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Residential Forced Air System Cabinet Leakage and Blower Performance

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Environmental Energy Technologies Division

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Arnold Schwarzenegger

RESIDENTIAL FORCED AIR SYSTEM CABINET LEAKAGE AND BLOWER PERFORMANCE

PIER FINAL PROJECT REPORT

Prepared For:

California Energy Commission

Public Interest Energy Research Program

Prepared By:

Lawrence Berkeley National Laboratory

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DISCLAIMER

Preface

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

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

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

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

Residential Forced Air System Cabinet Leakage and Blower Performance is the final report for the Residential Forced Air System Cabinet Leakage and Blower Performance project (contract number 500-06-007, work authorization number [insert #] or grant number [insert #]) conducted by Lawrence Berkeley National Laboratory. The information from this project contributes to PIER's Buildings End-Use Energy Efficiency Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-654-4878.

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Table of Contents

List of Figures

List of Tables

Abstract

This project evaluated the air leakage and electric power consumption of Residential HVAC components, with a particular focus on air leakage of furnace cabinets. Laboratory testing of HVAC components indicated that air leakage can be significant and highly variable from unit to unit – indicating the need for a standard test method and specifying maximum allowable air leakage in California State energy codes. To further this effort, this project provided technical assistance for the development of a national standard for Residential HVAC equipment air leakage. This standard is being developed by ASHRAE and is called "ASHRAE Standard 193P - Method of test for Determining the Air Leakage Rate of HVAC Equipment". The final part of this project evaluated techniques for measurement of furnace blower power consumption. A draft test procedure for power consumption was developed in collaboration with the Canadian General Standards Board: CSA 823 "Performance Standard for air handlers in residential space conditioning systems".

Keywords: Public Interest Energy Research (PIER) Program, air leakage, furnaces, forced air systems, residential HVAC, blowers, air flow measurement, blower power measurement

Executive Summary

Introduction

Air leakage from forced air thermal distribution systems contributes significantly to energy use in residences. In recent years much effort has gone into the sealing of the ducts in these systems but air leakage still remains in the heating and cooling equipment itself and is a barrier to achieving tight systems. The second part of this study addresses the energy used by blowers in central forced air systems. This study develops methods to measure and potentially regulate air leakage and blower energy use in building energy codes.

Purpose

The purpose of this study is to provide the enabling test methods and standards to allow the control of energy loss from air leakage and energy use of blowers in California Building Energy Codes.

Project Objectives

Project objectives are to:

- determine typical air leakage of forced air system cabinetry;
- develop a new laboratory test for air leakage of forced air system cabinetry; and
- develop a new laboratory test to evaluate the energy performance of air handlers;.

An important goal of this effort is to identify changes to existing residential building energy standards that can be incorporated into the 2008 and 2011 California Building Energy Efficiency and Appliance Standards to improve the performance of thermal distribution systems and reduce the energy use of air handlers.

Project Outcomes

Typical air leakage for forced air system cabinets is about 60 cfm or 5% of total system air flow. A reasonable target for an air leakage limit is 1.5% of the nominal air flow at a pressure difference of 0.5 in. water (125 Pa). This limit should be lowered in the future as manufacturers become better at constructing tight cabinets. Note that this limit is essentially equivalent to the current Florida and California requirements of 2% of nominal air flow at 1.0 in. water (250 Pa)

A new standard laboratory test for measuring HVAC cabinet air leakage has been developed as ASHRAE Standard 193P "Method of Test for Determining the Air Leakage Rate of HVAC Equipment". This standard should be complete in early 2010. In addition, a simplified method has been provided to the Commission for use in Building Energy Standards until Standard 193 is publically available.

The laboratory testing of the energy performance of air handlers undertaken for this and previous studies has been used in the development of a new standard in conjunction with the Canadian General Standards Board's (CGSB): C823 "Performance Standard for air handlers in residential space conditioning systems". This standard should be complete in early 2010, at which time it should be used by reference in California Building Energy Standards – potentially as an alternative to field testing and measurement of blower performance.

Conclusions

Forced air system cabinets have significant leakage particularly in the context of low-leakage systems defined in California Energy Code as 6% of total system flow. The air leakage of forced air system cabinets can be a significant contributor to systems failure to meet this standard. Improved and certified low leakage cabinetry is clearly desirable for the state of California. This study has shown that test methods can be produced that can be used to specify reduced system leakage and blower power consumption.

Recommendations

It is recommended that the State of California require air leakage testing of forced air system cabinets (primarily furnace and air handlers) using ASHRAE Standard 193 with a maximum air leakage of 1.5% of maximum air flow for each unit. For air handler power use it is recommended that field verification be required (as is currently implemented in the Residential ACM) for blower performance due to the dependence on duct system air flow resistance. Once the Canadian standard for blower performance is completed it could be referred to by California Building Energy Codes with appropriate performance specifications such as a minimum of 3 cfm/W. Builders and retrofitters of very energy efficient homes have reported that there is a difficulty meeting the tight duct specification in Title 24 for very small systems and is seen as a barrier for installing correctly sized systems. Specifying a maximum allowable air leakage for air handlers will improve this situation. In addition, it is recommended that the tight duct specification be altered so as to have fixed lower limit: e.g., 4% of total air flow or 20 cfm (10 L/s) – whichever is greater.

Benefits to California

Lowering cabinet air leakage from 5% to 1.5% of system air flow will result in a reduction in the energy used for space conditioning by about 5%. The same reduction, or more, is also applied to peak heating and cooling loads. Another benefit is that tighter cabinets will allow the smaller HVAC systems installed in energy efficient homes to meet tightness specifications.

The cabinet air leakage reductions will mean that more homes are capable of achieving the California State Building Energy Code requirements for a tight duct system. Currently, it is reported by implementers that they can install tight duct systems – but the residual leakage of the furnace or air handler is so great that they cannot achieve the tightness levels specified for the tight duct credit and this is a barrier to duct sealing because the credit is not obtained. Therefore tighter cabinets will encourage more users to pursue duct sealing and the tight duct credit.

Providing uniform ratings and test methods for blower power will allow designers to specify and installers to select equipment that uses less blower power. Currently this information is not available. Annual energy savings for a typical three-and-a-half ton air conditioner with typical California ducts are 45 kWh. Peak demand reductions are 50 W per system.

Both the air leakage and blower power reductions contribute to the California Long-Term Energy Efficiency Strategic Plan goals of net zero energy new homes by 2020, as well as the shorter term goals of 35% reductions relative to current state building energy codes. The approximately 5% reduction in heating and cooling energy use contributes significantly to these goals.

1.0 Introduction

Background and Overview

This project consists of two primary tasks. The first task develops test procedures and performance levels for the leakage of forced air system cabinets. The second task evaluates the effects of air handler energy use and develops a new standard test method for laboratory testing for power consumption of furnace blowers. This work will contribute to potential changes in 2008 and 2011 California Building and Appliance Standards. Information relating to each of these tasks is provided below, followed by an explanation of how these tasks relate to the overall PIER Goals.

The commonest metric for estimating residential forced air system leakage is an air flow at 25 Pa (0.1 in. water) because this metric is often used in field testing to meet the requirements of energy efficiency programs. Example methods for test can be found in ASTM E1554 (ASTM 2007), ASHRAE 152 (ASHRAE 2004) and quality control programs (such as those used by the US DOE's Building America teams) and numerous utility programs. California Building Standards currently use a variation of the test found in ASHRAE 152 and ASTM E1554 in section RA 3.1.4.3 of the Joint Appendixes (CEC (2008b)).

Concern about residential HVAC component leakage has been an issue for more than 10 years. Walker et al. (1998) measured six furnaces in a field study and found an average of 23 cfm at 25 Pa (11 L/s) or 34 cfm at operating pressures measured in the supply and return plenums. This study also showed a large system-to-system variation with a standard deviation of 17 cfm at 25 Pa and a range of 6 to 51 cfm at 25 Pa. The furnace leakage represented 24 to 76% of total system air leakage.

More recently, other researchers have performed field testing of residential HVAC equipment air leakage. The Florida Solar Energy Center (FSEC) presented data at ASHRAE Standard 193 meetings from a study on 69 Florida houses (Cummings et al. 2003). The air leakage from air handler/furnace cabinets was a significant fraction of total leakage from the tested systems: an average of 70 cfm at estimated operating pressure. In addition, Building Science Corporation (BSC) provided to the ASHRAE Standard 193 committee an overview of a study they conducted in which they worked with large home builders to reduce forced air system air leakage. The air handlers had a leakage of 45 to 60 cfm at 25 Pa about half of which was the furnace and the remainder in coil boxes and plenums. Converting to a higher static pressure of 125 Pa (half an inch of water) the average furnace leakage was 62 cfm. Smoke tests were used by BSC to determine leak locations. In the air handlers, the most significant leaks occurred at the partition between the blower compartment and the wiring compartment (especially corners), and around the blower compartment door. BSC stated that: "These problems should be correctable with gaskets and/or improved design.".

Currently, the State of Florida is the only entity attempting to legislate on the issue of furnace leakage. Florida Building Code section 610.2.A.2.1 (2003) specifies a pressurization test with the metric that the furnace cannot leak more than 2% of blower air flow at 250 Pa (1 in. water)[1]. However, no specific test procedure is provided and there is no consistency between ratings of different pieces of equipment.

[1] For example a furnace or air handler with a nominal air flow of 1000 cfm must have less than 20 cfm of air leakage.

Because the test procedures are not standardized there is concern from both regulators and equipment manufacturers about having a consistent basis for comparison or a "level playing field". The Florida requirements apply to furnaces only. In the new California Building Energy code (CEC (2008a)), there is a credit for having a tight air handler defined in Section 3.12.5 as a furnace with less than 2% air leakage at 250 Pa (1 in. water). This is the same as the Florida requirements and also does not specify how to do the test to ensure consistency of testing between different pieces of equipment.

Some furnaces come with certification that they meet air leakage specifications. An example from York states "Airflow leakage less than 1% of total airflow at ductblaster conditions. ". It is unclear what "ductblaster conditions" means. Another example from Goodman product literature states: "Factory sealed to achieve 2% or less leakage rate…..", however it does specify a test pressure of 1" water, but not specific test procedures. This lack of clarity and uniformity of testing is one of the reasons for undertaking the current study that aims to have a consistent test method for the industry to use.

The desire to have a consistent nation-wide test method led to the formation of a committee by The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE to develop a method of test of air leakage of forced air system cabinetry: SPC 193 Method of test for Determining the Air Leakage Rate of HVAC Equipment. This ASHRAE standard will apply to furnaces, air handlers, variable air volume boxes, etc. The first author of this report is the vice-chair of this committee and is the key technical contributor and editor as part of this project. The most recent draft of the ASHRAE Standard is included in Appendix A.

The intent of the ASHRAE method of test is to have a single test method for manufacturers (rather than have different test methods in different states) which can be referenced by state building codes for states, such as California and Florida, other standards, such as ASHRAE 62.2 and national/international building codes such as the IRC.

Information on test methods and procedures resulting from this study are being used by the ASHRAE Standard 193 committee to help develop the standard. Preliminary discussions with the ASHRAE Standard 193 committee indicated that there are several possible testing options that are evaluated in this project:

- tracer gas dilution
- pressurization
- depressurization
- splitting equipment that contains a blower into two parts at the blower to depressurize the parts of the equipment that normally operate under negative pressure and pressurize the parts of the equipment that normally operate under positive pressure.

