ok	ok	Correct Charge
69%	Total		

Source: Data – Rick Chitwood, Analysis – Proctor Engineering Group, Ltd.

The refrigerant charge results using the ± 3°F criterion show similar proportions to those in earlier studies (Downey and Proctor, "What Can 13,000 Air Conditioners Tell Us?" In *Proceedings from the ACEEE 2002 Summer Study on Energy Efficiency in Buildings,* Washington, D.C.: American Council for and Energy-Efficient Economy).

Specific units add additional information on refrigerant charge and TXV superheat criteria:

- Unit 3 passed the TXV superheat criteria, but had no subcooling. The unit was tested at 85°F ambient temperature. Adding 9oz. of refrigerant brought the unit to the target subcooling.
- Unit 37 failed the TXV maximum superheat criterion. The unit was tested at 56°F ambient temperature. The unit had negligible refrigerant and its TXV bulb was hanging loose in the attic (see Figure 15).

Figure 15: Unit 37—TXV Bulb Installation Error

Photo Credit: Rick Chitwood

- Unit 58 was borderline on the TXV superheat criterion (superheat was 24°F, Title 24 maximum is 25°F) and was undercharged in the first test. The unit was tested at 76°F ambient. Adding 43.5 oz. of refrigerant (31 percent of factory charge) brought the subcooling to within specification, but the superheat remained high at 26°F. The TXV had the temperature sensing bulb insulated to the cooling suction line. Since Unit 58 is a heat pump, the location is also the hot gas line in the heating mode. The temperature of the hot gas line in the heating mode exceeded the melting point of the insulation and the insulation dripped off, as shown in Figure 16.

Figure 16: Unit 58—Melted TXV Bulb Insulation

Photo Credit: Rick Chitwood

- Unit 59 passed the TXV superheat criteria, but had no subcooling. The unit was tested at 76°F ambient temperature. Adding 59.5 oz. of refrigerant (46 percent of manufacturer's charge) brought the unit to the target subcooling. The addition of refrigerant increased the unit's sensible capacity by 56 percent.
- Unit 67.1 failed the TXV minimum superheat criterion and was overcharged in the initial test. The unit was tested at 69°F ambient temperature. Removing 41 oz. of refrigerant (29percent of manufacturer's charge) brought the unit to the target subcooling.
- Unit 68 passed the TXV superheat criteria and had 6°F of subcooling. The target subcooling was not listed. The unit was tested at 84°F ambient temperature. Adding 14oz. of refrigerant (11percent of manufacturer's charge) brought the unit to 12°F of subcooling.
- Unit 71.1 failed the TXV maximum superheat criterion and had 6°F of subcooling. The unit was tested at 75°F ambient temperature. Adding 12.5 oz. of refrigerant (11 percent of manufacturer's charge) brought the unit to the target subcooling. The unit continued to fail the TXV maximum superheat criterion.
- Unit 71.2 was bordered the TXV maximum superheat criterion and had 6°F of subcooling with an 11°F target. The unit was tested at 75°F ambient temperature. Adding 15.5 oz. of refrigerant (13 percent of manufacturer's charge) brought the unit to the target subcooling. The unit continued to border the TXV maximum superheat criterion.

- Unit 72 failed the TXV maximum superheat criterion and was undercharged. The unit was tested at 75°F ambient temperature, but the location of the condensing unit caused recirculation (Figure 17) and a condenser air entering temperature of 103°F. Adding 87.5 oz. of refrigerant brought the unit to a subcooling of 10°F and changed the TXV superheat from 33°F to 4°F.

Figure 17: Unit 72—Condenser Recirculation

Photo Credit: Rick Chitwood

- Unit 73 passed the TXV superheat criteria and was mildly undercharged. The unit was tested at 70°F ambient temperature. Adding 11 oz. of refrigerant brought the unit to a subcooling of 10°F and reduced the TXV superheat from 20°F to 16°F.

3.3.1.5 Heating Unit Characteristics

The heating units sported 19 different brand names and 59 different models. Sizes ranged from 18,000 Btuh to 110,000 Btuh with an average input of 67,000 Btuh. The running watt draws (total electrical consumption at the inside air handler unit including gas valve and combustion air blower, if they were present) averaged 572 watts. The breakdown by system type is shown in Table 20.

Table 20: Heating Unit Watt Draw

System Type	Mean Standby W	Mean Burn W	Mean Running W	N
Combined Hydronic			381	11
Furnace Only	5.53	74	463	3
Heat Pump			354	5
Multiple Splits w Furnaces	6.59	89	569	17
Package Rooftop Heat Pump			490	1
Package Rooftop Unit	10.00	75	610	4
Single Split AC w Furnace	6.74	97	642	38

Source: Data – Rick Chitwood

System airflows and the power used to provide those flows are displayed in Table 21.

Table 21: Heating System Airflow

System Type	Mean CFM per kBTU	Mean W per CFM	N
Combined Hydronic	21.5	0.675	9
Furnace Only	12.6	0.597	3
Heat Pump	19.2	0.469	4
Multiple Splits w Furnaces	15.9	0.538	17
Package Rooftop Heat Pump	31.1	0.375	1
Package Rooftop Unit	14.9	0.572	4
Single Split AC w Furnace	14.2	0.607	37

Source: Data – Rick Chitwood

The heating system airflow is determined by the fan and motor characteristics as well as the air handler/furnace internal design and the duct system resistance to airflow. The typical circulating air fan motor is a permanent split capacitor motor (PSC). These motors provide multiple speed taps, but the lower speeds draw almost as much power as the highest speed. Four of the units had brushless permanent magnet motors (BPM) which have different characteristics. The speed setting of the PSC motors in heating are displayed in Table 22.

Table 22: Heating Fan Speed Settings

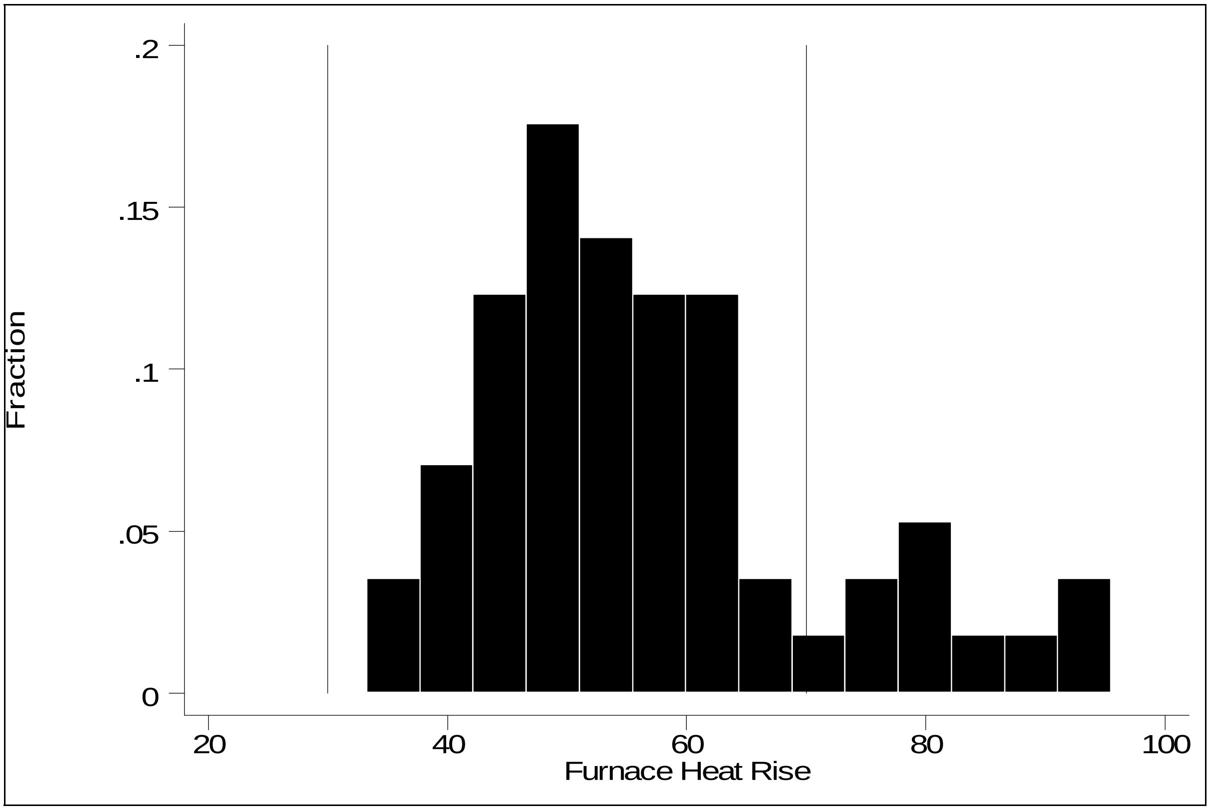
System Type	Low	Med/Low	Med	Med/High	High	Total
Combined Hydronic	2	0	5	1	3	11
Furnace Only	2	1	0	0	0	3
Heat Pump	0	0	3	0	4	7
Multiple Splits w Furnace	5	3	5	4	0	17
Package Rooftop Heat Pump	0	0	1	0	0	1
Single Split AC w Furnace	4	9	18	9	1	41
Total	13	13	32	14	8	80

Source: Rick Chitwood

Based on a steady state efficiency of .85, the heat rise ranges between 34°F and 96°F, as shown in Figure 18.[3]Approximately 20 percent of the units exceed the manufacturer's certified heat rise; substantially more of the units exceed their optimum heat rise. Higher heat rises reduce the efficiency of the units below their AFUE rating.

[3] Zoned systems with closed dampers have higher heat rises and have been excluded from this graph.

Figure 18: Calculated Furnace Heat Rise

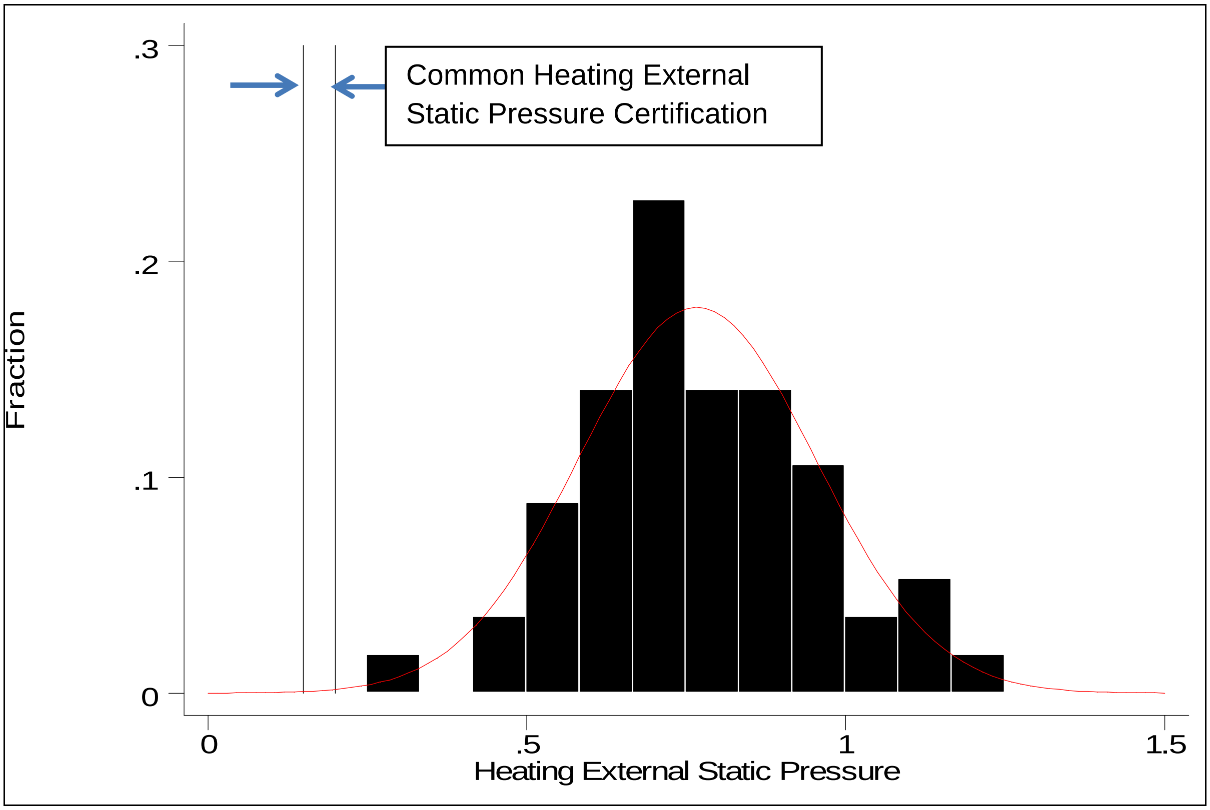


Source: Data – Rick Chitwood; Calculations – Proctor Engineering Group, Ltd.

With proper airflow the data would range from 30°F to 70°F and be a normal distribution centered at about 50°F. This data is skewed to the right (high heat rise due to low airflow). Low airflow not only lowers the efficiency of the furnace, it can also cause the unit to cycle the gas off and on by the limit switch, potentially increasing heat exchanger fatigue and corrosion.

The cause of the low airflow is a combination of unrealistically low heating static pressure certifications (generally between 0.15 Inches of Water Column and 0.25 Inches of Water Column) and the true external static pressures displayed in Figure 19.

Figure 19: Heating Mode External Static Pressure vs. Common Certification Range



Source: Data – Rick Chitwood

3.3.1.6 Air Distribution System Characteristics

The typical duct system in these residences made major use of helical coil plastic flex duct. The locations of the ducts are shown in Table 23.