The options for different test conditions that require evaluation include those that:

- reflect other test methods and are at a low static pressure like that used in Annual Fuel Utilization Efficiency (AFUE) and Season Energy Efficiency Rating (SEER) rating (typically 35 Pa to 60 Pa) resulting in underestimates of leakage.

- use static pressure values typically seen in new construction of 200 Pa (0.8 in. water) (as observed in new California homes and presented by Wilcox et al. (2006)) – with a split of half this pressure difference on supply side and half on return side – or +/- 100 Pa (0.4 in. water).
- use a static pressure difference that many manufacturers recommend as a maximum for residential furnaces - such as 125 Pa (0.5 in. water).
- use the operating static pressure number in the manufacturer's literature for a particular furnace.

The air leakage of HVAC equipment has two major issues of interest to the Commission:

1. Energy waste from air leakage.
2. Indoor Air Quality (IAQ) issues due to drawing air from crawlspaces, basements and garages.

It would be useful for the California Building Energy Code to specify tighter forced air system equipment to supplement the existing duct tightness requirements. It would also allow IAQ issues to be addressed in a simple fashion by requiring tight equipment as well as ducting in places outside conditioned space. In addition, the Energy Commission would be able to introduce its own regulations.

The State of California will benefit greatly from having a test standard in place either as an appliance standard or as part of the Building Energy Code requirements. Once the ASHRAE standard is completed then the Energy Commission could refer to the ASHRAE standard in California Building codes.

The electricity use of furnaces (and forced air equipment in general) has been receiving increasing attention in recent years. LBNL has been at the forefront of this issue and has performed studies for the Energy Commission and Pacific Gas and Electric (PG&E) as well as providing technical support for the American Council for an Energy Efficient Economy (ACEEE), Environmental Protection Agency (EPA) (EnergyStar), the Consortium for Energy Efficiency (CEE), Building America and ASHRAE. At a recent CEE conference call to address EnergyStar issues for furnaces all the participants, including regulators and utilities, agreed that a standard test procedure for electricity use of furnaces and other HVAC blowers is an urgent requirement. In Canada a new standard (CSA 823 Performance Standard for Air handlers in Residential Space Conditioning Systems) is being drafted as a laboratory test of blower power use. We are participating in the development of this Canadian standard[2] that could either be directly adopted (or referred to) or adapted by the Energy Commission for use in California Building Standards.

The key issue addressed by this work is that current test procedures for furnaces (AFUE (ASHRAE Standard 103 (ASHRAE 2007))) and air conditioners (SEER (ARI (2008))) do not test at the pressures found in real duct systems and therefore cannot be used as a measure of equipment performance. This leads to the following problems:

- California would like to require low electricity use forced air system blowers or at least be able to give credit for their use – but cannot use current product listings.

[2] Iain Walker is a member of the technical standards committee developing the standard.

- Contractors would like to be able to know if a particular blower would be likely to meet the proposed 2008 Title 24 credits for a good blower – but this information is often not available.
- EnergyStar would like to include electricity use in the EnergyStar Furnace specification using the descriptors in the Gas Appliance Manufacturers Association (GAMA) directory[3]. However, based on input from LBNL and others regarding the utility of the GAMA ratings, EnergyStar is reconsidering this strategy.

The energy consumption of central forced air system blowers has been examined in several studies nationwide and in Canada in both new and existing construction. The consensus is that most blowers move about 2 cfm (1 L/s) per Watt of fan power. The fan power depends on the size of the equipment and the air flow resistance of the duct system. Currently California Building Energy code gives a credit for improved blower performance and there is a desire to set a minimum performance standard for all systems. To achieve this requires reliable information for system designers and installers – such as that obtained from standard laboratory testing. To this end a collaborative effort is under way with the Canadian Standards Association to have a laboratory test method that evaluates equipment at the operating pressure differences found in field studies (0.5 in water (125 Pa) for heating operation and 0.8 in. water (200 Pa) for cooling operation) together with provisions for multi-speed equipment and combining heating and cooling performance.

The California Building Energy code focuses on cost-effective ways to minimize the energy-related impacts of providing building services such as thermal conditioning, lighting and water heating. Providing airtight HVAC systems is a simple effort to further this strategy. The energy performance of air handlers used for ventilation and distribution systems is a critical part of the energy impact of ventilation and standards compliance. As duct systems become tighter, the air leakage of the cabinets is becoming more critical and reduction in this leakage will insure greater compliance with minimum forced air system air leakage levels required in the California Energy Efficiency Standards.

Project Objectives

Project objectives are to:

- determine typical air leakage of forced air system cabinetry;
- develop a new laboratory test for air leakage of forced air system cabinetry; and
- develop a new laboratory test to evaluate the energy performance of air handlers;.

An important goal of this effort is to identify changes to existing residential building energy standards that can be incorporated into the 2008 and 2011 California Building Energy Efficiency and Appliance Standards to improve the performance of thermal distribution systems and reduce the energy use of air handlers.

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[3] The GAMA directory has an "e" designation for electrically efficient furnaces whose electricity consumption is less than 2% of total energy use.

2.0 Project Approach

This project has two parts. The first part addresses the air leakage of residential HVAC equipment and the second part addresses the measurement of residential furnace and air handler power consumption.

2.1. Air Leakage of Residential HVAC Equipment

The objective in this part of the project is to measure HVAC system component air leakage and to develop standard measurement techniques for laboratory evaluation of residential HVAC system component air leakage.

Concurrently, an effort is underway at ASHRAE to develop a standardized method of test for determining the air leakage of HVAC equipment. This test method is intended to apply to furnaces and air handlers as well as other HVAC components such as Variable Air Volume (VAV) boxes, tees and wyes, etc. During initial meetings to discuss the scope and nature of the test method, a concern was raised over the application of a simple pressurization test (as required by the State of Florida). For equipment with blowers, there are positive and negative pressure zones within the equipment. Negative pressure differences between inside and outside the equipment occur upstream of the blower and positive pressure differences downstream of the blower. It has been stated by some equipment manufacturers that the cabinet construction is such that the efficacy of air sealing is determined by the direction of the pressure difference. In particular, blower access panels tend to be on the negative pressure side of the blower and will be sucked onto the blower compartment when the blower operates. If a simple pressurization test is used to evaluate this equipment, the access panels will tend to be forced away from the blower compartment and this may lead to increased air leakage. This implies that a test is needed that has negative pressures upstream of the blower and positive pressures downstream.

This was seen to be a valid concern by the members of the ASHRAE Standard committee and the Energy Commission. Therefore this project looked at alternative methods to simple pressurization that attempt to maintain the same pressure direction as during normal operation for devices that have both positive and negative pressures. Typically, these are devices containing fans or blowers such as furnaces, air handlers and VAV boxes. This multi-pressure regime testing can be achieved a few ways:

1. An elegant and appealing approach is to use the blower itself to produce the test pressures. For this option an adaptation of the DeltaQ duct leakage test method (see ASTM E1554) was developed.
2. Install an air tight seal at the blower. This can be done one of two was: either remove the blower and place an air tight seal in place of the blower or seal the blower inlets or outlets. The upstream and downstream zones of the equipment could then be evaluated independently – with negative pressures used upstream and positive pressures used downstream.
3. Connect a blower to each side of the equipment and ensure that both sides are equally pressurized by nulling the pressure difference between the two sides. This is done for both

positive and negative pressures. The test results are taken from the positive side during pressurization and normally negative side during depressurization.

4. Using tracer gas dilution techniques to estimate air flows. A tracer gas is injected into the system and concentrations measured at various locations (together with system air flows) to estimate the flows into and out of the equipment.

This project evaluated the different test methods in the LBNL duct leakage test facility. Several furnaces, VAV boxes, some HVAC system components, and a single rooftop package unit were evaluated using combinations of the test methods outlined above. Additional tests were performed on two more furnaces at the test laboratories of IBACOS in Pittsburgh, PA. IBACOS shares an interest in this work because they want to install non-leaky systems in the homes they design and build. They are also a US Department of Energy Building America team and are interested in applying the information learned in this project to homes developed for the Building America program. Some tests were repeated multiple times to estimate the repeatability.

2.1.1. LBNL Furnace and HVAC Component Testing

The LBNL duct test laboratory is used for the Furnace and HVAC component testing at LBNL. Three test procedures are evaluated: Pressurization, DeltaQ and Tracer Gas.

Pressurization

Similar to laboratory testing of building envelope components (ASTM E283-04 for windows), whole house envelope leakage testing (ASTM E779-03) and duct leakage testing (ASTM E1554-07) the equipment under test has air blown in or out of it at a specified pressure and the required air flow to maintain this pressure is recorded. The test pressure difference was selected to be 125 Pa (0.5 in. water) based on a literature survey (see a summary in Walker (2008)) of field measurements of residential system static pressures.

For pressurization testing a combination fan/airflow meter was used to pressurize (and depressurize) the equipment under test. The fan/airflow meter was connected by a section of 10 in. flexible duct to the equipment under test. For very low leakage values (below 5 L/s [10 cfm]) a different fan with a separate low-flow nozzle system was used.

Although extensive measures were taken to seal the test apparatus, there was still some air leakage. Because this air leakage would be included in the measurements of equipment leakage, a background leakage test was performed so that this could be subtracted from the total. The resulting air flows would be for the equipment only. The outlet of the measurement equipment was sealed and the test apparatus was pressurized to 125 Pa using the fan/flow meter. The background leakage test was repeated with the test apparatus depressurized.

Three pressurization techniques were explored:

1. Whole system. This approach was applied to all the equipment tested. The whole piece of equipment is subjected to positive or negative pressures in two separate tests.

2. Positive/negative split. For equipment with a blower that has a zone of positive pressure and a zone of negative pressure a seal is placed between the positive and negative zones. The positive pressure zone is tested under positive pressure only and the negative pressure zone

under negative pressure only. This allows for separating the leaks in and out of the equipment under normal operating conditions. In practice, the seal was only effective when the equipment blower was removed because there is no access to the blower outlet in the equipment tested and the blower inlets had protrusions for the motor mounts that precluded any simple sealing techniques (Figure 1).

3. Two fan split. In this test two fans were used – one connected to the positive zone and one connected to the negative zone. The fans were adjusted to have the same pressure in the two zones with no pressure difference between the zones. The test is performed twice – first for pressurization – in which the air flow into the positive pressure zone is the positive pressure zone leakage; and secondly under depressurization where the air flow from the negative pressure zone is the negative pressure zone leakage.

Figure 1. Electric motor protruding from blower inlet that prevents easy air sealing

DeltaQ

The DeltaQ test uses the same principles as the field test for duct leakage of the same name in ASTM Standard E1554-07. It attempts to test under normal operating conditions with positive pressure on the supply side of the equipment and negative pressure on the inlet side. The pressures and air flows are created by the built-in blower. The pressures are controlled by exterior dampers. It only applies to equipment containing a blower that would maintain these positive and negative pressures under normal operation. Air leakage in and out of the negative and positive pressure zones of the equipment is measured simultaneously. The DeltaQ test uses measured air flows with the equipment blower on and off over a range of equipment cabinet pressure differences created by an external fan to determine leakage at normal operating conditions.

Figure 2 shows the experimental apparatus for the DeltaQ tests. The apparatus uses a test chamber to represent the "house". The test chamber dimensions are 9 m x 2.5 m x 2.5 m (30 ft x 8 ft x 8 ft). The test chamber is constructed from a wood frame sheathed in plywood with extensive air sealing (leakage was introduced so that the chamber air leakage was 125 L/s (250 cfm) at 25 pa). An external fan/flowmeter is used to operate the test chamber over a range of envelope pressures (±100 Pa (0.4 in. water)). This range of pressure is used to ensure that data are obtained near the operating pressures of (±125 Pa (0.5 in. water)). The furnace was connected to a test apparatus/test chamber ducting using flexible ducting (Figure 3). Dampers at the furnace inlet and outlet were used together with an in-line fan to control the furnace static pressures to be +125 Pa at the furnace exit and, and -125Pa in the blower compartment.

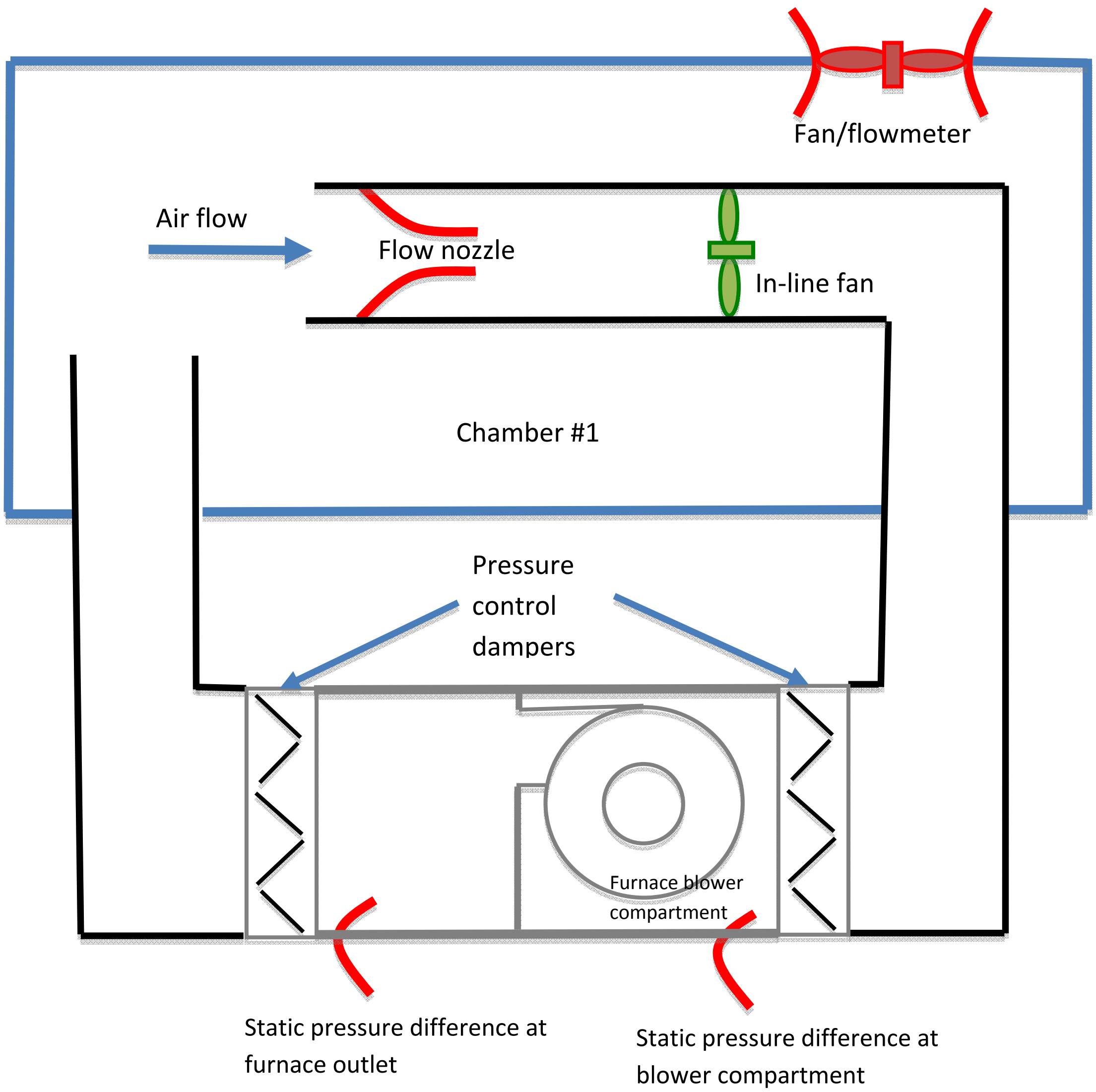

Figure 2. DeltaQ test apparatus

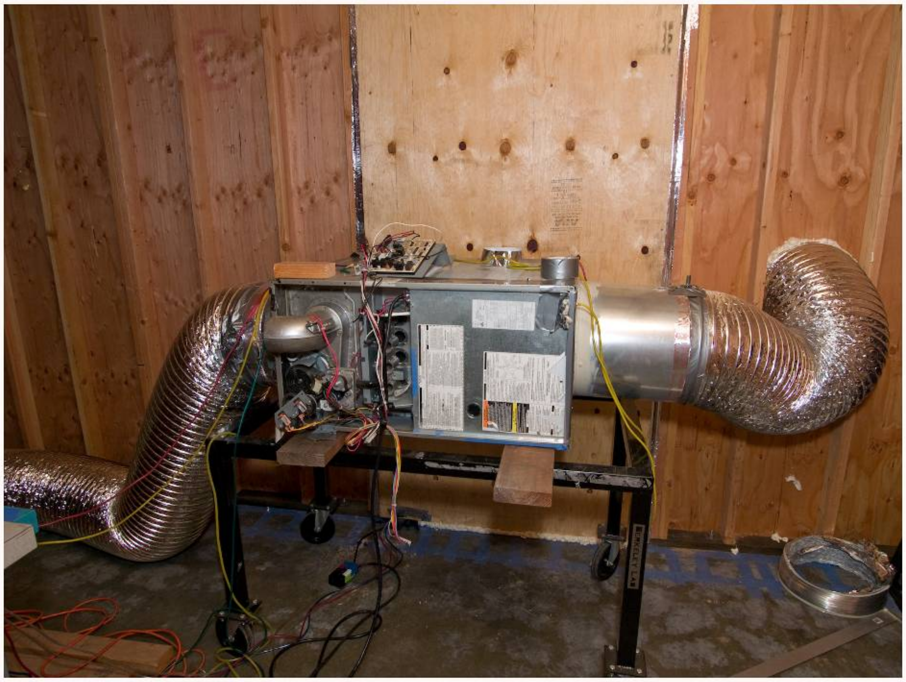

Figure 3. Furnace connected to test chamber with flexible ducting

An automated data acquisition system is used to measure the system static pressures, the total system air flow, chamber envelope static pressure difference and the flow through the chamber mounted fan/flowmeter. The ramping DeltaQ technique (Walker and Dickerhoff (2007)) is used such that the pressure in the test chamber is slowly increased over a time period of about two minutes. Pressure and air flow data are recorded approximately two times per second during the ramping process. The DeltaQ approach uses the differences between the air flow required to pressurize the test chamber to a given static pressure difference with the furnace blower on and off. The furnace blower operation changes the pressures across furnace leaks such that , for a given test chamber pressure difference, the air flows required to maintain this pressure difference change when the furnace blower is turned on. The DeltaQ test has four parts:

1. Furnace blower off, pressurization. The fan/flowmeter pressurized the test chamber and the furnace, slowly increasing pressures and flows over about 90 seconds from zero to about 125 Pa.

2. Furnace blower on, pressurization. Step one is repeated with the furnace blower on.

3. Furnace blower on, depressurization. The fan/flowmeter depressurized the test chamber and the furnace, with the furnace blower on.

4. Furnace blower off, depressurization. The final step repeats step 3 with the furnace blower off.

The differences between furnace blower on and off data are used in a mathematical air flow model of the test chamber and duct system (developed by Walker et al. (2001) and outlined in ASTM E1554) to estimate the flow in and out of the furnace. The dampers and in-line fan controlled the apparatus static pressures to be +125Pa +0.5 in. water) at the outlet of the furnace and -125 Pa (-0.5 in. water) in the blower compartment to match the system static pressures measured in field data.

Tracer Gas

Like DeltaQ, the tracer gas test attempts to measure the leakage under normal operating conditions. The equipment is connected to a duct system and tracer gas is injected into the duct/equipment system. Combined with measurements of air flows at critical parts of the duct/equipment system to allow the calculation of air flow into and out of the equipment under test.

The tracer gas tests used the same apparatus as for DeltaQ but with the furnace located in a second test chamber. The duct system is disconnected from the in-line flow nozzle used to measure the total return minus return leakage air flow and a tracer gas (CO_2) is injected at this point. The second test chamber is ventilated at a known rate using a fan/flowmeter. Tracer gas concentrations are measured at the entry to the furnace (C_r), at the exit of the ducting (C_s), in the furnace chamber (C_c) and outside the test chambers (C_{amb}). The total furnace blower flow (Q_{meter}) is measured with an in-line nozzle flow meter. The tracer gas test apparatus is shown in Figure 4. A pump is used to draw continuously from these locations. Every two to four minutes a sample is taken to research grade infrared analyzer that provides the concentration of tracer gas in the sample.

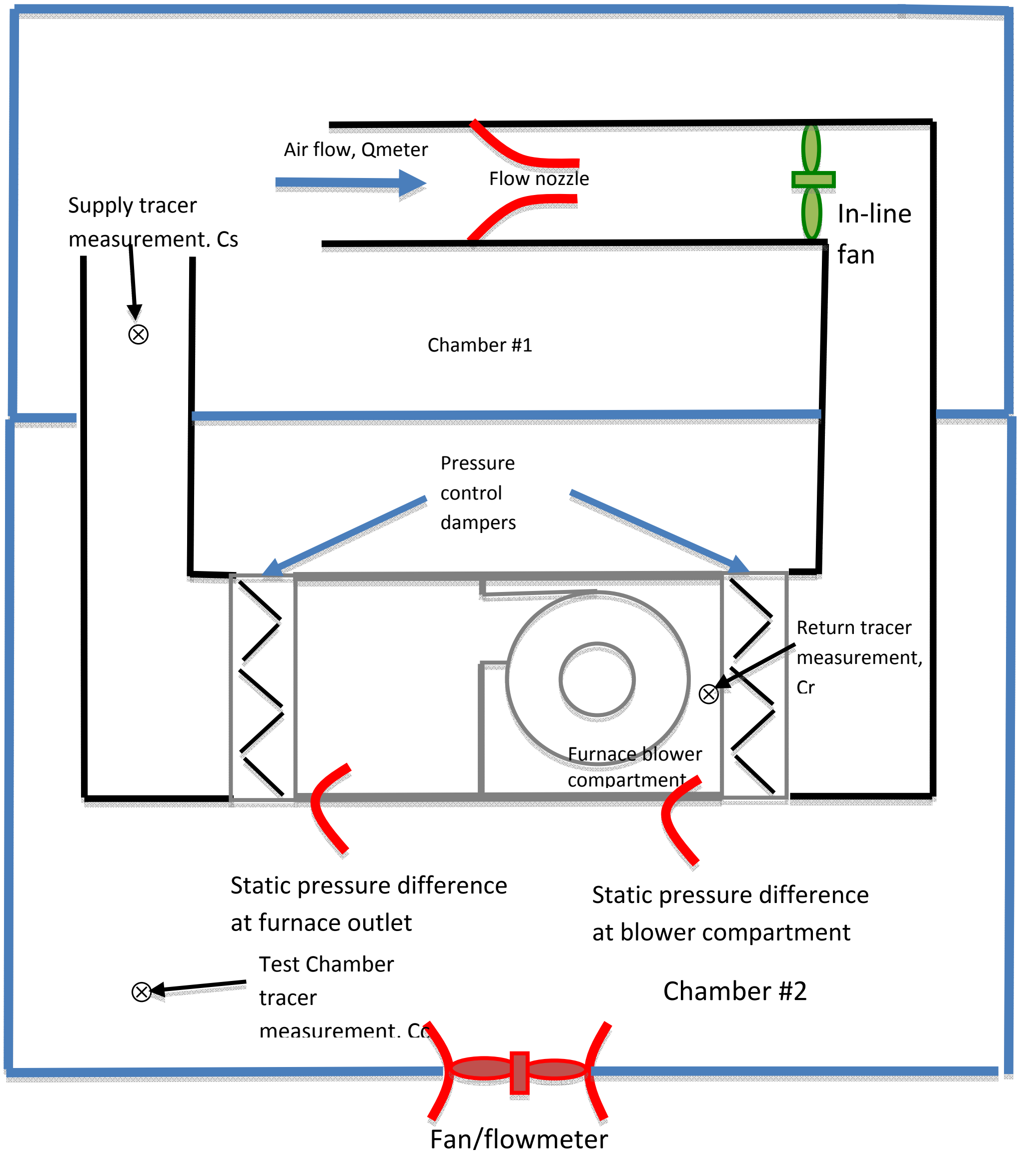

Figure 4: Tracer gas test apparatus.