Table 23: Duct Locations by Building Type

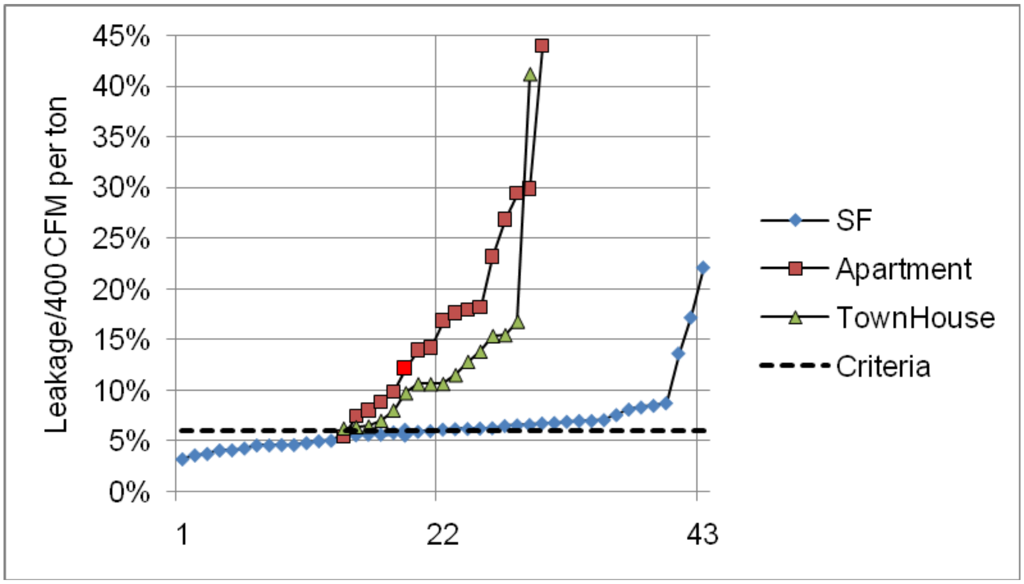
Duct Location	Apartment	Single-Family	Town House	Total
100% Inside	2	0	1	3
Attic	2	29	4	35
Attic, Floors	0	1	0	1
Attic, Walls	0	11	7	18
Attic, Walls, Floors	1	9	6	16
Cathedral Ceiling	0	0	1	1
Floors	1	0	0	1
Soffit	11	0	0	11
Walls	0	0	1	1
Walls, Floors	0	1	0	1
Walls, floors	1	0	0	1
Total	18	51	20	89

Source: Data – Rick Chitwood

Seventy-eight percent of the ducted systems had some or all of the ducts in the attic. This location provides the most severe case for conduction losses and return leakage problems. Most of the apartments had their ducts within the conditioned space or within a soffit.

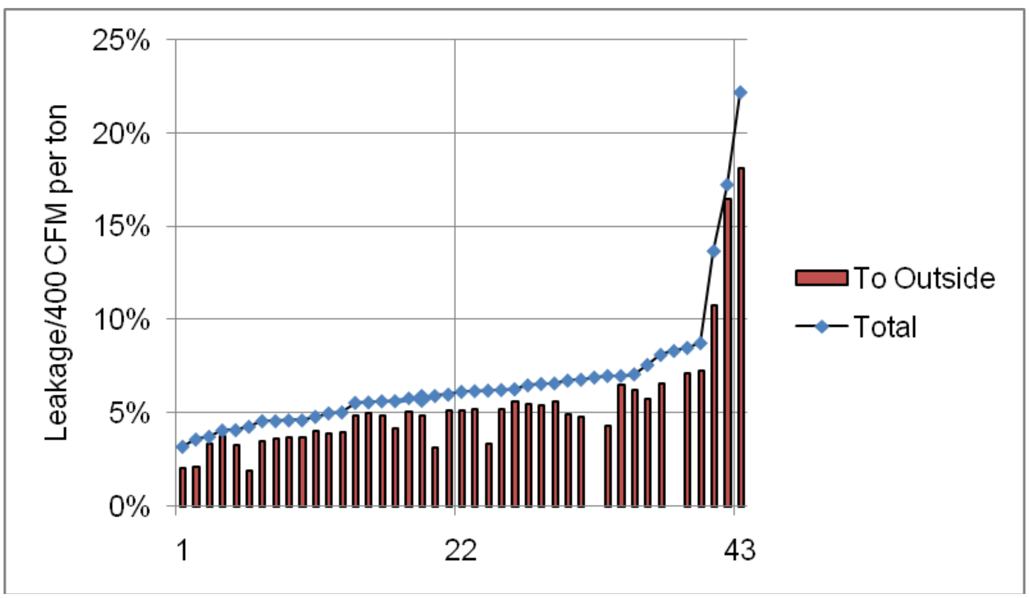
The total duct leakage and the duct leakage to outside are shown in Figure 20, Figure 21 and Figure 22.

Figure 20: Duct Leakage by Building Type



Source: Data – Rick Chitwood

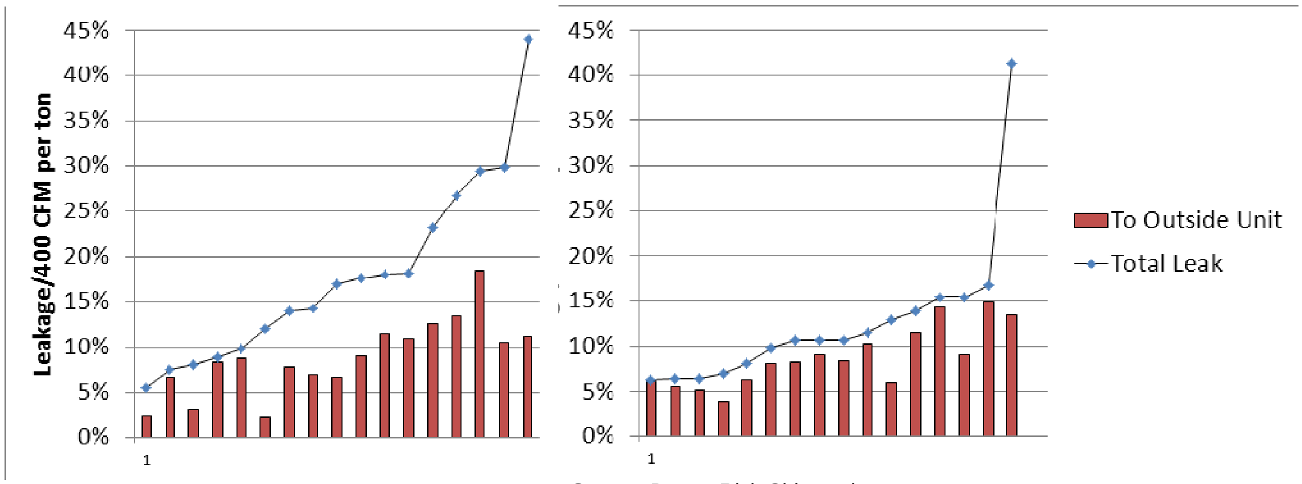
Figure 21: Duct Leakage for Single-Family Buildings



Source: Data – Rick Chitwood

Figure 22: Duct Leakage for Apartments and Town Houses

Apartments Town Houses



Source: Data – Rick Chitwood

In summer, the effects of duct leakage include capacity loss from supply leaks, infusion of superheated attic air from return leaks, and house infiltration due to the imbalance between supply and return leakage. The infiltration effect occurs regardless of whether the dominant leakage is in the supply system or the return system. The leakage imbalance was estimated by the "Half Nelson" test combined with the operating pressures of the system. The calculation occurs in the following manner:

The Half Nelson test blocks off all intentional openings in the duct system and uses the air handler to move an equal amount of air in through the return leaks and out through the supply leaks.

$$Flow_r = Flow_s$$

The flow through a leak is related to the pressure differential across the leak in this manner:

$$Flow_x = K_x \times \Delta P_x^a$$

Where:

K_x is a constant relating to the area of the leak

a is often taken as .5 but is also taken as .65 in house leakage. We used .5.

ΔP_x is the pressure difference across the leak

Since the flows are equal:

$$K_r \times \Delta P_r^{0.6} = K_s \times \Delta P_s^{0.6}$$

and

$$K_r = K_s \frac{\Delta P_s^{0.6}}{\Delta P_r^{0.6}}$$

We are interested in the proportion of the leakage area that occurs in each portion of the duct system. We assume that the characteristics of the leaks (other than area) are the same in both portions of the duct system.

Therefore we are interested in relative leakage areas which are:

$\frac{K_r}{K_r + K_s}$ and $\frac{K_s}{K_r + K_s}$

Since we are only looking for the relationship between the two leakage areas, we take K_s as unity ($K_s = 1$).

Therefore the relative return leakage area fraction is:

$$A_r = \frac{K_r}{K_r + 1} = \frac{K_s \frac{\Delta P_s^{0.6}}{\Delta P_r^{0.6}}}{1 + K_s \frac{\Delta P_s^{0.6}}{\Delta P_r^{0.6}}} = \frac{1}{1 + \frac{\Delta P_r^{0.6}}{\Delta P_s^{0.6}}}$$

and the relative supply leakage area fraction is:

$$A_s = 1 - A_r$$

The return leakage and supply leakage relative flows are:

$Flow_{opR} = A_r \Delta P_{opR}^{0.6}$ and $Flow_{opS} = A_r \Delta P_{opS}^{0.6}$

Where:

ΔP_{opR} designates the operating pressure at the return plenum and

ΔP_{opS} designates the operating pressure at the supply plenum after the evaporator coil.

The return and supply leakage fractions are:

$LF_R = \frac{Flow_{opR}}{Flow_{opR} + Flow_{opS}}$ and $LF_S = \frac{Flow_{opS}}{Flow_{opS} + Flow_{opR}}$

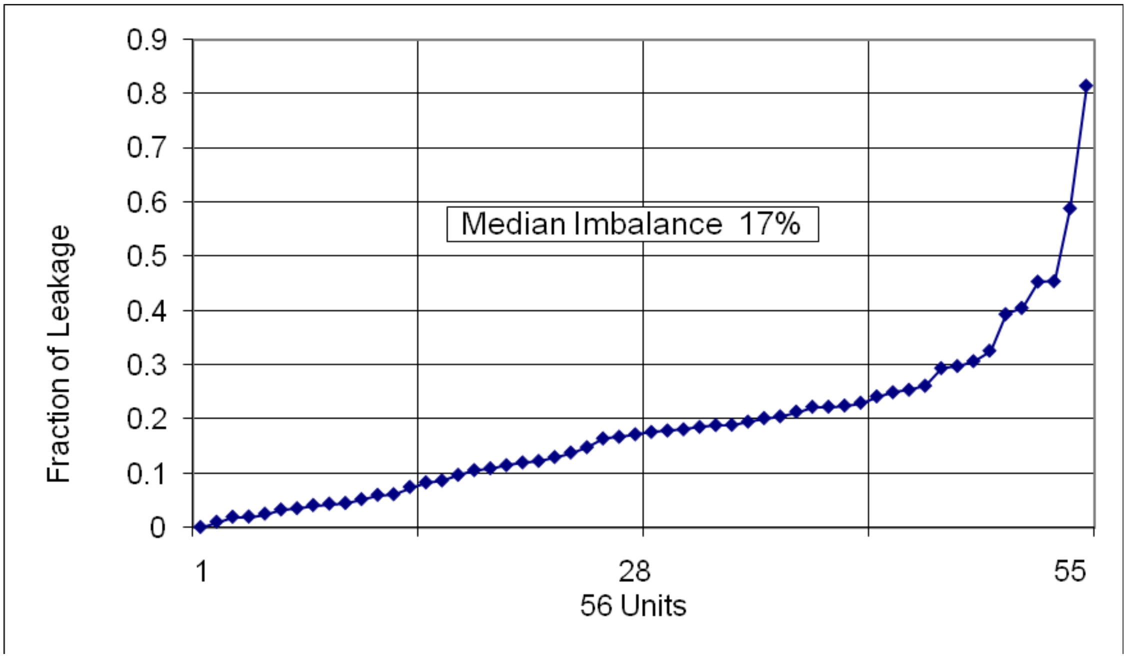
Balanced leakage flow would be 50 percent of the total leakage in both the return and supply sections.

The leakage imbalance therefore is:

$$LeakageImbalance = absolute\ value\ (0.5 - LF_R)$$

The leakage imbalance is displayed in Figure 23.

Figure 23: Duct Leakage Imbalance



Source: Data – Rick Chitwood

In 30 of the single-family systems, the air conditioners were tested for sensible capacity at the air conditioner and delivered at the registers. The test conditions were in cool weather wherein the conduction heat gains were low. The average register delivery was 90 percent of the sensible capacity measured at the unit.

It is notable that few of the units had insulation in the blower compartment. Given the high temperatures in the attic in the summer, insulating that section of ductwork should be cost effective and is probably required by the letter of the law in Title 24.[4] Figure 24 shows a typical uninsulated blower compartment and a not so typical insulated blower compartment.

[4] According to AFUE test procedure (ANSI/ASHRAE Standard 103-2007 Section 8.6.1.1) “the circulating air blower compartment is considered as a part of the duct system” not as part of the furnace. One interpretation is that these installations are out of compliance with Title 24 since that part of the duct system is not insulated.

Figure 24: Insulated and Uninsulated Blower Compartments

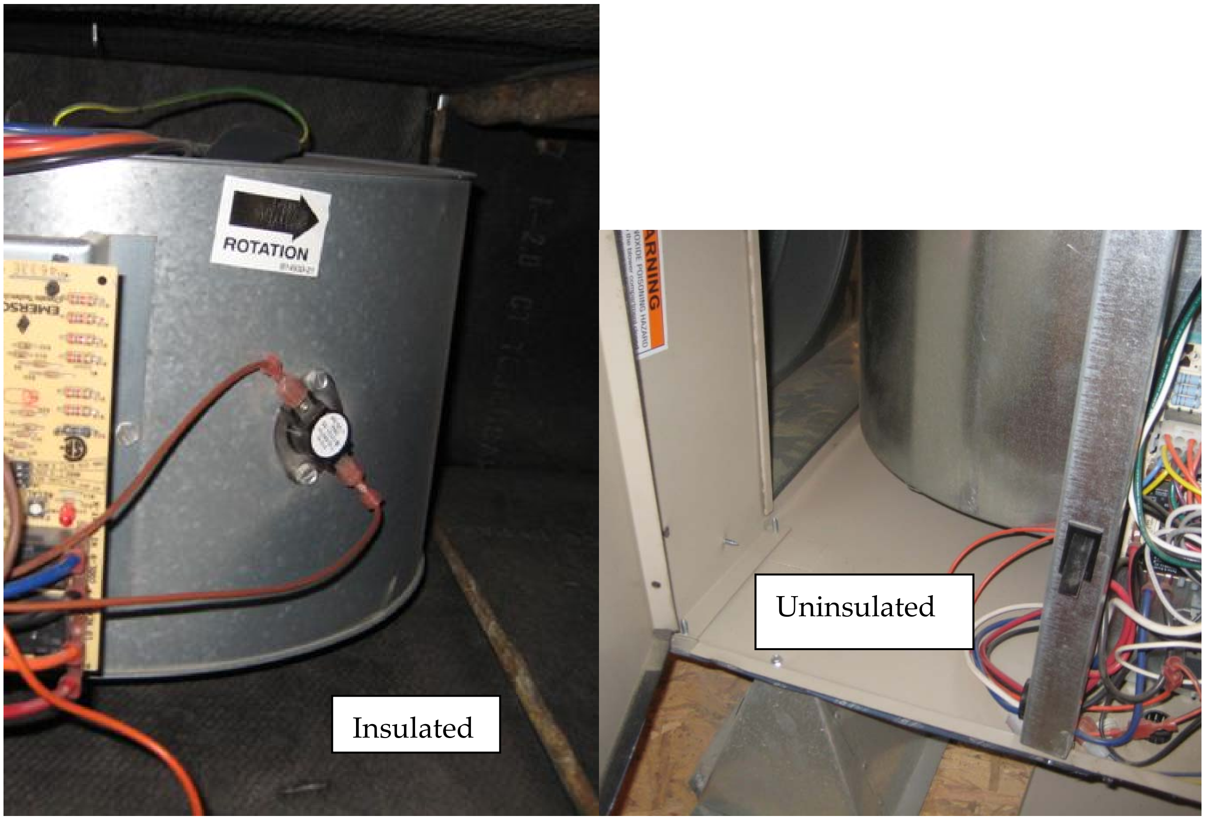


Photo Credit: Rick Chitwood

3.3.2 Phase Two

In Phase Two the project returned to 10 HVAC systems to:

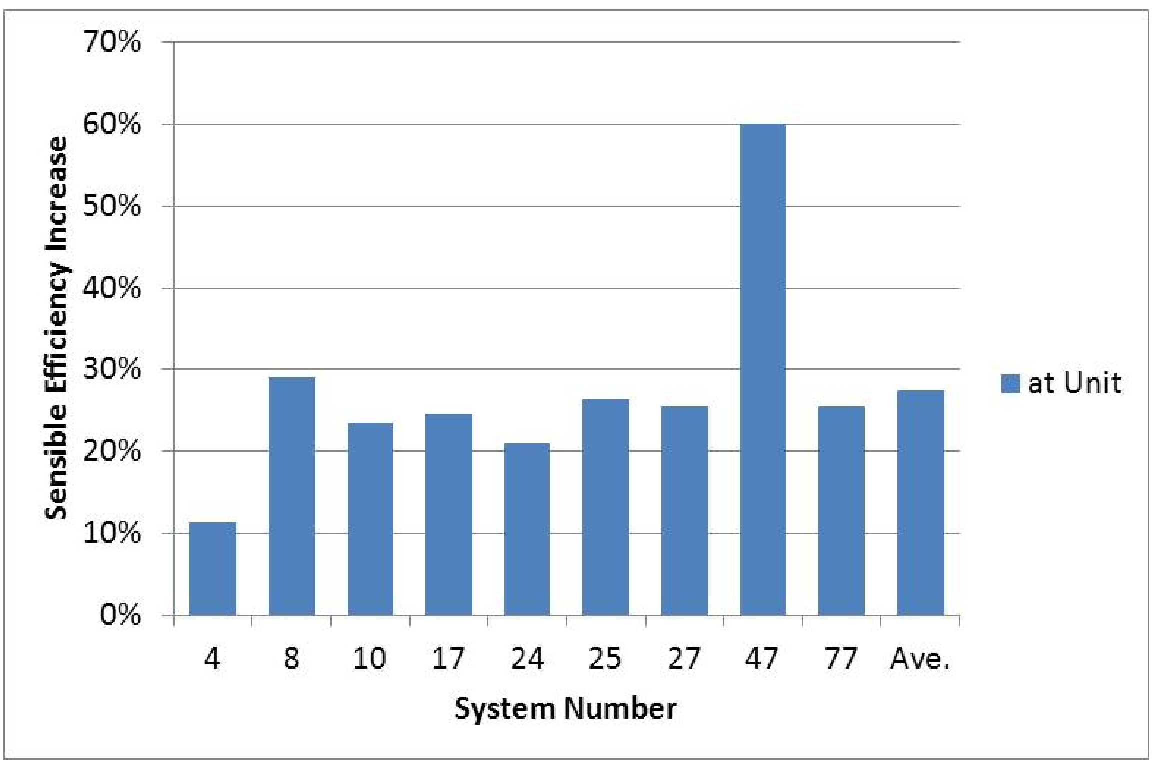
- determine the accuracy of the original measurements,
- refine the measurement methods,
- achieve moderate repairs for energy efficiency, and
- measure the effect of the repairs.

These units were sampled as covering the range of units found in the 80 unit survey, with particular focus on single-family units. One extreme case was investigated (an AC unit that never worked since the house was occupied). Where appropriate this unit (#74) is excluded from the averages.

Most systems were treated with two repairs. Two systems were treated with three repairs and one system had one repair. The repairs and the results of the repairs are listed in this section.

All three levels of repairs showed statistical significance at 0.01 level in paired tests. The repairs improved the measured Normalized Energy Efficiency Ratio – Sensible ($NEER_S$) between 3 percent and 60 percent. The $NEER_S$ is the sensible capacity corrected to standard conditions (95 °F outside, 80 °F 50 percent relative humidity inside). The efficiency improvement was 24 percent± 13 percent. Figure 25 displays the improvement from pre to post for each unit.

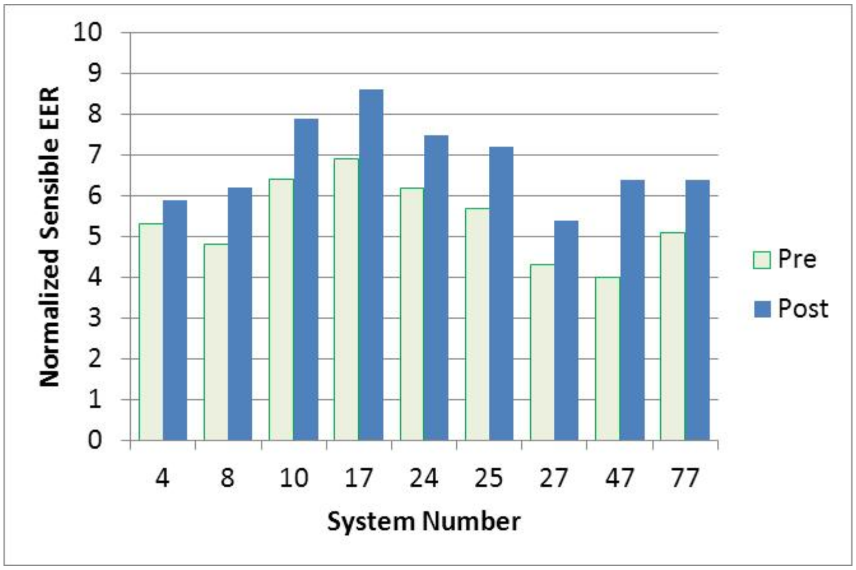
Figure 25: Normalized Sensible EER Improvement



Source: Data – Rick Chitwood

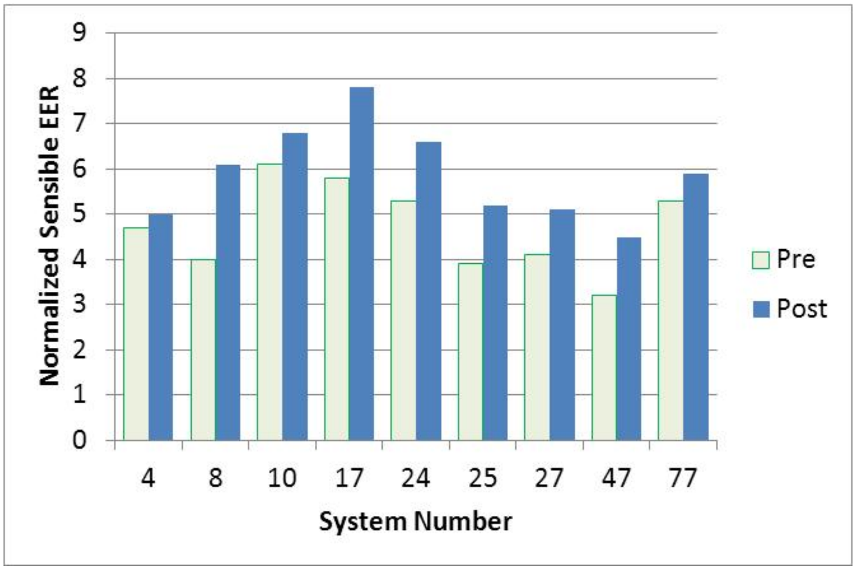
The Normalized Sensible EERs reported for each unit in this section are based on temperature and flow measurements at the air conditioners. This method is considered to have less uncertainty than the measurements at the registers. There is no statistically significant difference between the average efficiency increase using the two methods. The pre and post-repair efficiencies for each method are shown in Figure 26 and Figure 27.

Figure 26: Normalized Sensible EER from Measurements at the Unit



Source: Data – Rick Chitwood

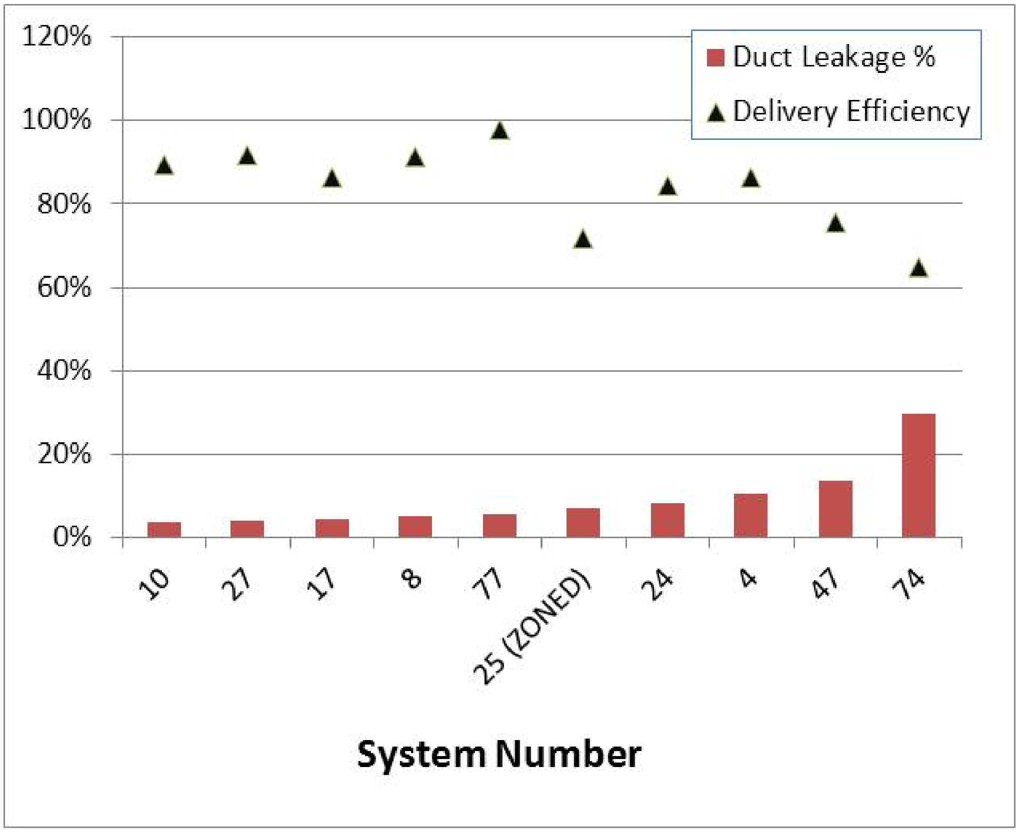
Figure 27: Normalized Sensible EER from Measurements at the Registers



Source: Data – Rick Chitwood

The relationship between the Normalized Sensible EER at the registers and at the unit is an indication of the duct efficiency. This is a steady state indicator that will vary with conditions and does not include the infiltration effect of unbalanced duct leakage. Figure 28 displays the delivery efficiency for each unit averaged over all the tests.

Figure 28: Delivery Efficiency and Duct Leakage



Source: Data – Rick Chitwood

3.3.2.1 System 4

Change #1:

- Removed all refrigerant and replaced with clean refrigerant — the efficiency dropped from 5.7 to 5.3.

Change #2:

- The baseline EER for this change is taken as 5.3 to determine the efficacy of the return system changes.
- Revised original return can from 10 in. x 20 in. to 20 in. x 30 in. (Figure 29)
- Added a 10 in. dia. flex duct from revised return can to return plenum (Figure 29).
- Shortened 16 in. dia. flex return duct by 15 in.
- Used a sheet metal elbow to tap 16 in. return into blower compartment (Figure 30).

- Moved shortened 16 in. duct to connect to open blower wheel side of the blower compartment (Figure 30).

Figure 29: System 4—Original and Revised Return Can With Added 10 in. Return Duct

Photo Credit: Rick Chitwood

Figure 30: System 4—New 10 in. to Plenum and 16 in. Into Blower Wheel Inlet Side With Metal Elbow

Photo Credit: Rick Chitwood

These changes increased the airflow by 32 percent from 280 CFM per ton to 371 CFM per ton. The return system static pressure was reduced from 0.48 IWC to 0.18 IWC. The Normalized Sensible EER increased by 11.3 percent.

3.3.2.2 System 8

Change #1:

- Removed and replaced refrigerant – unit's TXV is not metering properly. Replacing charge resulted in no efficiency improvement.

Change #2:

- Replaced clogged 20 in. x 25 in. filter.
- Installed a new 14 in. x 24 in. return filter grille by the original filter grille.
- Ran 12 in. dia. flex duct to blower compartment fan opening inlet.

Figure 31: System 8—Clogged Filter and Original Return Duct Into "Bottom" of Furnace

Photo Credit: Rick Chitwood

Figure 32: System 8—New Filter Grille Near Existing Grille and New 12 in. dia. Flex Duct Into Furnace Cabinet

Photo Credit: Rick Chitwood

These changes increased the airflow by 44 percent from 248 CFM per ton to 357 CFM per ton. The return system static pressure was reduced from 0.90 IWC to 0.26 IWC. The Normalized Sensible EER increased by 26.5 percent.

3.3.2.3 System 10

Change #1:

- Unit appeared properly charged based on superheat and subcooling. TXV had a “hunting” problem. Removed and replaced refrigerant.

This change resulted in an improvement in the Normalized Sensible EER of 18.8percent and eliminated the TXV hunting. The unit’s original refrigerant charge was probably contaminated with non-condensables.

Change #2 (see Figure 33):

- Installed “High Flow” filter in existing 20 in. x 30 in. filter grille.
- Installed a new 20 in. x 20 in. return filter grille with 14 in. flex duct beside the existing filter grille.
- Ran 14 in. dia. flex duct to metal elbow to blower compartment.

Figure 33: System 10—New Filter Grille Near Existing Grille and New 14 in. dia. Flex Duct Into Furnace Cabinet

Photo Credit: Rick Chitwood

These changes increased the airflow by 13.7 percent from 307 CFM per ton to 349 CFM per ton. The return system static pressure was reduced from 0.46 IWC to 0.17 IWC. The Normalized Sensible EER increased by 3.9 percent.

3.3.2.4 System 17

Change #1:

- Unit undercharged based on subcooling. Removed and replaced refrigerant.

This change resulted in a 3 percent improvement in the Normalized Sensible EER.

Change #2 (see Figure 34 and Figure 35):

- Unit had one 16 in. dia. return running 50 ft to a 20 in. x 30 in. filter grille in a distant hallway.
- The return duct took a sharp turn leaving the return grille so the actual duct was squashed to 5 in.
- The return was replaced with a new 20 in. x 30 in. filter grille in the hallway below the air conditioner feeding one 16 in. diameter duct and one 12 in. diameter duct into the blower compartment.