A mass balance is used to determine the air flows in (Qr) and out of (Qs) the furnace. The concentration of the tracer gas in the supply duct is

$$C_s = \frac{C_r Q_{meter} + C_c Q_r}{Q_{meter} + Q_r}$$

The air flow into the negative pressure zone of the furnace is given by:

$$Q_r = Q_{meter} \left(\frac{C_r - C_s}{C_s - C_c} \right)$$

The concentration of the tracer gas in the chamber is

$$C_c = \frac{C_{amb} Q_{chamber} + C_s Q_s}{Q_{chamber} + Q_s}$$

$$Q_s = Q_{chamber} \left(\frac{C_c - C_{amb}}{C_s - C_c} \right)$$

The tracer gas testing presented several challenges that make it difficult to recommend as a standardized test method. The first challenge is that it is difficult to ensure complete mixing of the supply leakage flow (Qs) with the chamber flow before this air stream was drawn back into the furnace by the return leakage. This potential short-circuiting results in a higher Cs and lower Cc than the perfect mixing assumed in the mass balance calculations. Several arrangements of small fans located near the furnace are used to evaluate this mixing issue. After several arrangements of fans were tried, the final arrangement requires strong (several hundred L/s (cfm)) air flow very near the furnace. The other mixing issues include mixing of tracer gas after injection into the return ducting. The presence of several duct elbows and an inline fan in the duct system and other mixing fans in the room contributed to this mixing. In developing this test, the sample locations were moved around outside the furnace to examine spatial variations that would indicate incomplete mixing. Analysis of the samples indicated that the concentration was uniform to within 10%.

The second challenge in using the tracer gas method was that the analysis is only valid at, or very near to, steady-state conditions, i.e., when the tracer gas concentrations no longer change with time. In the furnace leakage experiments the leakage flows are small compared to the volume of the test chambers so it took several hours for steady-state conditions to be reached. To obtain an initial estimate of the required testing time, a CONTAM model of the test set up was used to investigate the potential errors associated with measurements made before reaching steady state. The CONTAM model results indicate that errors in predicted air leakage flows would be less than 2% after about four to five hours (depending on the sample location). In the analyses performed in this study multiple cycles were averaged to obtain the tracer gas concentrations at each sample location. Figure 5 shows the CONTAM estimates of uncertainties and how they are reduced at longer experimental times. These CONTAM estimates of the time required to reach equilibrium concentrations were confirmed in the laboratory tracer gas testing. These long time periods to reach steady-state conditions meant that each furnace test took too much time to be considered reasonable as a standardized technique and also reduced the number of tests performed for this study.

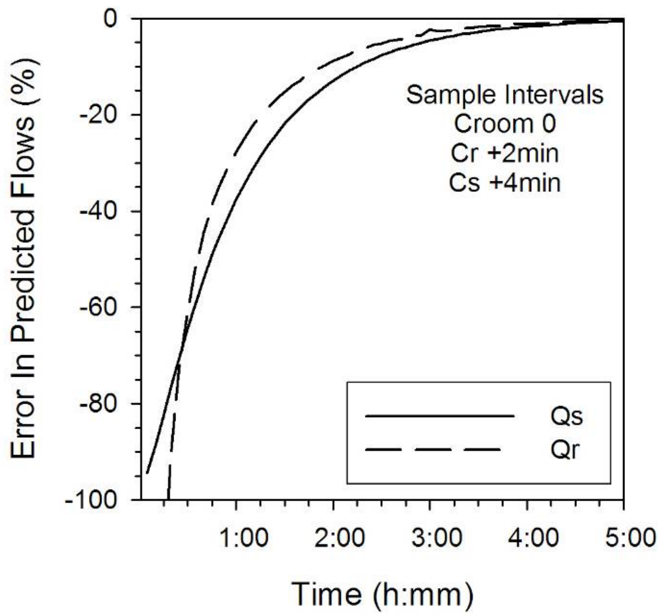

Figure 5. CONTAM estimates of errors in leakage flows decreasing as steady-state conditions are reached after about four to five hours.

Key observations from the laboratory testing are that the method is sensitive to short circuiting issues that would need adjustments for each furnace tested. In some cases it might be impossible to fully prevent the short circuiting without changing the leakage flows. Another issue is with the mixing fan inside the duct system that mixes the tracer gas. Too weak a fan will result in insufficient mixing and too strong a fan may change the air flow pattern such that the equipment leakage is changed. This would require extensive additional experiments and measurements inside the duct that are beyond the scope of this study.

2.1.2. Small Chamber Furnace Testing

Two furnaces (one Bryant and one Lennox) were tested using the pressurization and DeltaQ techniques at the test laboratories of IBACOS in Pittsburgh. These furnaces had been removed from houses and were a couple of years old. Both furnaces are configured with side entry and top supply. The Bryant (80kBu rated) furnace has a six inch extension mounted on the bottom of the furnace to allow for extra clearance around the blower. This sheet metal extension was added by the contractor who installed the furnace in the house from which it was removed. The contractor used this addition to ensure good air flow through the system because he considered the standard cabinet to be too restrictive. Clearance between the bottom of the furnace cabinet and the blower housing is approximately one inch, or less, without the blower compartment extension. Previous studies (e.g., Walker 2008) have confirmed that clearances this small can significantly reduce blower performance. The access panels on the Bryant are in two parts. An inner panel secured with two sheet metal screws sealed the blower compartment. This panel did not have any interlinked tabs at the edges – just flat sheet metal to sheet metal contact. The main front panel is held in place with two snap fasteners at the top of the cabinet.

The second furnace is a Lennox of a similar capacity. It has a friction fit blower compartment door, with a second friction fit door for the burner compartment. Both of these doors have mechanical closing clamps. The bottom of the furnace is a separate panel with no fasteners or seals. This panel is easily displaced during moving of the furnace and is carefully replaced before the air leakage testing No attempts were made to seal this furnace bottom, other than ensuring a good panel fit.

Two tests are performed on each furnace. The first test pressurizes and depressurizes the furnace to 125 Pa (0.5 in. water). The second test uses an approach based on the DeltaQ method developed for field testing of HVAC system leakage. Air flows are generated and measured using a combined fan/flowmeter for both tests. For the DeltaQ test a prototype software package developed in conjunction with the Energy Conservatory is used to control the duct blaster, record pressures and flows and perform the analyses.

A test apparatus was constructed to include a small test chamber that was connected to the furnaces using 350 mm (14 in.) diameter uninsulated flexible duct. Panels made from foam board insulation allow the mounting of starter collars to the furnace inlet and outlet. The foam board panels are sealed to the furnace inlet and outlet using duct tape. The fan/airflow meter is connected by a section of 250 mm (10 in.) diameter flex duct to the test chamber, as shown in Figures 10-13. Round manual dampers are connected to the starter collars so that the supply and return pressures could be controlled. This is shown schematically in Figure 6. Figure 7 shows the furnace connections and pressure probe locations.

System pressures are measured in three locations. The test chamber pressure is measured in the corner of the test chamber to avoid pressure fluctuations due to the large airflows through the confined spaces of the small test chamber. The return and supply pressures are measured using static pressure probes inserted through the mounting panel.

Deliberate holes cut into the test chamber for the DeltaQ testing allow for imbalanced air leakage flows in and out of the test apparatus. These holes are sealed for pressurization testing. Figure 5 shows one of the holes on the side of the test chamber.

When testing the Bryant furnace with the test chamber in an elevated position movement of the flex duct during testing may have resulted in changes in the pressures and air flows due to changes in air flow pattern and duct air flow resistance. Therefore, for the Lennox furnace the test chamber was placed on the ground and the flex duct was taped to the floor to limit movement during the testing.

The dampers are used during the DeltaQ testing to control the supply and return plenum pressures. The target plenum pressures are 125 Pa for supply and -125 Pa for the return relative to outside the furnace. These pressures are chosen to match those found in field tests of furnaces and also used in the pressurization testing. As shown in the results section below, this proved to be a very difficult task and these target pressures were not achieved. One reason for this is the cross talk between the two damper adjustments. As each damper is closed the air flow through the system is reduced – leading to reductions in pressure in the plenum whose damper was not being adjusted. As the dampers are progressively closed in an iterative manner, eventually the flow is choked to the point where the blower fan stalls. This is the second reason for not achieving the target pressures: some blowers are not rated at a full inch (250 Pa) of static pressure difference and do not work at these high static pressure differences. Therefore, the damper settings were limited to achieve a reasonable pressure difference without reaching the point of blower stall.

Figure 6. Schematic of small test chamber DeltaQ furnace cabinet leakage apparatus

Figure 7. Flex duct and damper mounting on Bryant furnace

Pressurization test procedure

The pressurization procedure is the same as the whole system approach used in the LBNL laboratory study where the whole furnace is pressurized at the same time. In order to isolate the leakage of the furnace from the leakage of the test apparatus, a background leakage test is performed with no furnace attached to the test chamber. The outlet of the foam board panels is sealed and the test apparatus pressurized to 125 Pa using the fan/flow meter. The corresponding air flow is the pressurization background leakage. The test is repeated with the test apparatus depressurized. The background leakage is subtracted from the test results for each furnace – with pressurization background leakage subtracted from pressurization results and depressurization background leakage subtracted from depressurization test results.

The fan/airflow meter is used to pressurize (and depressurize) the test chamber, the attached ducts and the furnace. The test chamber, ducts and furnace are pressurized to 125 Pa and the corresponding air flow is recorded. The test is repeated with the duct blaster depressurizing to -125 Pa. For all the pressurization testing the pressures in the test chamber, supply and return and for the air flow are ten second time averages.

DeltaQ test procedure

The DeltaQ test procedure followed the same steps as used in the LBNL laboratory testing with the test chamber used to simulate the "house". The furnace is outside the "house" therefore its leakage is leakage to outside and will be measured by the DeltaQ test. Automated software provided by The Energy Conservatory was used to adjust the fan/flowmeter and to record the pressure difference between in the inside and outside of the test chamber and the air flows used to pressurize (and depressurize) the test chamber. The software fit the DeltaQ equations to the difference between furnace blower on and off data to determine the air flow in and out of the furnace.

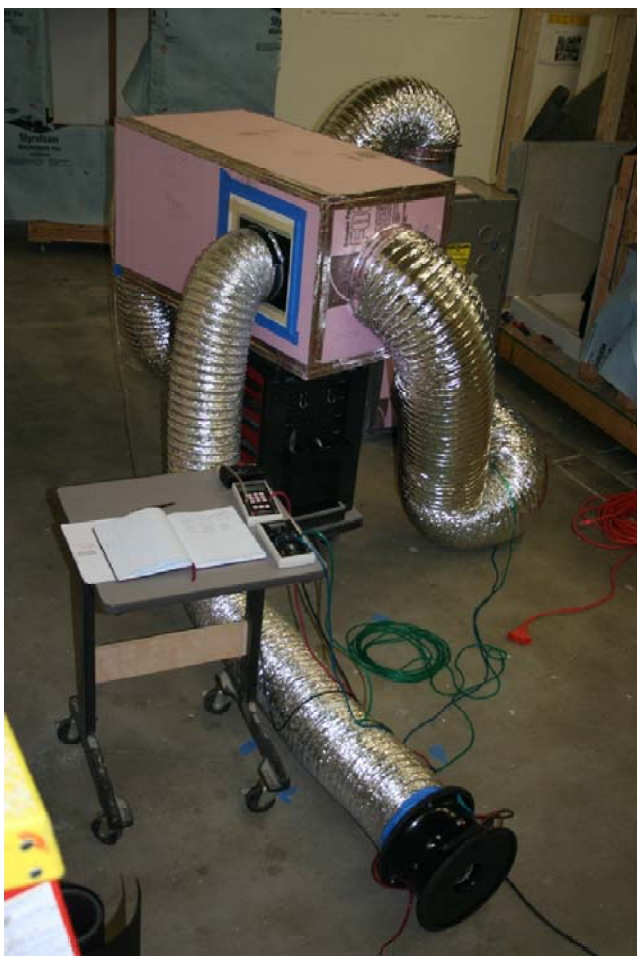

Figure 8. Test chamber connections to furnace and integrated fan/flowmeter

Figure 9. Flex duct connections to test chamber and Bryant furnace

Figure 10. DeltaQ Testing of Lennox furnace

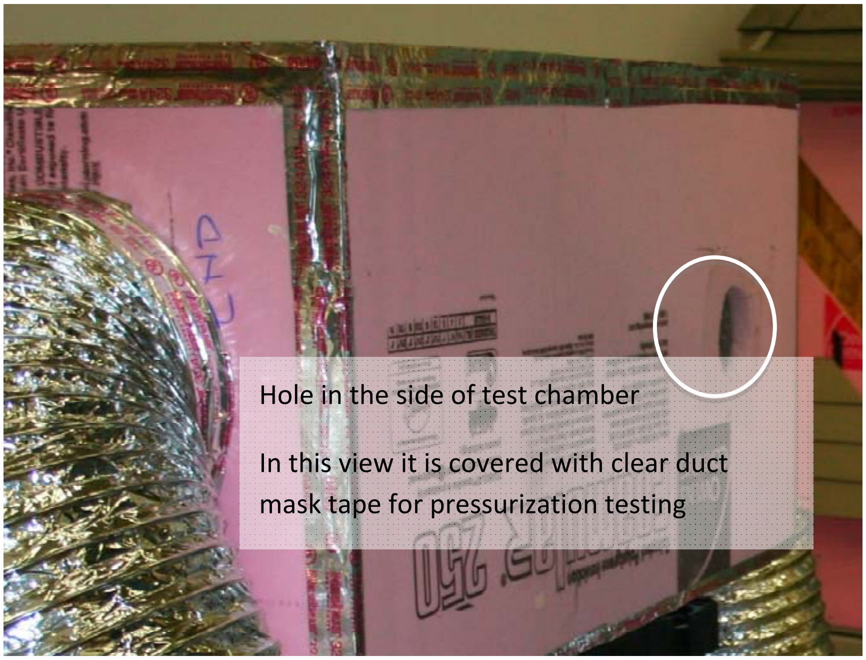

Figure 11. Side view of test chamber with one hole

Energy Savings Estimates

Energy savings associated with reductions in cabinet leakage from 5% to 1.5% of system air flow were made using ASHRAE Standard 152 (ASHRAE 2004). The calculations used the climate data for Sacramento, CA. for a 2000 ft^2 house, with a 90 kBtu/hr furnace and a three ton air conditioner, with R8 duct insulation. The HVAC system was located in the attic. Two cases were examined: one with balanced duct leakage and one with no supply leakage and just return leakage. In both cases the change from 5% to 1.5% leakage was made for return ducts only because the laboratory tests showed that the furnace leaks were predominantly on the return side. The calculations were performed for design conditions (that estimate peak demand reductions) and seasonal conditions (for energy savings estimates). The results are shown in Table 1.

Table 1. ASHRAE Standard 152 distribution system efficiency (Delivery Effectiveness) estimates

		Unbalanced		Balanced	
	Leakage (%)	**5**	**1.5**	**5**	**1.5**
Heating	**Design**	0.93	0.94	0.89	0.93
Heating	**Seasonal**	0.95	0.95	0.91	0.94
Cooling	**Design**	0.82	0.88	0.79	0.87
Cooling	**Seasonal**	0.89	0.92	0.85	0.91

The results in Table 1 show that the air leakage reduction leads to about 5% energy savings for cooling, with lower savings for heating. Peak demand savings are a couple of percent greater.

2.1.3. Development of test standards for air leakage and blower power consumption

Most of the effort for this project on the development of a standard test method for air leakage has focussed on ASHRAE Standard 193P "Method of Test for Determining Airtightness of HVAC Equipment". Efforts for this project include providing administration and guidance through Iain Walker's participation as the vice-chair of the committee as well as using the results of this project to guide test procedure development. The measured laboratory data has been very informative for the standard committee in the evaluation of potential measurement techniques and without this PIER project the ASHRAE standard would either not exist or would be many years from publication. The ASHRAE standard will be an ANSI approved consensus document prepared with input from equipment manufacturers, potential users and researchers. The Standard uses two experimental approaches that were evaluated in this study: pressurization of the whole piece of equipment and pressurization testing with separation of the negative and positively pressurized sections of equipment containing blowers. Appendix A is a draft of the standard. It is expected that this standard will be completed and published in late 2009/early 2010. Exact publication dates will depend on public review commentary and ASHRAE procedures.

There is currently a low-leakage air Handler credit in the 2008 Residential ACM Manual, Section 3.12.5 Duct/Air Handler Leakage that specifies the allowable air leakage (2% of nominal air flow) at a pressure difference of one inch of water (250 Pa). A test procedure was developed to determine the air leakage of air handlers for the purposes of the ACM. Based on the results of the testing done for this study, the test procedure depressurizes the air handler and measures the air flow to maintain a static pressure of one inch of water (250 Pa). The test procedure is attached as Appendix A.

The development of a test method for blower power consumption has focused on participation in the development of a new test method with the Canadian Standards Association: CSA 823 "Draft Performance Standard for Air Handlers in Residential Space Conditioning Systems". This test method also includes participation from a wide range of interested parties including equipment manufacturers, potential users, researchers and technical experts. This PIER project has allowed Iain Walker to participate in the development of this standard as a committee member and to contribute the knowledge gained in this PIER project to the development of the standard. This standard is currently in draft form and is expected to be completed and published in 2010.

The blower power consumption standard uses the concept of determining blower performance over a range of operating points. The results of field testing show that typical static pressure differences across furnaces are 125 Pa (0.5 in. water). The draft C823 test procedure requires that the pressure difference across the furnace is controlled by dampers to be this 125 Pa (0.5 in. water) with the blower at its heating operating speed. The damper settings act to create a duct system air flow resistance that is then used at all other operating points – such that higher air flows used for air conditioning or lower flow rates used for air mixing/filtration result in higher and lower static pressure differences respectively.

When ASHRAE Standard 193 and CSA 823 are complete and published they can be used by the Commission in relevant energy efficiency standards.

2.2. LBNL Power Consumption Testing and Test Procedure Development

The objective of this part of the project is to examine various techniques for measuring the power consumption of residential HVAC equipment forced air blowers and produce a draft experimental test procedure suitable for use in developing laboratory ratings. To achieve this objective several procedures and measurement devices were evaluated using a furnace in the LBNL duct laboratory. Recent studies by other researchers were used to establish some of the test parameters, to look at test repeatability and to provide insight for test method development.

To obtain credit in California Building Energy Code compliance for an energy efficient blower requires field measurements to confirm actual performance. An alternative approach would be to have equipment ratings that included blower power measurements so that installers, contractors and building designers would be able to specify low power consumption equipment and install it with confidence that it would meet a performance specification. This would reduce the effort required to obtain credit in Title 24, or reduce the effort to show compliance with future versions of Title 24 that may require minimum air handler performance rather than just having credit for improved performance. This study investigates several methods for obtaining data on blower power consumption. The objective is to produce recommendations on acceptable methods for power measurements that can be used in future versions of Title 24 and to provide draft language for standard measurement approaches. One key possible application is to add the measurement of blower power to the current AFUE rating procedure – including revisions to the static pressures used in the AFUE testing. This study focuses on gas furnaces that are the predominant forced air system blower equipment, but the same approaches apply to the air handlers of heat pumps and electric furnaces.

Residential furnace blower power consumption is in the range of 100 W to 1000 W using single phase power. Typical power consumption is about 500 W based on a wide range of field surveys (summarized in Walker (2008)). The power consumption depends on the selected blower speed, blower capacity, and system air flow resistance. There are two types of motors used in furnace blowers: Permanent Split Capacitor (PSC) and Brushless Permanent Magnet (BPM). PSC's are the most common blower motor (roughly 95% of the market (Sachs and Smith (2003)). BPM's are often referred to as variable speed blowers and are used in higher end premium equipment that often utilize their ability to operate over a wide speed range and gradually increase speed and air flow.

Three approaches were considered for estimating power consumption:

1. Using Correlations between rated furnace input capacity and blower power

A simple approach to determine furnace blower power would be to assume a correlation between blower power and furnace input rating. This would only require a reading of the furnace nameplate to determine the power consumption.

2. Using Manufacturer's data and measured static pressures

Some manufacturers publish data on air flow and power consumption based on static pressures. If these data are available, then a measurement of static pressure will allow both air flow and power consumption to be known without additional measurements. This allows for cheaper and quicker testing. This has the appeal of being a useful diagnostic for other system problems – e.g., overly high static pressure. Many contractors will have a pressure measuring tool already and they don't have to learn how to use another tool. They also understand the concept of static pressures for duct systems. It is also a simple test – not requiring additional equipment or training for electrical and air flow measurement.

3. Direct Measurement

Direct measurements of furnace blower power consumption are somewhat complicated by the electric power consumed by other parts of the furnace such as the operation of combustion air blowers, gas valves and ignitors. Typical power consumption for these ancillary loads are (from Lutz et al. (2006)):

- Controls: 5W (PSC) and 10 W (BPM)
- Combustion air blower: 75 W (reduced to 60 W for a two-stage furnace in low-fire mode)
- Ignitor: 400 W

This implies that care must be taken when inferring blower power consumption from total furnace power consumption. There are two paths to directly measure blower power:

Path 1. Isolate the blower from the other power consuming equipment in the furnace.

This is the more difficult and time consuming path, but provides the measurement of blower power only. To isolate the blower it is necessary to open the furnace and make electrical connections to the blower power wiring. An electric power meter is used that measures the current to the blower motor using a clamp-on current transformer (CT). For a meter with a remote CT, wires from the CT to the meter are fed out of the furnace through an existing knockout or a hole is drilled for this purpose. For direct clamp-on meters (with the CT built into the meter itself) the meter is left inside the blower cabinet during operation – and a meter with automatic datalogging capacity is used to record the power consumption. A voltage measurement is made either in the power supply to the furnace, or within the furnace itself.