Figure 34: System 17—Existing Return With 5 in. Opening

Photo Credit: Rick Chitwood

Figure 35: System 17—New Return Location and Double Ducts Into Blower Compartment

Photo Credit: Rick Chitwood

These changes increased the airflow by 39.6 percent from 278 CFM per ton to 388 CFM per ton. The return system static pressure was reduced from 0.63 IWC to 0.18 IWC. The Normalized Sensible EER increased by 18 percent.

Change #3:

- The Permanent Split Capacitor (PSC) blower motor was replaced with a Brushless Permanent Magnet (BPM/ECM) blower motor adjusted to the same airflow.

This change reduced the fan watt draw by 102 Watts and increased the Normalized Sensible EER by 4 percent.

3.3.2.5 System 24

As built, the 4 ton unit had a 20 in. x 24 in. return grille with a 1 in. filter feeding an 18 in. duct routed to the "bottom" of the furnace cabinet. The unit also had a 14 in. x 14 in. wall return grille with a 1 in. filter feeding a 10 in. duct routed to the motor side of the blower compartment.

Change #1:

- A 20 in. x 30 in. ceiling filter grille and large duct board return box was installed with a 2 in. "high flow" filter. The new return box fed a 20 in. flex duct routed to the "bottom" of the furnace cabinet.
- The 20 in. x 24 in. filter grille was modified to accept a 2 in. deep filter, a 2 in. "high flow" filter was installed and the existing 18 in. duct was shortened and fed into the new ductboard return box.

This change increased the airflow by 37 percent from 244 CFM per ton to 335 CFM per ton and lowered return static pressure from 0.73IWC to 0.38 IWC. The flow increase resulted in a 22.6 percent improvement in the Normalized Sensible EER.

Change #2:

- Corrected a minor undercharge.

This change resulted in a no noticeable improvement in the Normalized Sensible EER.

3.3.2.6 System 25

This is a 3 ton zoned system with a supply plenum to return plenum bypass and a single 20 in. x 25 in. filter grille.

Change #1:

- Eliminated the bypass.

This change increased the airflow delivered to the house by 50 percent from 223 CFM per ton to 334 CFM per ton and resulted in a 17 percent improvement in the Normalized Sensible EER.

Change #2:

- Replaced refrigerant.

This change made no noticeable change in the Normalized Sensible EER.

Change #3:

- Added a 20 in. x 25 in. filter grille next to existing 20 in. x 25 in. filter grille and ducted via a 14 in. R-8 duct into furnace cabinet with a metal elbow at the motor side of the blower cabinet (see Figure 36).

Figure 36: System 25—Metal Elbow Into Cabinet and New Filter Grille

Photo Credit: Rick Chitwood

This change lowered the return static pressure from 0.39 IWC to 0.10 IWC and increased flow 11 percent. The Normalized Sensible EER increased by 8 percent.

3.3.2.7 System 27

This unit started with the second lowest normalized sensible EER (4.3) which is 55 percent of the manufacturer's specified Sensible EER at 350 CFM per ton. It is a 3 ton package system with a cabinet return opening of 16 in. x 10 in. fed from an "ovalized" 14 in. dia. duct from a single filter grille (Figure 37).

The inlet to the evaporator coil is restricted by baffles around the compressor compartment (Figure 37).

Figure 37: System 27—"Ovalized" Return Duct and Airflow Restriction to Evaporator Coil

Photo Credit: Rick Chitwood

The system has no cooling inlet for the compressor and has melted the insulation on the TXV bulb (Figure 38 and Figure 39).

Figure 38: System 27—Compressor Compartment With Melted TXV Insulation

Photo Credit: Rick Chitwood

Figure 39: System 27—Compressor Heat Outlet With No Cooling Inlet

Photo Credit: Rick Chitwood

The insulation of this package rooftop unit is both thin and loose (Figure 40).

The ECM fan motor setting was "Nominal" 350 CFM per ton (Figure 40) but the actual airflow was 289 CFM per ton.

Figure 40: System 27—Rooftop Unit Cabinet Insulation and Fan Motor Setting Table

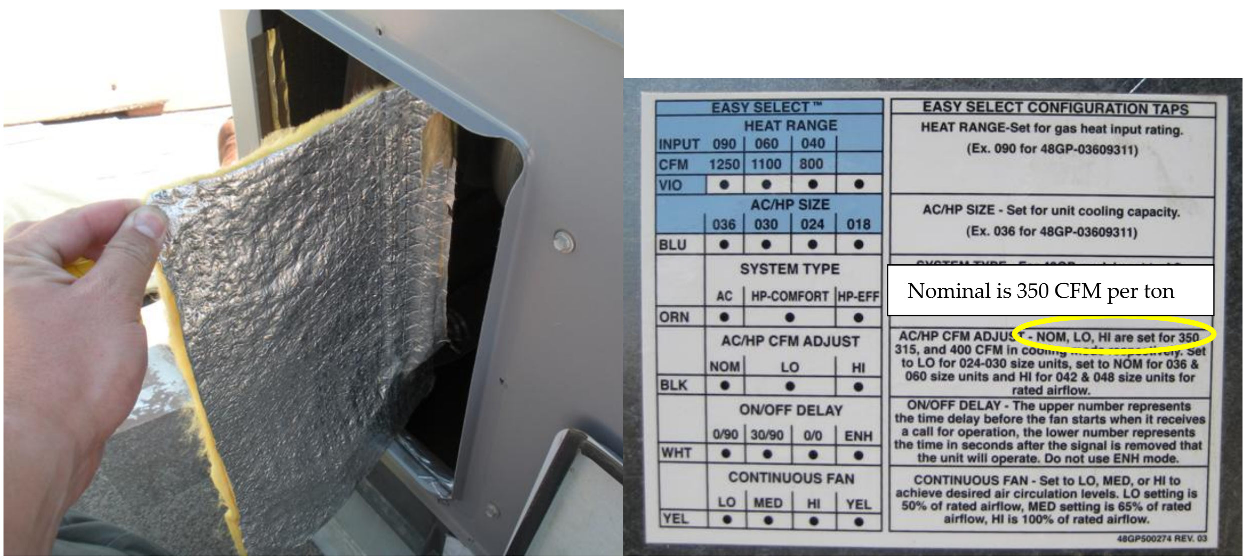


Photo Credit: Rick Chitwood

Change #1:

- Replaced 14in. x25 in. filter grille with 20 in. x30in. filter grille with a "high flow" filter (Figure 41).
- Replaced 14 in. dia. duct with 16 in. dia. duct.
- Opened closed supply air grilles.

Figure 41: System 27—Original Return Grille With Pleated Filter and Enlarged Return Grille With "High Flow" Filter

Photo Credit: Rick Chitwood

This change lowered the return static pressure from 0.69 IWC to 0.22 IWC and increased the airflow by 19 percent from 289 CFM per ton to 343 CFM per ton. The fan watts dropped from 620 to 350watts.

The result was a 26 percent improvement in the Normalized Sensible EER.

Change #2:

- Fan motor was set to "High."

The result was an increase from 343 CFM per ton to 383 CFM per ton and an increase in fan watt draw from 350 to 550 watts.

3.3.2.8 System 47

This unit started with the lowest normalized sensible EER (4.0) which is 49.6 percent of the manufacturer's specified Sensible EER at 350 CFM per ton. It is a 3 ton split system. The unit showed high superheat and low subcooling indicating undercharge.

This unit is a zoned system without a bypass. The tests were all completed with all zone dampers open.

Unit had one 14 in. X 24 in. X 1 in. ceiling return grille feeding a 14 in. diameter duct to a return plenum feeding an electronic air cleaner then into the "bottom" of the furnace (Figure 42).

Figure 42: System 47—Original Return Grille Near Skylight and Side Feed "Bubble Wrapped" Return Plenum and Electronic Air Cleaner

Photo Credit: Rick Chitwood

Change #1:

- Removed 4 lb 15 oz. of refrigerant.
- It took over two hours with two vacuum pumps to pull vacuum (indicating non-condensables in unit).
- Added 6 lb 6 oz. of refrigerant.

This change resulted in a 35 percent improvement in Sensible EER, which was the largest efficiency improvement in the sample.

Change #2:

- Removed electronic air cleaner and "bubble wrapped" return plenum.
- Added 20 in. x 25 in. x 1 in. (16 in. duct) return from skylight sidewall to "bottom" of furnace (Figure 43).
- Changed existing "high efficiency" filter to a "high flow" filter.
- Moved existing 14 in. return duct to non-motor side of fan cabinet (Figure 43).

Figure 43: System 47—New Return Grille in Skylight Channel and New Feed Into Side of Furnace Cabinet

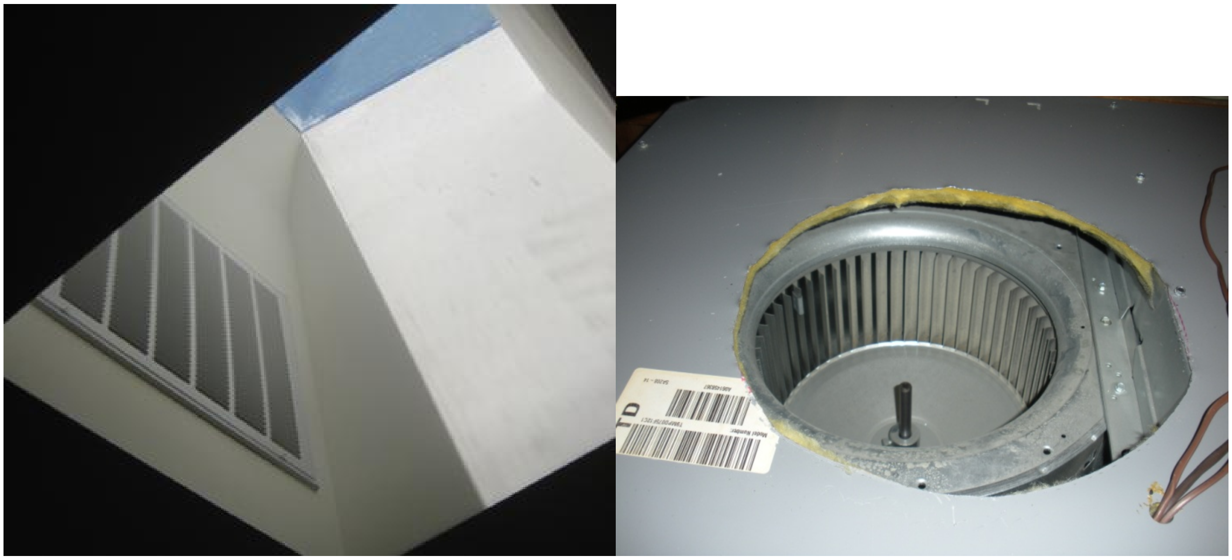


Photo Credit: Rick Chitwood

This change resulted in lowered the return static pressure from 0.69 IWC to 0.16 IWC and increased flow from 253 CFM per ton to 386 CFM per ton. The Sensible Efficiency was improved by 18.5 percent.

3.3.2.9 System 74

This unit was not operational. There were multiple refrigerant leaks and the low voltage controls were not connected. This system is a 3 ton unit on a platform return. The platform return was poorly constructed without hard ducting and with undersized openings into the platform and furnace. Supply ducts were inaccessible between floors or in the "attic-less" ceiling. See Figure 44 and Figure 45.

Figure 44: System 74—Poor Quality Installation: Control Wires Not Connected and Bad Brazing

Photo Credit: Rick Chitwood

Figure 45: System 74—Poor Quality Installation: Restricted Return Grille Opening and Restricted Platform Opening

Photo Credit: Rick Chitwood

Change #1:

- Repaired refrigerant system leaks, including one accessible only through a recessed ceiling lamp and one in the evaporator coil caused by the manufacturer-installed clip that retained the TXV sensor bulb (Figure 46).
- Connected 24 volt control system.
- Evacuated and charged to near proper charge.

Figure 46: System 74—Refrigerant Leaks

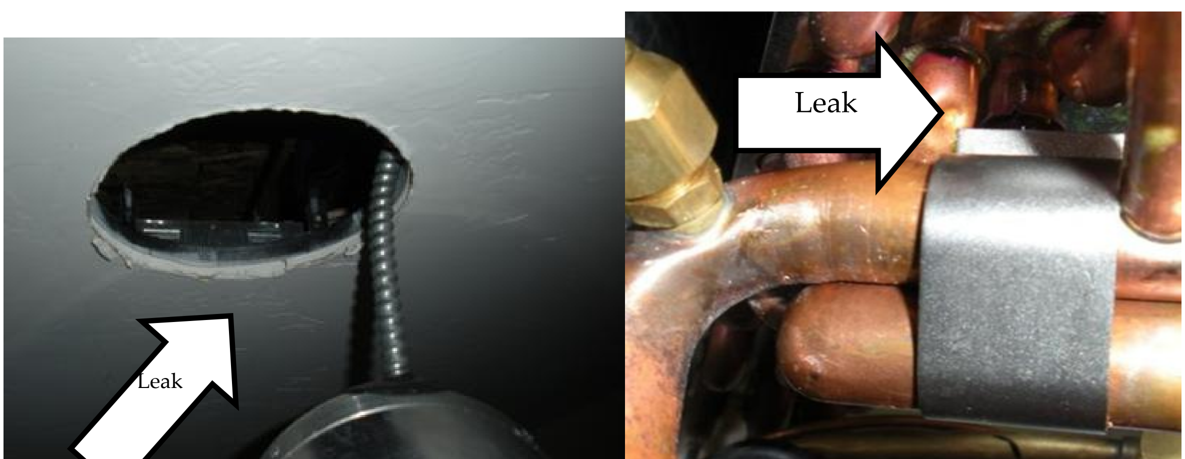


Photo Credit: Rick Chitwood

One leak is accessible through a recessed lamp, the other from the TXV bulb clip.

These repairs produced the highest Normalized Sensible Efficiency of the sample (7.0). Distribution efficiency, however, was the worst of the sample, with the highest duct leakage and delivering only 64percent of the sensible capacity from the unit to the registers.

Change #2:

- Opened return opening into the furnace (Figure 47).
- Removed 11.75 oz. of refrigerant.

Figure 47: System 74—Furnace Return Opening Before and After Enlargement

Photo Credit: Rick Chitwood

This change resulted in a reduction in return static pressure from 0.22 IWC to 0.13 IWC. The airflow marginally increased from 298 CFM per ton to 301 CFM per ton. The Normalized Sensible EER increased 4 percent to 7.2.

Change #3:

- Sealed 11 supply boots and boots to drywall (Figure 48).

This change resulted in a 37 CFM25 reduction in duct leakage.

Figure 48: System 74—One of Many Boot and Boot-to-Drywall Leaks

Photo Credit: Rick Chitwood

3.3.2.10 System 77

This unit is a 3 ton heat pump with a draw through slab coil and a side feed return plenum (Figure 49). The 16 in. return duct is constricted in multiple locations (Figure 50).