Path 2: Measuring the total energy consumption of the furnace.

On this path the blower compartment is not opened to isolate the blower power. Instead the total power for the furnace is measured and the power for the ancillary devices is subtracted from this total to obtain the blower power.

The measurement of electrical power usage in a system with fixed or “hard wired” configuration may require the services of a licensed electrician. This will limit the ability of many practitioners to perform this test.

3.0 Results

3.1. LBNL Furnace and Component Leakage

Three furnaces and a rooftop package unit (RTU) were tested using the methods outlined in section 1.1.1 . The three furnaces had blower compartment doors that were friction fits with no gaskets or seals. The RTU had a blower compartment door with a foam rubber gasket.

For multiple tests performed for repeatability evaluation, the instrumentation was completely removed and the apparatus re-set to simulate the uncertainty as if the testing was done in different laboratories (with the same instrumentation). In Tables 1 through 4, the values following the ± sign represent the standard deviation of the multiple tests.

Figure 12 illustrates example DeltaQ data from Furnace 3. The first plot shows the individual air flow measurements with the blower on and off. The second plot shows the differences between the fan on and off data binned every 1 Pa of house pressure. The error bars in the second plot are calculated by combining the standard deviations of the on and off data in the bin in quadrature (i.e., the square root of the sum of the standard deviations squared).

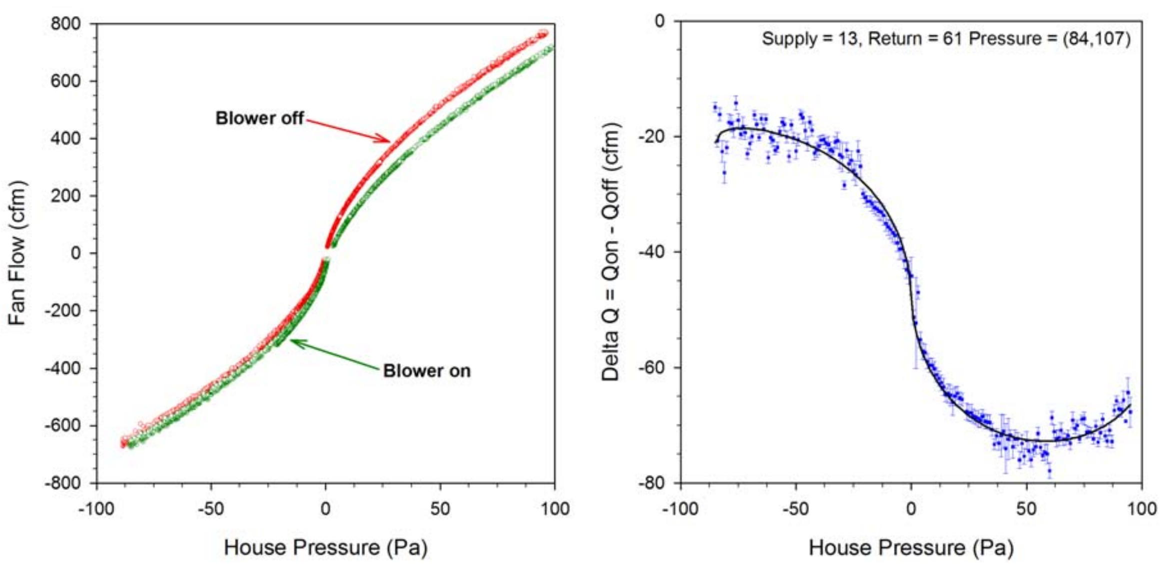

Figure 12. RTU undergoing a pressurization test

The first furnace (Capacity = 71 kBtu/h, nominal air flow = 470 L/s (1000 cfm)) had the whole furnace pressurization test performed ten times and the DeltaQ test six times. The repeated test results indicate a standard deviation of about 10% of the measured leakage - about ±4 L/s (8 cfm) for this furnace. Examining the whole furnace pressurization results in more detail shows that if the furnace is not moved from test to test the repeatability is much better at about 1.5 L/s (3 cfm). The big changes occur when the furnace is moved in and out of the test apparatus. It was also observed during testing that small changes to the blower access panel location would significantly change the air leakage. Thus the standard deviation for the testing is due to the furnace itself changing leakage as well as the repeatability of the test method. This can be thought of as capturing the test-to-test variability for different laboratories testing the same furnace - but having the furnace cabinet components shift during transportation and installation in each test apparatus.

For this furnace the tests that separated the leakage in the positive and negative pressure zones showed that majority of the air leakage was in the negative pressure zone and whole furnace pressurization gave higher leakage results than depressurization.

The tracer gas results are consistently lower than for the other tests. Given the caveats about mixing problems with this testing we believe that these test results are due to experimental error for the tracer testing.

To see how much this could be improved, the visible leaks were sealed with duct tape and the negative pressure zone was retested. Surprisingly there was no observed reduction in the measured air leakage.

Table 2. Air leakage test results for Furnace #1 in LBNL laboratory

Test	Positive pressure zone leakage, L/s (cfm)	Negative pressure zone leakage, L/s (cfm)	Total Leakage, L/s (cfm)
Whole Furnace Pressurization			40±4 (84±8)
Whole Furnace Depressurization			30 (64)
DeltaQ	10±1 (21±2)	29±3 (62±6)	39 (83)
Two Fan Split	6 (13)	27 (57)	33 (70)
Tracer	5 (10)	15 (32)	20 (42)
Pressurization all leaks sealed – return only		28 (60)	

The second furnace (Capacity = 71 kBtu/h, nominal air flow = 470 L/s (1000 cfm)) had the whole furnace pressurization test performed five times, the DeltaQ test three times, and the tracer gas testing three times. The repeated test results indicate different variability for the test methods. For whole furnace pressurization the standard deviation was of about 15% of the measured leakage - about ±4.5L/s (9 cfm). As for furnace #1, examining the whole furnace pressurization results in more detail shows that if the furnace is not moved from test to test the repeatability is much better: for the three tests on one day of testing the results were 2% to 5% of measured leakage (0.5 to 1.5 L/s (1 to 2.5 cfm)). Once again, the big changes occur when the furnace is moved in and out of the test apparatus. The DeltaQ tests were performed on three different days and reflect a larger standard deviation as a consequence of the furnace being moved each time. The tracer tests were all performed without moving the furnace and show very little variation from test to test. This is another indication that much of the test to test variability comes from the furnace leakage actually changing due to moving furnace rather than from the test methods themselves.

For this furnace the tests that separated the leakage in the positive and negative pressure zones showed that majority of the air leakage was in the negative pressure zone and whole furnace pressurization gave higher leakage results than depressurization.

As with furnace #1, the tracer gas results are consistently lower than for the other tests. This further reinforces concerns about mixing problems and that the differences between tracer gas testing and the other tests are due to experimental error for the tracer testing.

To see how much the leakage could be improved, the visible leaks were sealed with duct tape and the whole furnace was retested using pressurization. There was a significant reduction to about one third of the unsealed leakage - but not zero. This indicates that there are leaks that are not easily observable and that better sealing could lead to reduced air leakage. This is significant for this furnace in terms of whole system air leakage reduction credits in Title 24. Pre-sealing, the total pressurization leakage is about 2.5% of the nominal air flow of 565 L/s (1200 cfm) for this three ton system (note that the duct pressurization for total duct system leakage in Title 24 is at 25 Pa (0.1 in. water) so these furnace leakage results were reduced accordingly from the 125 Pa (0.5 in. water) test pressure results presented in the table). This is a substantial fraction of the 6% allowable leakage in order to obtain the tight duct system credit in Title 24. After sealing, the furnace only contributes about 1% of air leakage at 25 Pa (0.1 in. water).

The two-fan split test data were used to estimate the internal air leakage between the positive and negative pressure zones at 17 L/s (36 cfm).

Table 3. Air leakage test results for Furnace #2 in LBNL laboratory

Test	Positive pressure zone leakage, L/s (cfm)	Negative pressure zone leakage, L/s (cfm)	Total Leakage, L/s (cfm)	Notes
Whole Furnace Pressurization			30±4 (63±9)	Same day test results with furnace not moved between tests: Day 1: 33±1.5 (69±2.5) and Day 2: 25±0.5 (54±1)
Whole Furnace Depressurization			25 (52)	
DeltaQ	7±1.5 (15±3)	26±6 (55±13)	33 (70)	
2 Fans	5 (11)	21 (45)	26 (56)	
Two Fan Split	22 (47)	38 (80)	60 (127)	
Tracer	4±1 (8±2)	11±0.5 (24±1)	15 (32)	
All leaks sealed Pressurization			11 (23)	

The third furnace (Capacity = 80 kBtu/h, nominal air flow = 470 L/s (1000 cfm)) had the whole furnace pressurization test performed three times and the DeltaQ test three times. For whole furnace pressurization the tests were done on two days with a 12 L/s (25 cfm) change from day 1 to day 2. The two tests on the second day are only different by 0.5 L/s (1 cfm). As with the other two furnaces, the big changes occur when the furnace is moved in and out of the test apparatus. The DeltaQ tests were performed on three different days and show a similar variability to the pressurization tests.

For this furnace the tests that separated the leakage in the positive and negative pressure zones showed that the majority of the air leakage was in the negative pressure zone. This is consistent with the other two furnaces.

As with furnace #1, the tracer gas results are consistently lower than for the other tests although the repeatability is excellent - indicating that the test procedure has systematic bias (almost certainly due to insufficient mixing of tracer gas).

Table 4. Air leakage test results for Furnace #3 in LBNL laboratory

Test	Positive pressure zone leakage, L/s (cfm)	Negative pressure zone leakage, L/s (cfm)	Total Leakage, L/s (cfm)	Notes
Whole Furnace Pressurization			20±7 (43±15)	Day1: 12 (25) Day 2: 24 (51) and 25 (52)
DeltaQ	5±1.5 (11±3)	23±6 (48±13)	28 (60)	
Tracer	4±0.5 (8±1)	14±0.2 (30±0.4)	18 (38)	

The RTU (Capacity = 6 tons; nominal air flow = 1130 L/s (2400 cfm)) in Figure 13 was only tested a single time and the pressurization test result was 11 L/s (23 cfm). This was the least leaky piece of heating/cooling equipment tested despite being by far the physically largest. Visual inspection showed that the cabinet was constructed better than the furnaces and it used gaskets on seams - in particular the blower compartment door that had a foam rubber gasket seal that worked well even with a dent in the blower access panel (see Figure 14).

Figure 13. RTU undergoing a pressurization test

Figure 14. Dent in RTU blower access panel.

Two sheet metal wye fittings using different construction techniques (labeled A and B) were tested using pressurization to 125 Pa (05 in. water). The fittings shown in Figure 15 had 2 L/s (4 cfm) and 0.5

L/s (1 cfm) respectively. The leakage was from sheet metal seams shown in Figure 16 from wye A and B.

Figure 15. Sheet metal wyes "A" and "B"

Figure 16. Sheet metal seams on wyes “A” and “B”

Two factory assembled splitter boxes (Figure 17) using duct board and starting collars were tested using pressurization. Initial tests were made on them as they came from the factory and they had 4 L/s (8 cfm) and 3.5 L/s (7cfm) leakage. Observations during testing indicated that the leakage was from around the starter collars (Figure 18). The collars were sealed with mastic (Figure 19) and retested. The sealing reduced leakage to 0.7 L/s (1.5 cfm) and 0.5 L/s (1 cfm) respectively.

Figure 17 Foam board and starter collar splitter boxes

Figure 18 Splitter box starter leak locations

Figure 19. Splitter box starter collars sealed with mastic.

Two VAV boxes were tested using pressurization. Figure 20 shows the VAV boxes being tested with clear duct mask used to seal one end and a small brass air flow nozzle and fan used to pressurize them. Their 125 Pa air leakage was 6 L/s (13 cfm) and 7 L/s (16 cfm). At a nominal air flow of 425 L/s (900 cfm) this corresponds to about 1.5% to 2% of nominal air flow.

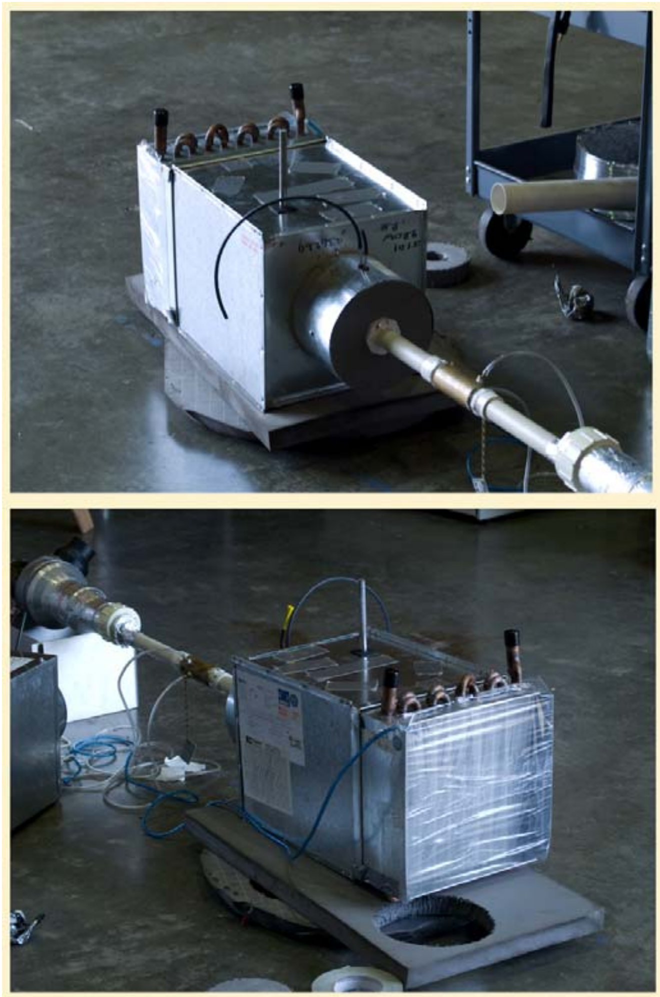

Figure 20. VAV boxes undergoing pressurization testing.

3.2. Small Chamber Furnace Leakage at IBACOS test laboratory

3.2.1. Background Air Leakage

Before any furnace air leakage measurements were made, the background air leakage of the test apparatus was measured by covering the furnace mounting panels with duct masking tape and pressurizing the test apparatus to 125 Pa (0.5 in. water). For pressurization the background air leakage was 17 cfm (8 L/s) and for depressurization 15 cfm (7 L/s). These results were subtracted from subsequent furnace leakage testing.

3.2.2. Bryant Furnace Pressurization

With the test chamber pressurized to 125 Pa (0.5 in. water), the supply pressure was 132 Pa (0.53 in. water) and return pressure 67 Pa (0.27 in. water). These variations in static pressure in the system are an illustration of how much variability there can be in this ostensibly simple measurement[4]. The corresponding air leakage was 30 L/s (64 cfm). For depressurization the pressures were more uniform (indicating that the non-uniformity is likely due to internal air flows) and with the test chamber at -125 Pa (0.5 in. water), the supply pressure was -128 Pa (-0.514 in. water) and return -92 Pa (-0.37 in. water). The depressurization air leakage was 60 cfm. Subtracting the background air leakage results in pressurization air leakage of 22 L/s (47 cfm) and depressurization leakage of 21 L/s (45 cfm).

3.2.3. Bryant Furnace DeltaQ

To allow comparison to pressurization testing the target supply and return plenum static pressure differences were ±125 Pa (0.5 in. water) for supply and return. However, this was not attainable with this furnace mostly because the blower could not maintain this pressure difference of 250 Pa (1.0 in. water). The test pressures were set lower: at 60 Pa (0.241 in. water) for the supply side and -65 Pa(-0.261 in. water) for the return side. These difficulties and limits of blower performance indicate that less extreme pressures may have to be used for some furnace in laboratory DeltaQ testing. A reasonable proposed target is 125 Pa (0.5 in. water) pressure difference between the positive and negative sides of the equipment under test, with targets of +62.5Pa (0.25 in. water) supply side static pressure difference and -62.5 Pa (-0.25 in. water) return side static pressure difference.

The first set of DeltaQ measurements were performed using a different controller (Energy Conservatory APT). The test method and analysis are the same – the major difference is that the controls for slowly ramping up the duct blaster fan are not as good as when using the controller the software was developed for (Energy Conservatory DG700). The ramping with the APT tends to be too fast in certain ranges that could lead to increased hysteresis effects and errors in the DeltaQ testing. The data shown in Figure 21 show evidence of this hysteresis. In Figure 21, the upper plot shows the raw data – i.e., every measured pressure and flow. These data are acquired at a rate of approximately 4.5 points per second resulting in over 3000 data points being used in the analysis. These individual data were sorted into 5 Pa wide bins and averaged within each bin. The figures showing the test results have both the raw data (all 3000 points) and the binned data together with the standard deviation of the data points in each bin represented by uncertainty bars. These binned data

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[4] If the air leakage flows were large enough, they could cause static pressure changes of this magnitude, however, this was not the case for this furnace.

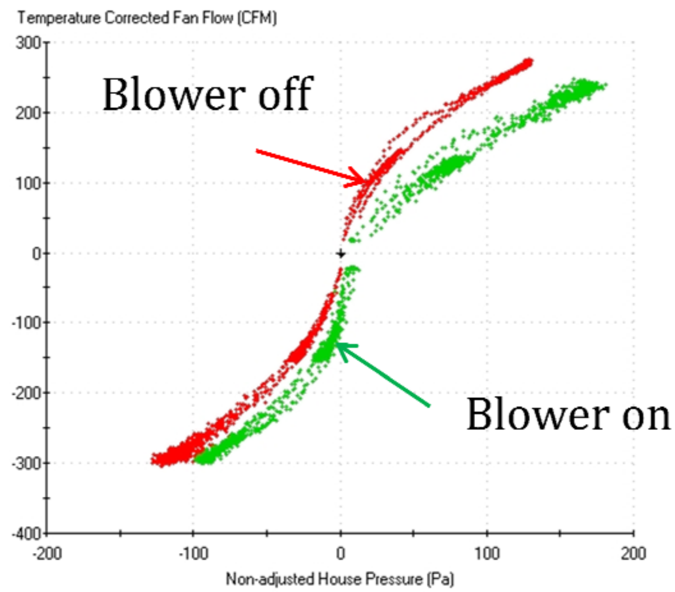
were then used in the DeltaQ analysis. Slower ramping performed at the LBNL test facility does not show this hysteresis (see Figure 12) and indicates that ramping speeds longer than the 90 seconds used in these small chamber tests would give better test results.

Figure 21 illustrates the test results from the first Bryant test. In this, and subsequent, figures the upper plot shows all the measured data points and the lower plot shows the binned DeltaQ data together with the fitted DeltaQ model (shown by the solid line) . The upper plot shows how the measured fan/flowmeter air flow changes as the system (or "house") pressure varies for the furnace blower on and off. The blower on data are green and blower off data are red. The data were binned every 5 Pa and the difference between blower on and off (the DeltaQ) calculated for each bin by least squares fitting to the on and off data and calculating the on and off points (and their differences – the "DeltaQ") at the mid-point of the bin. The binned data are shown in the lower plot. For each bin an uncertainty estimate was made for DeltaQ by combining the residuals of the two least squares fits (one to the On data and one to the off data) in quadrature (i.e., the square root of the sum of the squares). This uncertainty estimate is shown in the vertical uncertainty bars in the lower plot. The results from the DeltaQ analysis indicated 25 L/s (54 cfm) of return leakage and -7 L/s (-14 cfm) of supply leakage. This negative result for supply leakage has the wrong sign. The nature of the DeltaQ test can sometimes lead to negative air flow results if the uncertainty in the air flow is greater then the actual air flow – or if there are other sources of error such as the hysteresis effects and duct movement problems that were observed in these tests. In these cases, previous experience (based on DeltaQ tests in other laboratory duct systems of known air leakage combined with the knowledge that the DetlaQ test is better at measuring the difference between supply and return leakage rather than its sum (Andrews (2000)) has shown that it is best to assume that the negative leakage is really zero and combine the negative result with the dominant leakage. In this case it means that the supply leakage is close enough to zero to be called zero and the return leakage is 32L/s (68 cfm) combined. Subtracting the 8 L/s (16 cfm) average background leakage results in a leakage of 25 L/s (52 cfm). The measured air flow and pressure signals show significant fluctuations in Figure 21, particularly at higher air flows and with the furnace blower operating. For this test, the results showed air flow fluctuations of ±10 L/s (± 20 cfm). This represented 10 to 20% of the measured flow differences (DeltaQs) and is most likely responsible for these negative air leakage results. The most probable source of these fluctuations is excessive turbulence within the test chamber due to its small size relative to the attached ducts.

The DeltaQ in this furnace test was repeated using the DG700 controller with its improved control algorithm. The results for this test (shown in Figure 22) showed the same incorrect sign issues for supply leakage: -14 L/s (-30 cfm) for supply and 26 L/s (53 cfm) for return. Combining this leakage leads to zero supply leakage and 39 L/s (83 cfm) return leakage, or 32 L/s (67 cfm) after the background was subtracted. Examination of the test data and observations of the test system during the experiments indicate that this test result may be due to a combination of the flex ducts shifting during the test and the large offset pressures[5]. To address these issues for subsequent tests, the ducts

[5] The offset pressures are the pressures introduced with the duct blaster off by the imbalance between supply and return leakage. It could be reduced by increasing the size of the hole in the test chamber – however this trades off test precision.

were rearranged to be on the floor so that they could be secured with duct tape and a second leak hole was added to the test chamber.