The return layout should have been straight into the "bottom" of the air handler.

Figure 49: System 77—Slab Coil and Separate Side Feed Return Plenum

Photo Credit: Rick Chitwood

Figure 50: System 77—One of Multiple Constrictions in the Return Duct

Photo Credit: Rick Chitwood

The condenser coil is caked with dryer lint from a poorly placed dryer vent (Figure 51).

Figure 51: System 77—Dryer Vent and Lint on the Condenser Coil

Photo Credit: Rick Chitwood

Change #1:

- Switched PSC fan motor from medium to high speed (Figure 52).
- Removed clogged filter and replaced with "high flow" filter.
- Opened all supply registers.
- Opened the louvers on the return grill. It was making a bad noise.
- Sealed 16 CFM25 of duct leaks around air handler.

Figure 52: System 77—Speed Tap on PSC Motor Moved From Medium to High

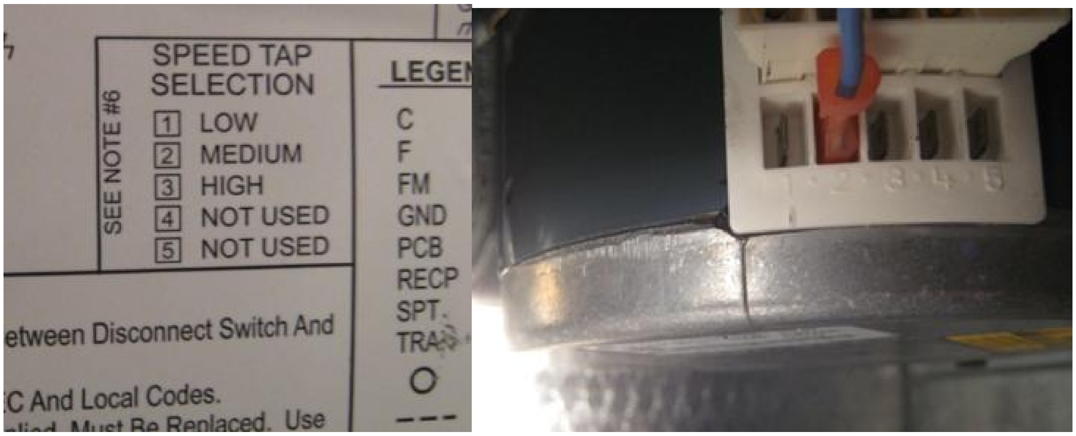


Photo Credit: Rick Chitwood

This change increased the 242 CFM per ton to 346 CFM per ton. In spite of the increased flow the return static pressure prior to the draw through coil was reduced from 0.60 IWC to 0.44 IWC. The supply static pressure increased from 0.16 to 0.26 IWC. The fan motor wattage increased from 330 to 440 watts.

The Normalized Sensible EER was increased by 21 percent from 5.1 to 6.1.

Change #2:

- Unit had low subcooling. Refrigerant was removed and properly charged with clean refrigerant.

This change improved the Normalized Sensible EER by 4 percent.

3.4 Building Shell

3.4.1 Fireplaces

Twenty-seven fireplaces were tested for leakage. The leakage to outside ranged from 25 CFM50 to 517 CFM50. The types of fireplaces and their leakage statistics are shown in Table 24.

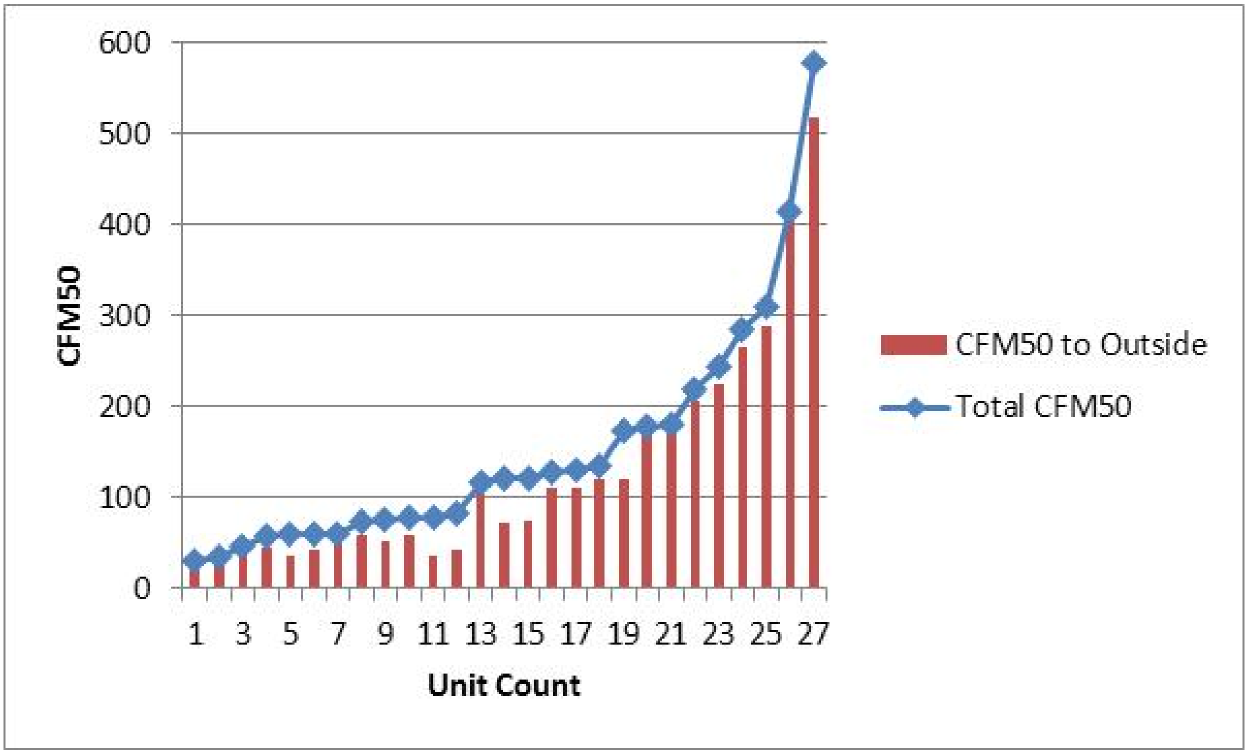
Table 24: Fireplace Statistics by Fuel and Building Type

Building and Fuel	Mean CFM 50 to Outside	Std. Dev.	Range	N
Condo or Apartment – Gas	66	36.7	35 – 119	4
Single-Family – Gas	142	126	25 – 517	18
Single-Family – Wood with Gas	136	157	42—412	5

Source: Data – Rick Chitwood.

As expected most of the fireplace leakage was to outside the building, as shown in Figure 53.

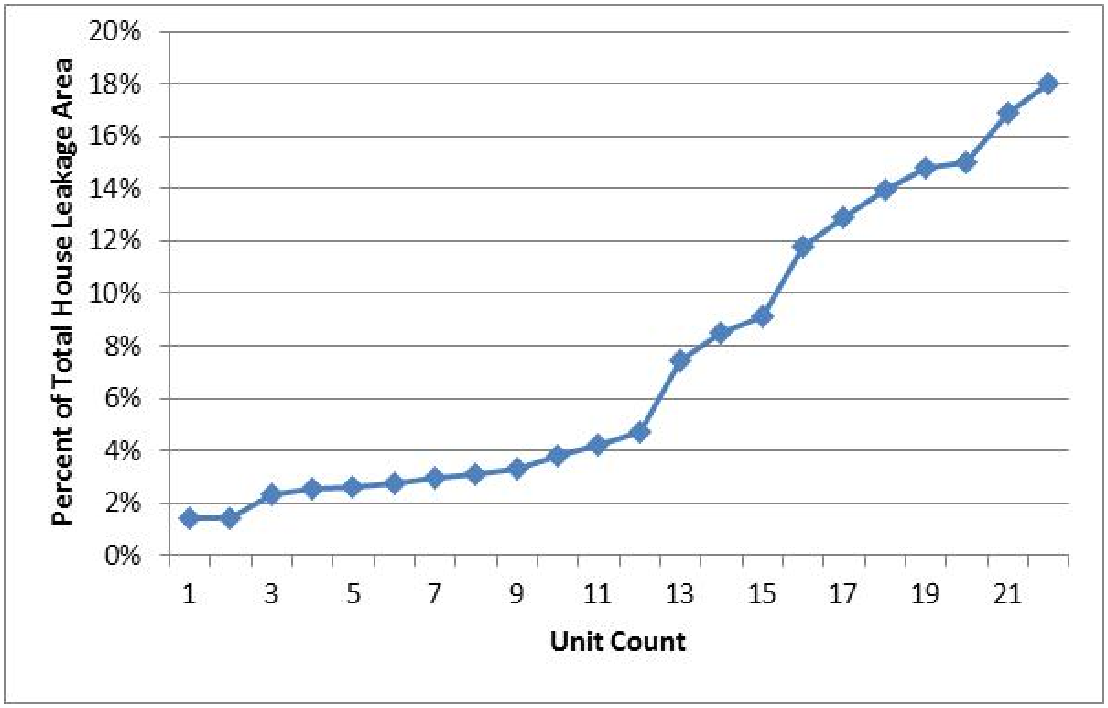
Figure 53: Fireplace Leakage



Source: Data – Rick Chitwood

Half of the 23 fireplaces in single-family dwellings contributed less than 5 percent of the total leakage area of the homes. The remaining half contributed between 7 percent and 18 percent of the total house leakage area. Figure 54 displays the range of leakage percentages for the single-family units. One single-family unit had two fireplaces (count number 15 in Figure 54).

Figure 54: Fireplace Leakage as Percent to Total House Leakage Area



Source: Data – Rick Chitwood

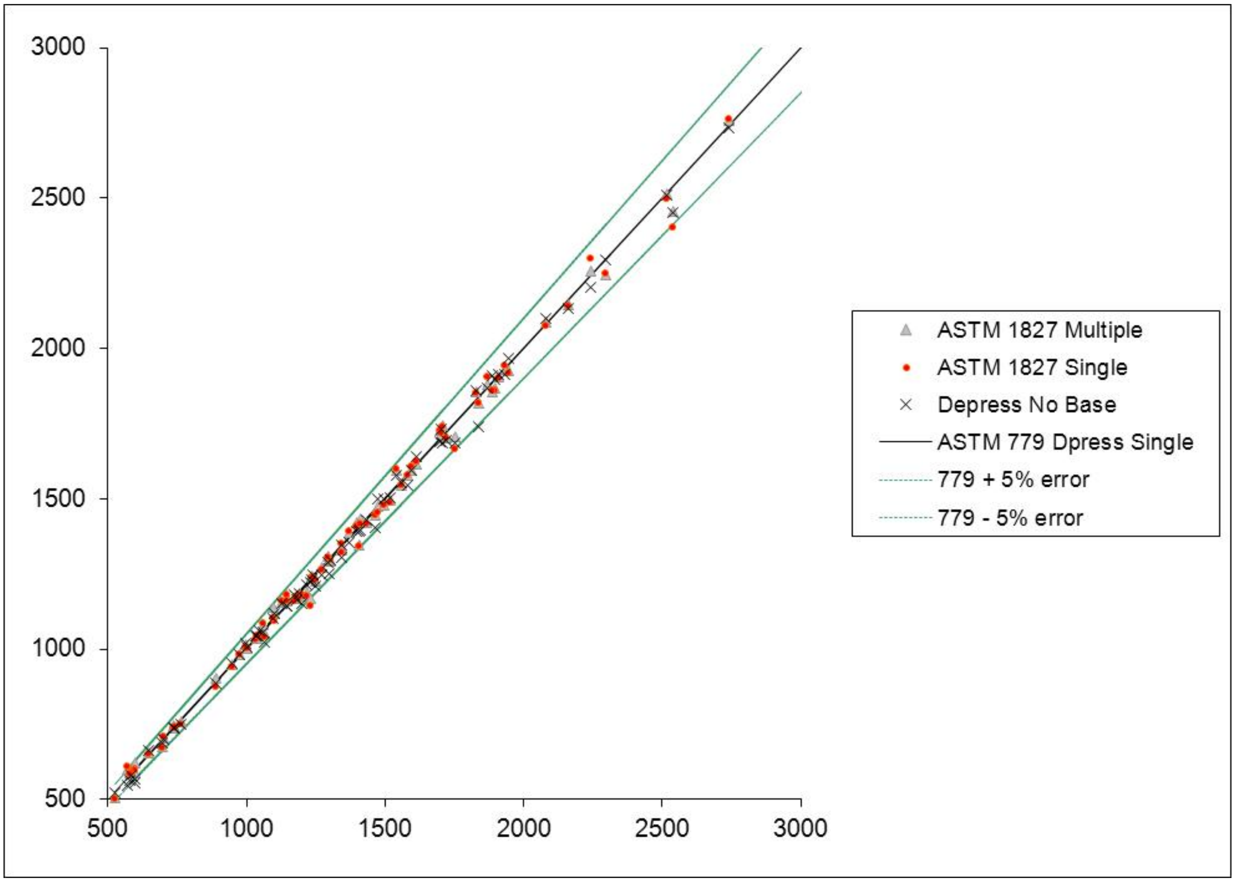
3.4.2 Comparing House Leakage Measurement Methods

The study compared four methods of measuring house leakage:

- Building shell leakage using Single Point Depressurization at 50 pascals
- Building shell leakage with range hoods and fans sealed using Single Point Pressurization at 50 pascals
- Building shell leakage using ASTM E779-03 (automated, both pressurized and depressurized)
- Building shell leakage using ASTM 1827-02 (five tests depressurized).

The study found very little difference between the various house leakage measurements. Figure 55 shows that most measurements lay within ± 5 percent of the ASTM 779 single test depressurization results.

Figure 55: Building Leakage Test Comparison

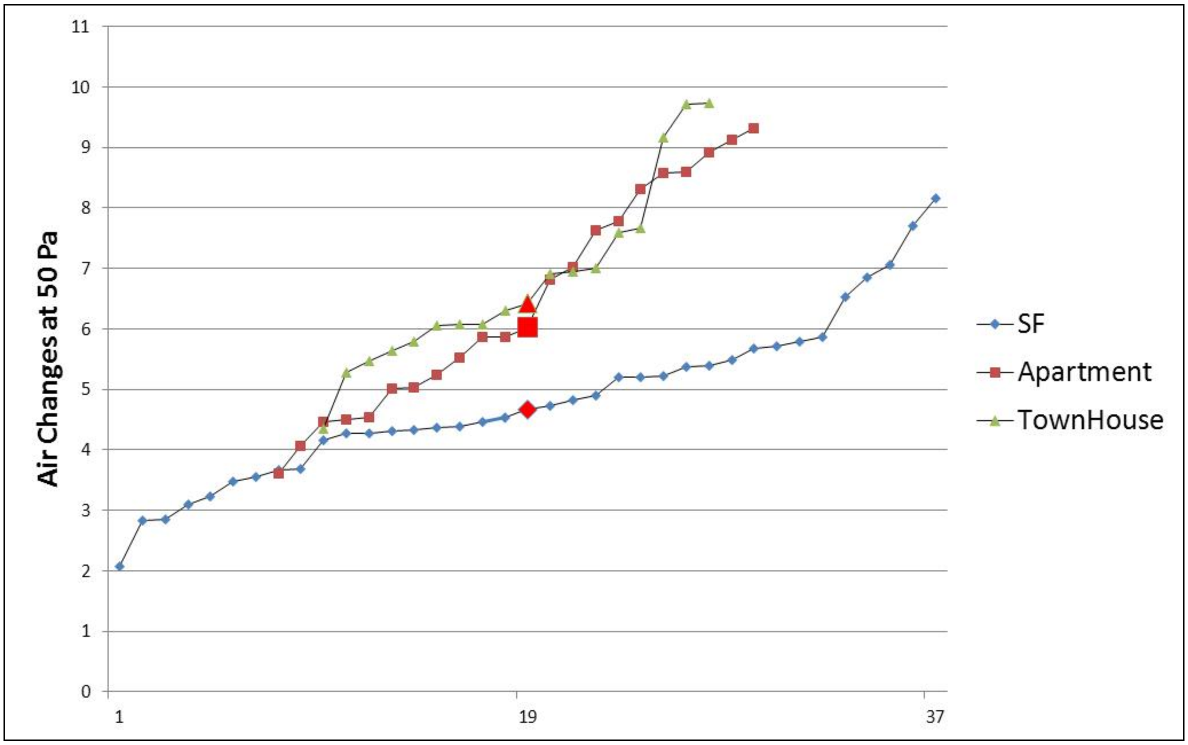


Source: Data – Rick Chitwood

3.4.3 House Leakage

The study found that single-family homes were very tight, with a median value of 4.66 Air Changes per Hour at 50 pascals pressure (ACH50). Multifamily homes, however, were substantially leakier to outside the individual unit. The median leakage for multifamily units was over 6 ACH50. Both apartments and townhomes showed higher leakage to outside the unit than single-family homes. These results are displayed in Figure 56.

Figure 56: Building Air Change Rates at 50 Pascals



Source: Data – Rick Chitwood

3.4.4 Leakage Between Conditioned Space and Undesirable Locations (Attics and Attached Garages)

Leakage areas high in the building to and from the attic produce substantial energy efficiency losses since they introduce superheated air in the summer and are at the point of maximum positive pressure in the winter. Such leaks result from insufficient attention to detail at the top of the building cavities. Figure 57 is an example of one such error.

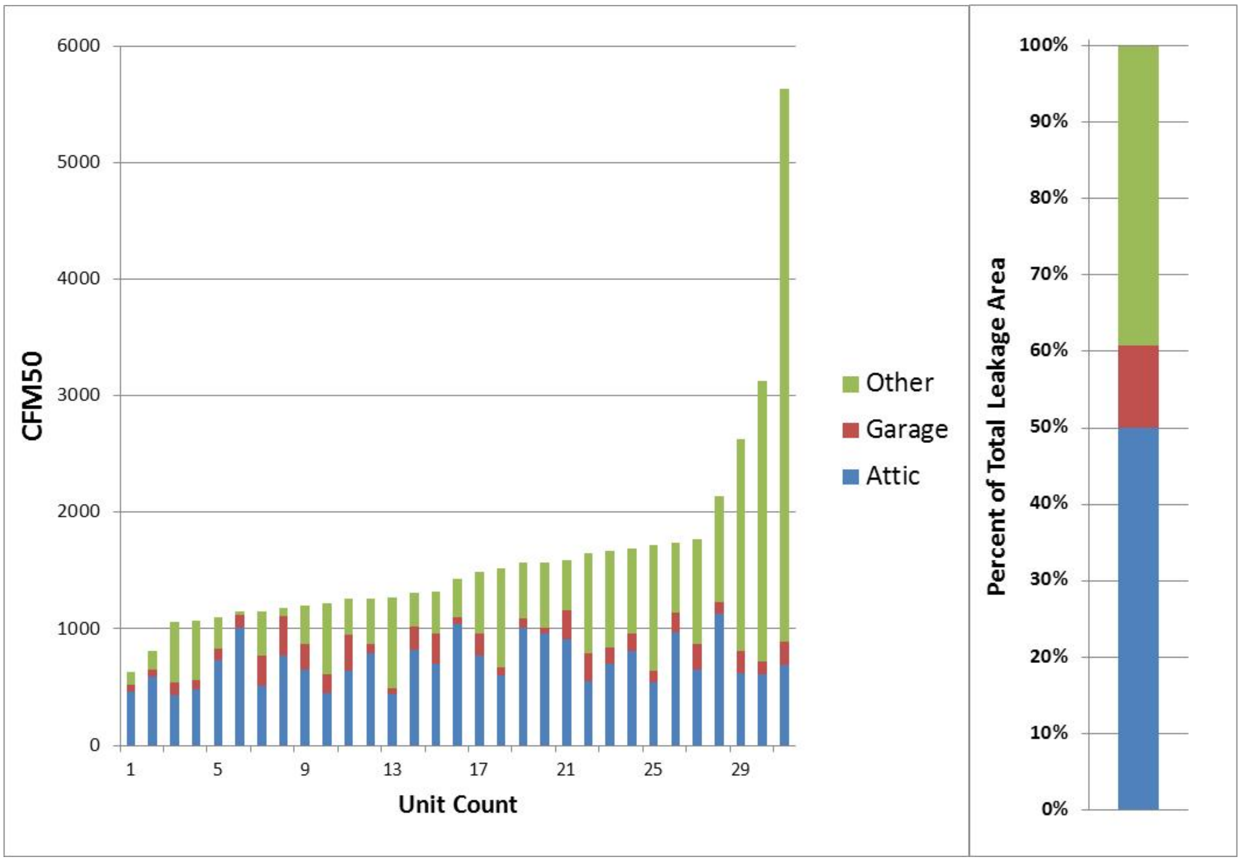
Figure 57: Leakage between Conditioned Space and Attic

Photo Credit: Rick Chitwood

Leakage paths between conditioned spaces and attached garages have been responsible for carbon monoxide and benzene intrusion into houses.

As shown in Figure 58, for homes with attics and attached garages, an average 51percent of the total house air leakage area is between the conditioned space and the attic and 11 percent is to/from the garage.

Figure 587: Leakage Areas Between House and Attic/Attached Garage



Source: Data – Rick Chitwood

CHAPTER 4:
Summary, Discussion, and Recommendations

4.1 Summary

4.1.1 Recruiting

The Efficiency Characteristics and Opportunities for New California Homes (ECO) project recruited 80 newly constructed homes from the electricity customers of Pacific Gas and Electric Company, Southern California Edison, and San Diego Gas and Electric Company. Forty single-family and 40 multifamily homes first connected to the electric grid in 2007 were randomly recruited from the utility customer lists, stratified by new construction in the three digit zip code prefixes. Field visits were conducted and included a census of lighting for each home, multiple tests on the heating and air conditioning system, and multiple tests of the building air leakage performance.

In a second phase of this project, additional cooling system tests and simple upgrades were performed on 10 of the single-family homes.

4.1.2 Lighting

This lighting census provides researchers information about many previously unknown statistics on actual residential homes. The data set is available to researchers. Seventy-eight percent of the lighting wattage in single-family and town houses were incandescent. In apartments, 68 percent of the wattage was in incandescent lamps. The majority of the lamp wattage was controlled by switches while dimmers controlled 10 percent of the wattage in apartments and 33 percent of the wattage in single-family homes.

4.1.4 HVAC Phase One

The predominant heating and cooling system (HVAC) in apartments was a combined hydronic coil from the water heater and an evaporator coil from a split air conditioner. The predominant HVAC system in single-family and town homes was a split system air conditioner with a gas furnace.

The average air conditioner performed well below expectations with low airflow across the indoor coils averaging 322 CFM per ton of cooling capacity. The 10 combined hydronic units had an average airflow of only 280 CFM per ton. Airflow across the indoor coil is a statistically significant predictor of the sensible efficiency of air conditioning systems. On the units in this sample, an increase of 100 CFM per ton would translate to a 14 percent increase in sensible cooling capacity.

The split system air conditioner evaporator blowers drew an average 650 watts per 1000 CFM of airflow, 178 percent of the watt draw assumed in the SEER test procedure. Zoned HVAC systems were the largest offenders drawing 206 percent of the SEER assumed fan wattage with all dampers open and 233 percent of the assumed fan wattage with the main zone calling for cooling.

The 2008 California Title 24 Standards provide a prescriptive standard for cooling airflow and fan power. Only 28 percent of the systems tested met the standard. The predominant cause of low airflow in these units was excessively high return system static pressure (including the filter).

Low airflow was also a problem in the heating mode with the temperature rise through the furnace higher than desirable. Low airflow not only lowers the efficiency of the furnace, it can also cause the unit to cycle the gas off and on by the limit switch, potentially increasing heat exchanger fatigue and corrosion.

Thermostatic expansion valves (TXVs) are used to control the flow of refrigerant in an air conditioner. A TXV controls the flow by providing a nearly constant temperature difference between the refrigerant entering the indoor coil and the refrigerant exiting the coil. Title 24 provides a liberal requirement that this temperature difference be between 4°F and 25°F. Thirty-one percent of the units tested failed this criterion, indicating problems with the TXVs and/or refrigerant charge or flow restrictions.

One indicator of proper refrigerant volume is a measurement called subcooling. Air conditioners perform well over a range of subcooling. When liberal requirements are applied to the units in this sample, an additional 12 percent of the units indicated errors in installing the units that resulted in too little or too much refrigerant.

Seventy-eight percent of the ducted systems had some or all of the ducts in the attic. This location provides the most severe case for conduction losses and return leakage problems. Most of the apartments had their ducts within the conditioned space or within a soffit. The median duct leakage for single-family homes met the Title 24 prescriptive standard. Townhomes and apartments however showed higher leakage rates to outside the units.

Duct leakage causes three problems: conditioned supply air loss, return air dilution (often with attic air) and additional house infiltration. The additional house infiltration is due to pressures in the house caused by an imbalance between supply and return leaks. The median imbalance for these ducted systems was 17 percent of the leakage.

The vast majority of the air conditioners/furnaces are in the attic. This location is very hot in the summer and cold in the winter. Since the Federal Test Standard classifies the cabinet around the furnace blower as part of the duct system (not part of the furnace), the majority of the blower cabinets are not insulated, causing excessive heat gain in the summer and heat loss in the winter.

4.1.5 HVAC Phase Two

Ten single-family units were the subject of additional investigations and repairs. One unit was not operational from the time the house was first occupied. That unit had multiple refrigerant leaks and the control wiring was never connected. The repairs/upgrades on the other nine units resulted in an average efficiency improvement of 24 percent.

The most common and successful repair was reducing the flow resistance of the return duct system between the house and the furnace/air conditioner. These changes were commonly an increase in return grille size (often adding an additional return grille), adding additional or larger ducts between the grilles and the furnace, installing "high flow" rather than "high efficiency" air filters, and revising the entry into the furnace blower compartment.

The efficiency of one unit increased by 19 percent when the refrigerant was removed and replaced with clean, pure refrigerant. This efficiency improvement indicates that non-condensables were probably contaminating the refrigerant. The efficiency of one other unit increased by 35 percent when the existing refrigerant was removed, the circuit de-humidified (remove moisture – a non-condensable) and proper refrigerant volume installed. Removing refrigerant and replacing it with clean refrigerant in five other cases made no significant change in efficiency. Two out of seven of the units in this sample (29 percent) are judged to have had contaminated refrigerant.

The efficiency of the only zoned unit in the ten was increased by 17 percent when the zoning bypass was eliminated.

One unit had the PSC fan motor replaced by a BPM fan motor adjusted to the same airflow. The fan watt draw dropped by 102 watts and the efficiency increased by 4 percent.

4.1.6 Fireplace Air Leakage

Fireplaces in single-family homes produced a range of air leakage to outside between less than 2 percent of the house leakage to 18 percent of house leakage. Almost half of the 23 fireplaces in the units studied were responsible for between 7 percent and 18 percent of the total house leakage area.

4.1.7 House Air Leakage

A variety of house leakage test methods were compared. The study concludes that a single point method at 50 pascals provides results within 5 percent of the other methods.

The median of single-family homes were found to be reasonably tight (4.66ACH50). The leakage to outside the units for apartments and townhomes was significantly higher (apartment median 6.02ACH50, townhouse median 6.42ACH50).

The residences in this study that have both attached garages and accessible attics, on average have 51 percent of the leakage area between the conditioned space and the attic. These residences also have an average of 11 percent leakage between the garage and conditioned space.

4.2 Discussion

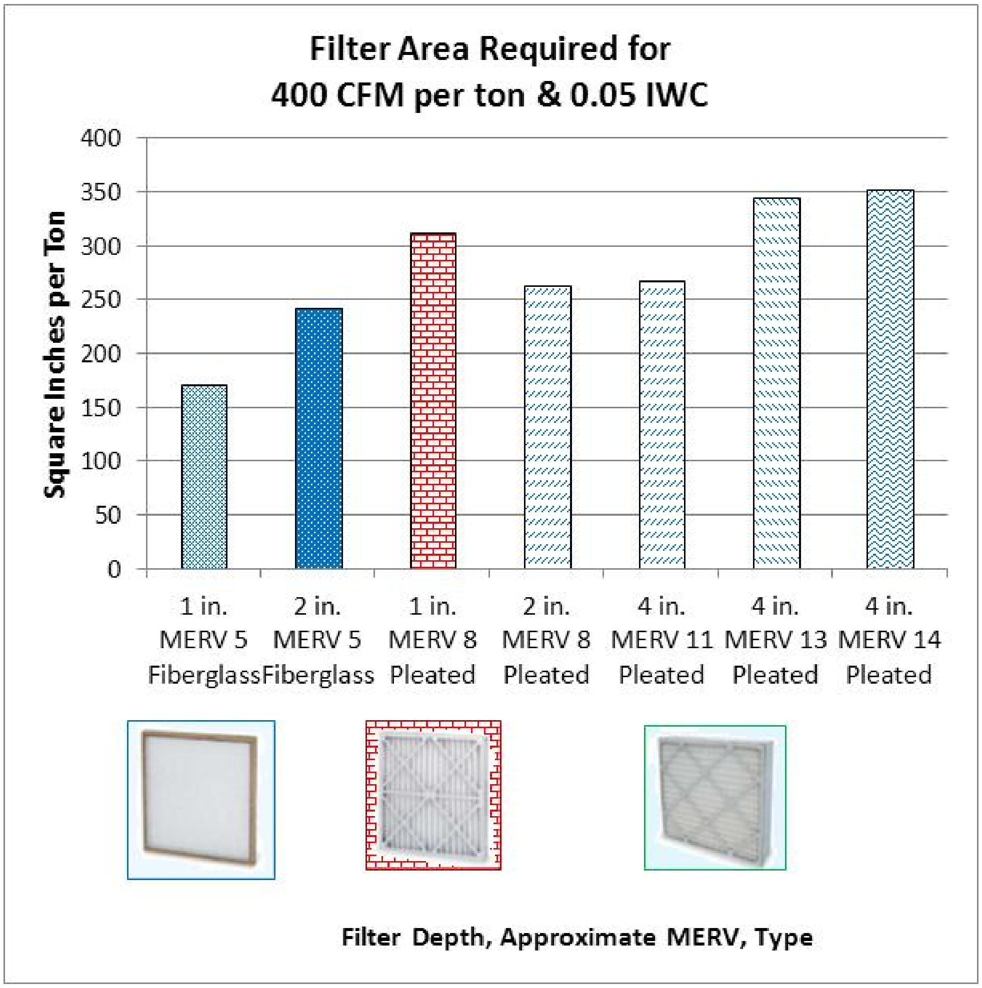
In the areas of heating, ventilating and air conditioning a major problem in new homes is low airflow through the furnace and across the evaporator coil. This study showed the primary driver of the low airflow is the restrictive nature of the return system. There are three components that lead to the high flow resistance in the return systems: filter resistance, ducts and their fittings, and the entry conditions to the furnace/air handler blower compartment.

The return systems in California homes are rather simple with one or two filter grilles, ducting to the furnace and entry into the blower compartment. With prescriptive criteria this is an important area that building inspectors can observe and determine passing or failing without additional instrumentation or much effort.

4.2.1 Filter Flow Resistance

The standard filters assumed by the manufacturers and installed "in the old days" were simple 1 in. deep fiberglass mats with the ability to stop large objects like pet hair and dryer lint. Increasingly, homeowners are substituting 1 in. "high efficiency" pleated filters that offer about twice the resistance of the fiberglass filters to airflow when clean. These filters are nearly "a brick wall" to airflow. This is a growing problem and a significant contributing factor to low sensible efficiency. Figure 59 displays the filter area needed per ton to have a clean filter drop of 0.05 IWC at 400 CFM per ton.

Figure 59: Required Air Filter Area for 400 CEM per Ton at 0.50 IWC



Source: Data – Air Handler® Catalog

To overcome the negative effects of the use of 1 in. deep pleated filters, the filter area will need to be increased to over 300 square inches per ton.

4.2.2 Return Duct System Flow Resistance

The total external static pressure specification for most furnaces is 0.50 IWC. The median resistance of split system evaporator coils in this study is 0.21 IWC at a median airflow of about 350 CFM per ton. At 400 CFM per ton this would increase to 0.27 IWC. With a pressure drop of 0.05 IWC at the filter, a drop of 0.03 for the grille, and 0.27 for the evaporator coil, 0.15 IWC remains for the supply ducts and the return ducts. The vast majority of the 0.15 needs to be available to the more complex supply ductwork. Using 0.0375 for the return duct work we prescribe the duct sizes in Table 25. Any turn over 75° requires a metal elbow.

Table 25: Prescriptive Return Systems

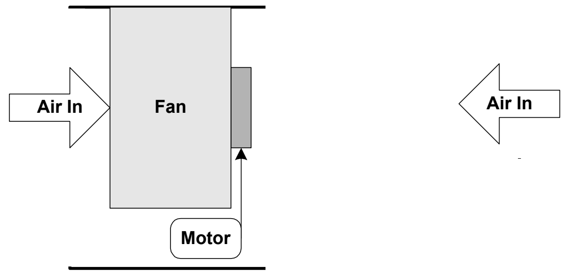
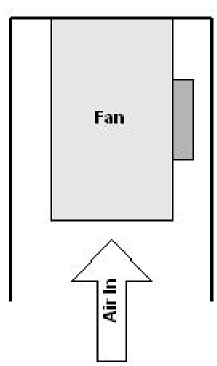
Tonnage	1.5	2	2.5	3	3.5	4	5
	One Return Acceptable		**Two Returns Required**				**Three Returns Required**
Filter Area minimum square in. 4 in. filter stop depth min. 10 in. can depth	450	600	750	900	1050	1200	2000
Return 1 Metal Elbow minimum diameter	16 in.	18 in.	18 in.	18 in.	18 in.	18 in.	18 in.
Return 1 Duct minimum diameter	16 in.	18 in.	18 in.	18 in.	18 in.	18 in.	18 in.
Return 1 Duct maximum length	30 ft	30 ft	30 ft	30 ft	30 ft	30 ft	30 ft
Return 2 Metal Elbow minimum diameter			10 in.	12 in.	14 in.	18 in.	18 in.
Return 2 Duct minimum diameter			10 in.	12 in.	14 in.	18 in.	18 in.
Return 2 Duct maximum length			30 ft	30 ft	30 ft	30 ft	30 ft
Return 3 Metal Elbow minimum diameter							14 in.
Return 3 Duct minimum diameter							14 in.
Return 3 Duct maximum length							30 ft

Source: Data – Rick Chitwood;Analysis – Proctor Engineering Group, Ltd.

4.2.3 Entry into Blower Compartment

The entry into the blower compartment is the remaining determinant of the airflow performance of the return system. Prior research shows that the priority of entry is as shown in Table 26.

Table 26: Priority Entry Into Blower Compartment

Inlet Both Sides	
Bottom or Back Inlet	

Source: Proctor Engineering Group, Ltd. Report CEC 500-2008-056

For single entry points, the bottom is a priority to side entry. Given these priorities, entry for single return system should be from the bottom or back; entry for double returns should be from the bottom or back and side. The priority for triple returns should be bottom and two sides when practical.

4.3 Recommendations

The study team has the following recommendations as a result of this study:

1. Title 24–2013 should mandate a confirmed airflow greater than or equal to 400 CFM per ton and a fan watt draw less than or equal to 0.510 watts per CFM; with an acceptable alternative of the return system sizes specified in Table 25, as verified by the building inspector.

2. Title 24–2013 should mandate labeling HVAC return locations with the size, maximum clean filter pressure drop at 400 CFM per ton clean filter airflow.

3. Title 24–2013 should mandate that all HVAC filters sold in California be labeled with a standardized clean filter pressure drop and clean filter airflow table.

4. Title 24–2013 should mandate a confirmed total duct leakage less than or equal to 24 CFM25 per ton for single-family homes and townhomes.

5. Title 24–2013 should mandate a confirmed total duct leakage of less than or equal to 48 CFM25 per ton for apartments regardless of the location of the duct systems.

6. Title 24–2013 ACM should calculate energy consumption based on 17 percent duct leakage imbalance.

7. Title 24–2013 ACM should calculate energy consumption based on 51 percent of the house air leakage area between the occupied space and the attic.

8. Title 24–2013 should clearly define that the fan cabinet and return plenum on furnaces is part of the duct system and mandate that it must be insulated to the levels specified for duct systems in the space in which they are located.

9. Title 24–2013 should revise the acceptable limits for HERS inspections of TXV air conditioners. The limits should be greater than 2° F and less than or equal to the manufacturer's target subcooling of 8°F.

10. California Energy Commission should sponsor additional field research to determine the extent of non-condensables in the refrigerant of newly installed air conditioners.

11. Title 24–2013 should mandate that any zoned system must not have a bypass from the supply to the return and that the airflow in all potential operating modes meet recommendation number 1.

12. For single-family buildings and townhouses, Title 24–2013 should mandate a confirmed building shell air leakage of less than or equal to 4 ACH at 50 pascals using a single point test.

13. For multifamily buildings, Title 24–2013 should mandate a confirmed unit air leakage of less than or equal to 6 ACH at 50 pascals using a single point test.

14. Title 24–2013 should mandate that air conditioner condensing units may not be placed within 5 ft of a dryer vent.

15. Title 24–2013 should mandate that there be no obstruction within 5 ft. of the condenser coil inlet and condenser coil outlet.

16. Title 24–2013 should mandate that furnace heat rise must not exceed the manufacturer's specification.

List of Acronyms

AC	Air Conditioner
ACCA	Air Conditioning Contractors of America
ACEEE	American Council for an Energy-Efficient Economy
ACH	Air Changes Per Hour
ACM	Alternative Calculation Method
AFUE	Annual Fuel Utilization Efficiency
ASHRAE	American Society of Heating Refrigeration and Air-Conditioning Engineers
ASTM	American Society for Testing and Materials
BPM	Brushless Permanent Magnet
BTU	British Thermal Unit
BTUH	British Thermal Unit Per Hour
CASE	Codes and Standards Enhancement
CFL	Compact Fluorescent Light
CFM	Cubic Feet Per Minute
ECO	Efficiency Characteristics and Opportunities for New California Homes Project
EER	Energy Efficiency Ratio
EER_S	Energy Efficiency Ratio - Sensible
FLA	Full Load Amps
HERS	[California] Home Energy Rating System
HVAC	Heating, Ventilation And Air Conditioning
IWC	Inches Water Column
LED	Light Emitting Diode
$NEER_S$	Normalized Energy Efficiency Ratio - Sensible
OEHHA	[California] Office of Environmental Health Hazard Assessment
PG&E	Pacific Gas & Electric Company
PIER	Public Interest Energy Research Program
Ppb	Parts Per Billion
PSC	Permanent Split Capacitor
RD&D	Research, Development And Demonstration
REL	Reference Exposure Level
RLA	Rated Load Amps
SEER	Seasonal Energy Efficiency Ratio
TXV	Thermostatic Expansion Valve

APPENDIXA

Field Survey Data Collection Form

California New Home Energy Survey

California New Home Energy Survey #__________ Date of Data Collection:_______________________

Official Site ID# __________

Occupant: ____

Address: ____

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Notes and Observations: ____

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☐ Production Builder ☐ Other

☐ Photo#1 – House from street ☐ Multi-family ☐ Single-family

Type of Exterior Finish (stucco, board, etc.) ____

Signs of Moisture Problems in Bathrooms or Kitchen

1. ____
2. ____
3. ____

☐ Photo#2 – Evidence of Moisture Problems

Notes: ____

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Lighting Page 1

□ Home seems to be built under 2005 Standards □ Home seems to be built under 2001 Standards

Room	Light Type (hard wired or portable, CFL, CFL base type, incan., LED	Wattage	Control (Sw, Dim, Occ.)

Lighting Page 2

Room	Light Type (hard wired or portable, CFL, CFL base type, incan., LED	Wattage	Control (Sw, Dim, Occ.)

House Physical Characteristics

California Climate Zone _________ Number of Bedrooms _________ Number of Stories __________

Conditioned Floor Area ____________ (square feet) Ceiling Height _________________________(feet)

House Volume _______________ (cubic feet) LBL Factor for ACH ___________________

HVAC

Furnace Make/Model# ____

Furnace Input _______________________ (Btu/H)

Cooling Coil Make/Model# ____

Fan/motor type: ☐ Multi-speed Direct-drive Blower

Fan Speed: Heating _________Cooling _________ Number of Speeds _______

☐ Variable-speed direct-drive Blower

Dip Switch Setting: Heating __________Cooling __________ (CFM)

☐ Photo #3 – Furnace cabinet and cooling coil case

☐ Photo #4 – TXV installed: ☐ YES ☐ NO
(accessible TXV: ☐ YES ☐ NO, TXV bulb location: ☐ Inside ☐ Outside)

☐ Photo #5 – The furnace fan control board (furnace cabinet insulated: ☐ YES ☐ NO)

☐ Photo #6 – roof sheathing (radiant barrier: ☐ YES ☐ NO)

Air Handler maximum static pressure rating ______________________ (inches WC)

Furnace; stand-by watts, ____ (watts)

Furnace; stand-by watts, induced draft blower, and gas valve, _________________________________ (watts)

Furnace; stand-by watts, induced draft blower, gas valve, and furnace fan _______________________ (watts)

Heating mode static pressure at the **furnace inlet** (30 second average) ________________________ (Pascals)

Heating static pressure at the **furnace outlet** (30 second average) ______________________ (Pascals, NSOP)

Heating static pressure at the **cooling coil outlet** (30 second average) _________________________ (Pascals)

Heating air flow (10 second average, after 5 min. run time) ________________________ (Un-corrected CFM)

Condensing Unit Make/Model# ____ Nominal Tons _______

☐ Photo #7 – Condensing Unit

Cooling mode fan power ____ (watts)

Air Handler **Power Factor** in Cooling Mode _____________________ (decimal)

Cooling mode static pressure at the **furnace inlet** (30 second average) ________________________ (Pascals)

Cooling static pressure at the **furnace outlet** (30 second average) ______________________ (Pascals, NSOP)

Cooling static pressure at the **cooling coil outlet** (30 second average) _________________________ (Pascals)

Cooling air flow (10 second average reading, after 5 min. run time) _________________ (Un-corrected CFM)

Circulation mode fan power (if present) ____ (watts)

Circulation air flow (10 second average, after 1 min. run time) ______________________(Un-corrected CFM)

Type of air filter installed (full label info.) ____

Filter Size ____

Filter Static Pressure, cooling mode ____ (Pascals)

☐ Photo #8 – Air filter

Notes: ____

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Refrigerant Charge

	Test 1	Test 2	Test 3	
Condenser Air Entering Temperature	________	________	________	°F
Return Air Wet Bulb Temperature	________	________	________	°F
Return Air Dry Bulb Temperature	________	________	________	°F
Supply Air Dry Bulb Temperature	________	________	________	°F
Supply Plenum Temperature Measurement Range	________________		°F	
Temperature Split	________	________	________	°F
Temperature Split Target	________	________	________	°F
Measured Air Flow ________________	Temp Split OK (+/- 3°F): ☐ YES ☐ NO			
Suction Line Temperature	________	________	________	°F
Evaporator Saturation Temperature	________	________	________	°F
Superheat	________	________	________	°F
Superheat Target (5°F – 10°F for TXV)	________	________	________	°F
Condenser Saturation Temperature	________	________	________	°F
Liquid Line Temperature	________	________	________	°F
Subcooling	________	________	________	°F
Subcooling Target	________	________	________	°F
Direct Measure of Subcooling	________	________	________	°F
Suction (low side) Pressure	________	________	________	PSI
Discharge (high side) Pressure	________	________	________	PSI

Ounces Added __________ (oz.) Condensing Unit RMS Wattage ________ (watts)

Condensing Unit Data Plate: Compressor Current ________ (amps) Fan Current ________ (amps)

Condensing Unit VA: Current ________ (amps) ________ (volts)

VA (apparent power): ________ (watts) Power Factor ________ (decimal)

Room Air Flows

Room	Type of Supply Grille and Size	CFM (dry coil)	Delivery Temperature	Room Pressures

Return Grille Size ______________________ Start Temperature ___________ Final Temperature ___________

Attic Temperature ___________

Duct Leakage

Duct Leakage ______________ (CFM_{25}) Duct Leakage to the Outside of the Home ______________ (CFM_{25})

Half Nelson Supply ________________ Pascals Half Nelson Return _________________ Pascals

Duct Location ____

Fireplace Face Leakage

Fireplace 1 Leakage ______________ (CFM_{50}) Leakage to the Outside of the Home ______________ (CFM_{50})

Fireplace Model # ____

Fireplace Location ____

☐ Photo #9 – Fireplace #1

Fireplace 2 Leakage ______________ (CFM_{50}) Leakage to the Outside of the Home ______________ (CFM_{50})

Fireplace Model # ____

Fireplace Location ____

☐ Photo #10 – Fireplace #2

Notes: ____

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House Pressurization Testing

☐ Simple Single Point House Depressurization Test (no baseline adj.) ________________________ (CFM_{50})

☐ Simple Single Point House Pressurized Test (no baseline adj., exhaust fans covered) ___________ (CFM_{50})

Number of fans sealed __________ Range hood sealed __________

☐ **ASTM E779-03 Air Leakage Test** (automated, both pressurized (with sealed exhausts) and depressurized)

Pressurized ________________________ (CFM_{50} from software)

Depressurized 1A ________________________ (CFM_{50} from software)

Depressurized 1B ________________________ (optional CFM_{50} from software if less that 10:1 ratio)

Depressurized 2 ________________________ with garage door open (CFM_{50} from software)

Depressurized 3 ________________________ with attic hatch open (CFM_{50} from software)

File name ____

☐ **ASTM E1827-02** Air Leakage Test (DG-700, depressurized, 10 second averages, any vents in garage sealed)

	House CFM_{50}	Attic Pressure Pa	Garage Pressure Pa	
Test 1				
Test 2				
Test 3				
Test 4				
Test 5				
Test 6				(garage vents open)

☐ Check Water Heater

APPENDIX B

Field Survey Data Collection Form for Re-test

California New Home Energy Survey
HVAC System Re-test

California Home Energy Survey #__________ Date(s) of Data Collection: ____________________________

Official Site ID# ______________

Occupant: ____

Address: ____

____

Notes and Observations: ____

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HVAC

Confirm Furnace Make/Model#____

Confirm Furnace Input _______________________ (Btu/H)

Confirm Cooling Coil Make/Model# ____

Confirm Characteristics:

☐ Multi-speed Direct-drive Blower

Fan Speed: Heating _________Cooling _________ Number of Speeds _______

☐ Variable-speed direct-drive Blower

Dip Switch Setting: Heating __________Cooling __________ (CFM)

TXV installed: ☐ YES ☐ NO

TXV accessible: ☐ YES ☐ NO, TXV bulb location: ☐ Inside ☐ Outside)

Furnace heat exchanger area insulated: ☐ YES ☐ NO)

Furnace blower compartment insulated: ☐ YES ☐ NO)

CONFIRM HEATING MEASUREMENTS (Run at least 5 minutes before test)

Air Handler maximum static pressure rating	________________	inches WC
Furnace; stand-by watts	________________	W
Furnace; stand-by watts, induced draft blower, and gas valve	________________	W
Furnace; stand-by watts, induced draft blower, gas valve, and furnace fan	________________	W
Heating mode static pressure at the furnace inlet (5 second average)	________________	Pa
Heating static pressure at the furnace outlet (5 second average)	________________	Pa, NSOP
Heating static pressure at the cooling coil outlet (5 second average)	________________	Pa
Heating air flow (5 second average, un-corrected T.F. CFM)	________________	CFM
Furnace temperature rise	________________	^{o}F
Allowable temperature rise range	________________	^{o}F

CONFIRM COOLING MEASUREMENTS

Condensing Unit Make/Model# ____ Nominal Tons ________

Condensing Unit Data Plate: Compressor Current _________________ A

Condensing Unit Data Plate: Fan Current _________________ A

(Run at least 15 minutes before test – time sensitive measurements in BOLD)

Cooling fan wattage _________________ W

Cooling fan power factor _________________ decimal

Cooling static pressure at furnace inlet (5 second average) _________________ Pa

Cooling static pressure at furnace outlet (5 second average) _________________ Pa, NSOP

Cooling static pressure at cooling coil outlet (5 second average) _________________ Pa

Cooling air flow (5 second average, un-corrected T.F. CFM) _________________ CFM

Air Flow Notes: ____

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Start Return Grill Wet Bulb Temperature _________________ **^{o}F**

Start Return Grill Dry Bulb Temperature _________________ **^{o}F**

Start Return Plenum Wet Bulb Temperature _________________ **^{o}F**

Start Return Plenum Dry Bulb Temperature _________________ **^{o}F**

Start Supply Plenum Dry Bulb 0.75/0.75 Left _________________ **^{o}F**

Start Supply Plenum Dry Bulb 0.75/0.75 Right _________________ **^{o}F**

Start Attic Temperature _________________ **^{o}F**

Temperature Split _________________ ^{o}F

Temperature Split Target _________________ ^{o}F

Start Condenser Air Entering Temperature _________________ **^{o}F**

Start Condensing Unit RMS Wattage _________________ **W**

Power Factor: ________ volts, ________ Amps, ________ VA _________________ decimal

Suction Line Temperature ________________ **°F**

Evaporator Saturation Temperature ________________ **°F**

Superheat ________________ **°F**

Superheat Target (5°F – 10°F for TXV) ________________ °F

Condenser Saturation Temperature ________________ **°F**

Liquid Line Temperature ________________ **°F**

Subcooling ________________ **°F**

Subcooling Target ________________ °F

Discharge (high side) Pressure ________________ **PSIG**

Suction (low side) Pressure ________________ **PSIG**

Pressure across TXV ________________ PSIG
(If less than 150 psi a retest with restricted condenser outlet is needed)

Finish Condenser Air Entering Temperature ________________ **°F**

Finish Condensing Unit RMS Wattage ________________ **W**

Finish Return Grill Wet Bulb Temperature ________________ **°F**

Finish Return Grill Dry Bulb Temperature ________________ **°F**

Finish Return Plenum Wet Bulb Temperature ________________ **°F**

Finish Return Plenum Dry Bulb Temperature ________________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Left ________________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Right ________________ **°F**

Finish Attic Temperature ________________ **°F**

Notes: ____

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Confirm Room Air Flows (Cooling Mode Wet Coil)

Room	CFM (wet coil)	Delivery Temperature

Circulation mode fan power ________________(W) air flow __________________ (un-corrected T.F. CFM)
(If ECM motor)

Air Filter Installed ____

Filter Size ____

Filter Static Pressure ____

System Revision # 1

Revision Description: ____

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Notes on System Revisions:

1. Remove Refrigerant Charge, Evacuate to 500 microns or less. Isolate the system from the vacuum pump and let sit for at least 5 minutes. The micron gauge should not raise more than 300 microns above the initial vacuum level. Install new refrigerant. Set charge.
2. Describe initial TXV installation and take photos.
3. Return system upgrade
4. If fan motor is ½ HP or less replace with BPM motor and adjust BPM motor to same flow (using the Supply plenum pressure match method).

Removed PSC Motor Make____________ Model_____________ W_____________ StaticP____________

Installed BPM Motor Make____________ Model_____________ W_____________ StaticP____________

<u>AFTER REVISION #1 HEATING MEASUREMENTS</u> (Run at least 5 minutes before test)

Furnace; stand-by watts, induced draft blower, and gas valve ________________ W

Furnace; stand-by watts, induced draft blower, gas valve, and furnace fan ________________ W

Heating mode static pressure at the <u>furnace inlet</u> (5 second average) ________________ Pa

Heating static pressure at the <u>furnace outlet</u> (5 second average) ________________ Pa, NSOP

Heating static pressure at the <u>cooling coil outlet</u> (5 second average) ________________ Pa

Heating <u>air flow</u> (5 second average, un-corrected T.F. CFM) _______________ CFM

Furnace temperature rise ____ ^{o}F

Allowable temperature rise range ________________ ^{o}F

AFTER REVISION #1 COOLING MEASUREMENTS

(Run at least 15 minutes before test – time sensitive measurements in BOLD)

Cooling fan wattage ________________ W

Cooling fan power factor ________________ decimal

Cooling static pressure at furnace inlet (5 second average) ________________ Pa

Cooling static pressure at furnace outlet (5 second average) ________________ Pa, NSOP

Cooling static pressure at cooling coil outlet (5 second average) ________________ Pa

Cooling air flow (5 second average, un-corrected T.F. CFM) ________________ CFM

Air Flow Notes: ____

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Start Return Grill Wet Bulb Temperature ________________ **^oF**

Start Return Grill Dry Bulb Temperature ________________ **^oF**

Start Return Plenum Wet Bulb Temperature ________________ **^oF**

Start Return Plenum Dry Bulb Temperature ________________ **^oF**

Start Supply Plenum Dry Bulb 0.75/0.75 Left ________________ **^oF**

Start Supply Plenum Dry Bulb 0.75/0.75 Right ________________ **^oF**

Start Attic Temperature ________________ **^oF**

Start Condenser Air Entering Temperature ________________ **^oF**

Condensing Unit RMS Wattage ________________ **W**

Power Factor: ________ volts, ________ Amps, ________ VA ________________ decimal

Temperature Split ________________ ^oF

Temperature Split Target ________________ ^oF

Suction Line Temperature ______________ **°F**

Evaporator Saturation Temperature ______________ **°F**

Superheat ______________ **°F**

Superheat Target (5°F – 10°F for TXV) ______________
°F

Condenser Saturation Temperature ______________ **°F**

Liquid Line Temperature ______________ **°F**

Subcooling ______________ **°F**

Subcooling Target ______________ °F

Discharge (high side) Pressure ______________ **PSIG**

Suction (low side) Pressure ______________ **PSIG**

Pressure across TXV ______________ PSIG
(If less than 150 psi a retest with restricted condenser outlet is needed)

Furnish Condenser Air Entering Temperature ______________ **°F**

Finish Condensing Unit RMS Wattage ______________ **W**

Finish Return Grill Wet Bulb Temperature ______________ **°F**

Finish Return Grill Dry Bulb Temperature ______________ **°F**

Finish Return Plenum Wet Bulb Temperature ______________ **°F**

Finish Return Plenum Dry Bulb Temperature ______________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Left ______________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Right ______________ **°F**

Finish Attic Temperature ______________ **°F**

Notes: ____

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After Revision #1 Room Air Flows (Cooling Mode Wet Coil)

Room	CFM (wet coil)	Delivery Temperature

Circulation mode fan power _________________(W) air flow ___________________ (un-corrected T.F. CFM)
(If ECM motor)

Air Filter Installed ____

Filter Size ____

Filter Static Pressure ____

System Revision # 2

Revision Description: ____

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Notes on System Revisions:

1. Remove Refrigerant Charge, Evacuate to 500 microns or less. Isolate the system from the vacuum pump and let sit for at least 5 minutes. The micron gauge should not raise more than 300 microns above the initial vacuum level. Install new refrigerant. Set charge.
2. Describe initial TXV installation and take photos.
3. Return system upgrade
4. If fan motor is ½ HP or less replace with BPM motor and adjust BPM motor to same flow (using the Supply plenum pressure match method).

Removed PSC Motor Make____________ Model_____________ W_____________ StaticP____________

Installed BPM Motor Make____________ Model_____________ W_____________ StaticP____________

<u>AFTER REVISION #2 HEATING MEASUREMENTS</u> (**Run at least 5 minutes before test**)

Furnace; stand-by watts, induced draft blower, and gas valve ________________ W

Furnace; stand-by watts, induced draft blower, gas valve, and furnace fan ________________ W

Heating mode static pressure at the <u>furnace inlet</u> (5 second average) ________________ Pa

Heating static pressure at the <u>furnace outlet</u> (5 second average) ________________ Pa, NSOP

Heating static pressure at the <u>cooling coil outlet</u> (5 second average) ________________ Pa

Heating <u>air flow</u> (5 second average, un-corrected T.F. CFM) _______________ CFM

Furnace temperature rise ____ ^oF

Allowable temperature rise range ________________ ^oF

AFTER REVISION #2 COOLING MEASUREMENTS

(Run at least 15 minutes before test – time sensitive measurements in BOLD)

Cooling fan wattage	_________________	W
Cooling fan power factor	_________________	decimal
Cooling static pressure at furnace inlet (5 second average)	_________________	Pa
Cooling static pressure at furnace outlet (5 second average)	_________________	Pa, NSOP
Cooling static pressure at cooling coil outlet (5 second average)	_________________	Pa
Cooling air flow (5 second average, un-corrected T.F. CFM)	_________________	CFM

Air Flow Notes: ____

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Start Return Grill Wet Bulb Temperature	_________________	**°F**
Start Return Grill Dry Bulb Temperature	_________________	**°F**
Start Return Plenum Wet Bulb Temperature	_________________	**°F**
Start Return Plenum Dry Bulb Temperature	_________________	**°F**
Start Supply Plenum Dry Bulb 0.75/0.75 Left	_________________	**°F**
Start Supply Plenum Dry Bulb 0.75/0.75 Right	_________________	**°F**
Start Attic Temperature	_________________	**°F**
Start Condenser Air Entering Temperature	_________________	**°F**
Start Condensing Unit RMS Wattage	_________________	**W**
Power Factor: ________ volts, ________ Amps, ________ VA	_________________	decimal
Temperature Split	_________________	°F
Temperature Split Target	_________________	°F

Suction Line Temperature ________________ **°F**

Evaporator Saturation Temperature ________________ **°F**

Superheat ________________ **°F**

Superheat Target (5°F – 10°F for TXV) ________________ °F

Condenser Saturation Temperature ________________ **°F**

Liquid Line Temperature ________________ **°F**

Subcooling ________________ **°F**

Subcooling Target ________________ °F

Discharge (high side) Pressure ________________ **PSIG**

Suction (low side) Pressure ________________ **PSIG**

Pressure across TXV ________________ PSIG
(If less than 150 psi a retest with restricted condenser outlet is needed)

Finish Condenser Air Entering Temperature ________________ **°F**

Finish Condensing Unit RMS Wattage ________________ **W**

Finish Return Grill Wet Bulb Temperature ________________ **°F**

Finish Return Grill Dry Bulb Temperature ________________ **°F**

Finish Return Plenum Wet Bulb Temperature ________________ **°F**

Finish Return Plenum Dry Bulb Temperature ________________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Left ________________ **°F**

Finish Supply Plenum Dry Bulb 0.75/0.75 Right ________________ **°F**

Finish Attic Temperature ________________ **°F**

Notes: ____

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After Revision #2 Room Air Flows (Cooling Mode Wet Coil)

Room	CFM (wet coil)	Delivery Temperature

Circulation mode fan power ________________(W) air flow __________________ (un-corrected T.F. CFM)
(If ECM motor)

Air Filter Installed ____

Filter Size ____

Filter Static Pressure ____

Duct Leakage _____________ (CFM_{25}) **Duct Leakage w/seal inside boot** _____________ (CFM_{25})