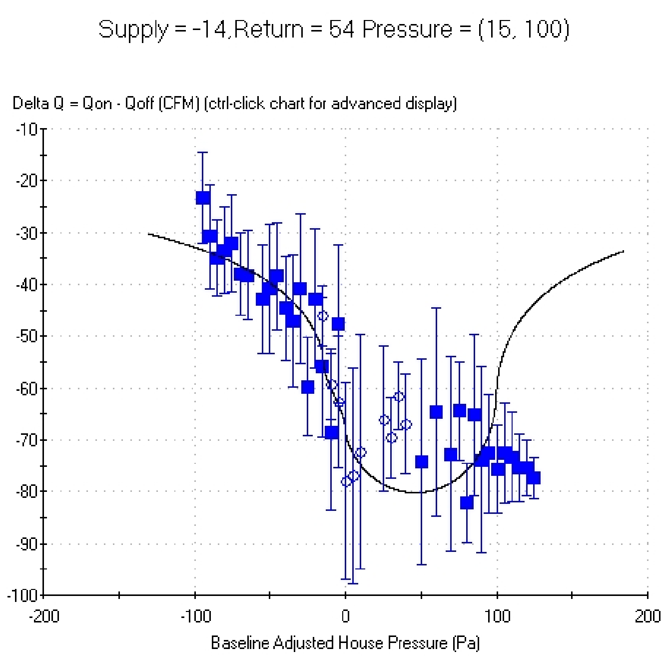
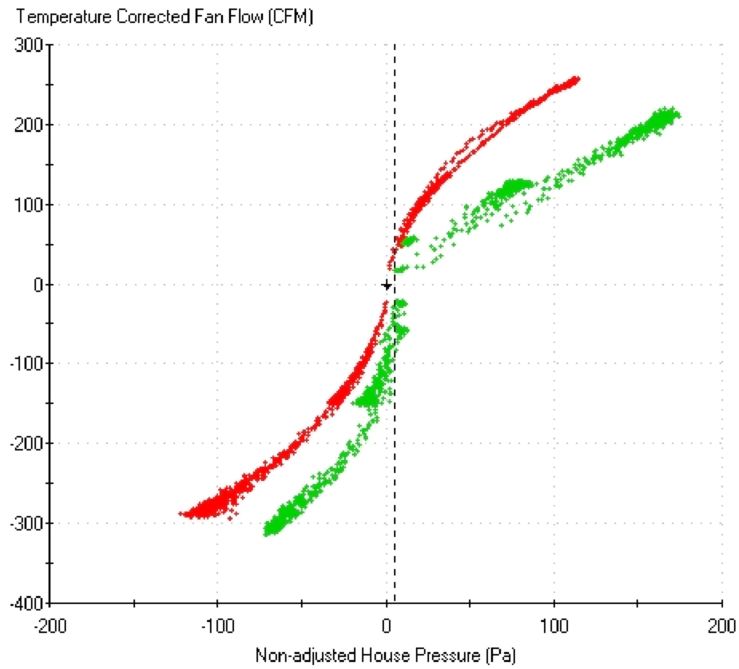


Figure 21. DeltaQ tests results the Bryant Furnace using the APT and one hole in the test chamber

Because of the concern over large offset pressures and the interaction with the air flow into the test chamber and the flow through the hole in the test chamber, a second hole was added to the test chamber that approximately doubles its total leakage. The DeltaQ tests were repeated using this configuration and are shown in Figure 23. The corresponding DeltaQ test results were no better at -12 L/s (-26 cfm) for supply and 44 L/s (94 cfm) for return 49 Ls/ (104 cfm) background corrected). There is still a negative result for supply leakage that has the wrong sign. As discussed above, the high pressure and flow fluctuations contribute to this result. It is also possible that the close proximity (just a few inches) between where air flow enters the chamber and the hole in the chamber meant that these flows could interact significantly. This would mean that the measured chamber pressure difference was not an indicator of the pressure difference (and therefore air flow) acting across the hole in the test chamber. An indicator that this may be the case is seen in the details of the measured air flow data that do not follow the characteristic curves expected in DeltaQ testing. Another possibility is the valving of leaks – where they get larger or smaller depending on the direction and magnitude of air flow through the leak. For example, the large changes in DeltaQ in Figure 23 near zero pressure could be due to the blower access panel shifting during the test – this type of result was observed in tests at the LBNL test facility.



Supply = -30,Return = 53 Pressure = (100, 100)

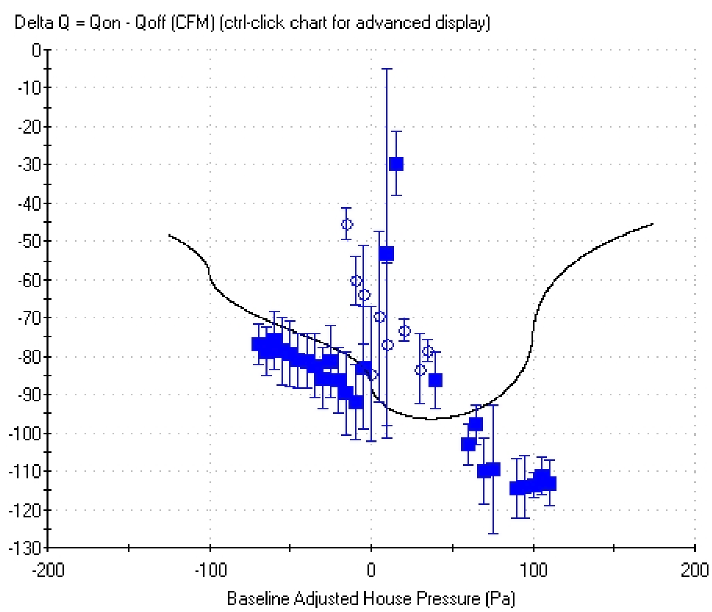
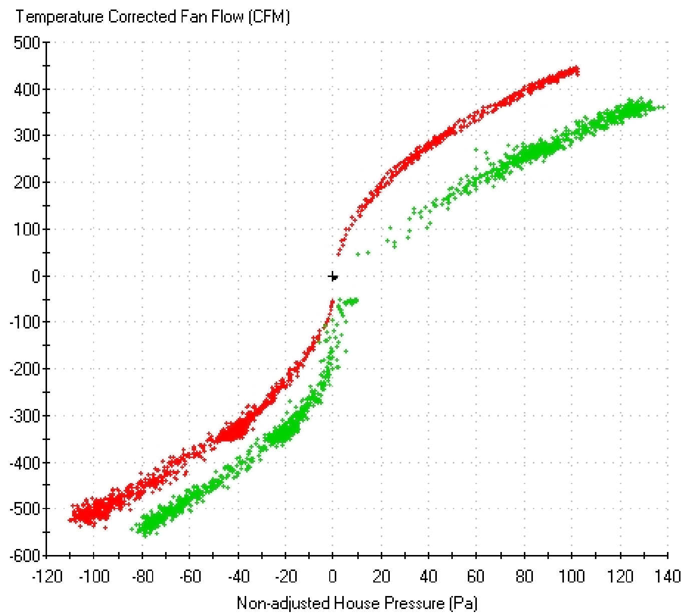


Figure 22. Bryant using the DG700 with one hole in test chamber



Supply = -26,Return = 94 Pressure = (85, 100)

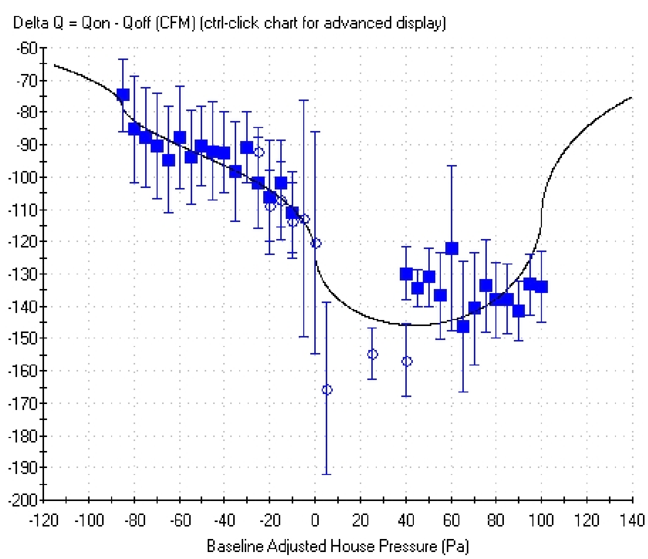


Figure 23. Bryant with second hole in test chamber

3.2.4. Lennox Furnace Pressurization

With the test chamber pressurized to 125 Pa (0.5 in. water), the corresponding air leakage was 51 L/s (108 cfm). For depressurization the air leakage was 92 L/s (195 cfm). Subtracting the background air leakage results in pressurization air leakage of 43 L/s (91 cfm) and depressurization leakage of 84 L/s (178 cfm). Averaging the pressurization and depressurization results gives an air leakage of 63 L/s

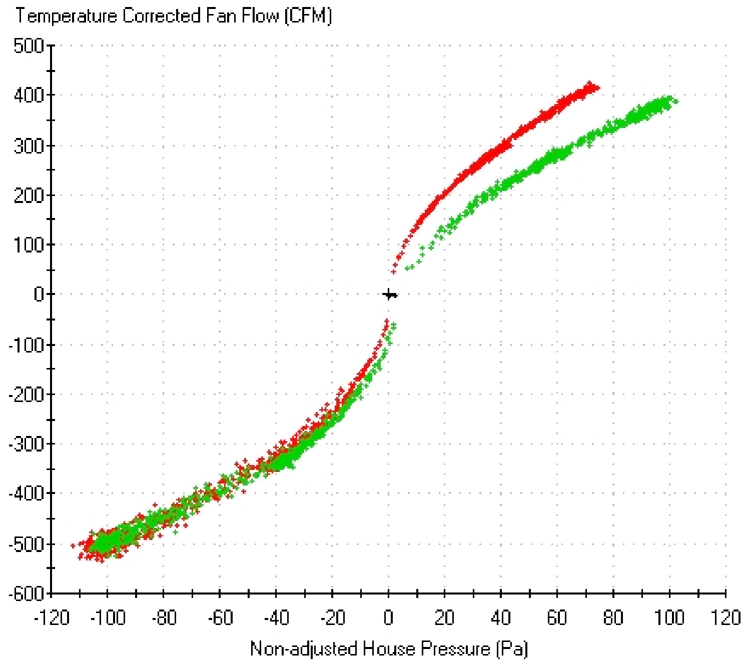
(135 cfm). This is much greater air leakage than the Bryant furnace – a result that could not have been obtained by visual observation.

3.2.5. Lennox Furnace DeltaQ

For the Lennox furnace tests the test chamber was operated with two open holes, the ducts were taped to the ground and an effort was made to remove any sharp bends on the ducts between the furnace and the test chamber. The Lennox furnace supply and return pressures were much easier to modulate than the Bryant furnace. For the Lennox furnace the dampers were set to have 110 Pa (0.442 in. water) supply pressure and -125 Pa (-0.5 in. water) return pressure.

All the Lennox tests were performed using the DG700. The first Lennox test (shown in Figure 24) gave results of 16 L/s (33 cfm) supply leakage and 38 L/s (81 cfm) return leakage (or 46 L/s (98 cfm) total background corrected). This test was repeated with system vent openings (for A/C coil drain and combustion product exhaust) sealed and the results are shown in Figure 25. The results were similar to the first Lennox test: 18 L/s (38 cfm) and 39 L/s (82 cfm) for supply and return respectively (49 L/s (104 cfm) total background corrected), indicating that these openings were not major contributors to the leakage.

The Lennox DeltaQ data were much better than for the Bryant furnace. The data shown in Figures 24 and 25 show less hysteresis and a less noisy signal than for the Bryant furnace. This is reflected in the reductions in variability in each DeltaQ data bin for this furnace. It is possible that the steady improvements made to the apparatus helped: adding leakage to the test chamber, fixing the ducts in place and smoothing out the bends on the ducting. The pressure and air flow fluctuations were smaller and had the correct signs. This improvement of results with the refining of the test apparatus indicates that care must be taken when performing these DeltaQ style experiments. For future it is recommended that a larger more leaky test chamber be used, that the leaks added to the test chamber are placed as far away as possible from the inlet and outlet connections to the ducts and the fan/flowmeter, that ducts are restrained so as not to move during the test and that sharp turns in the ducting are avoided.



Supply = 33,Return = 81 Pressure = (100, 100)

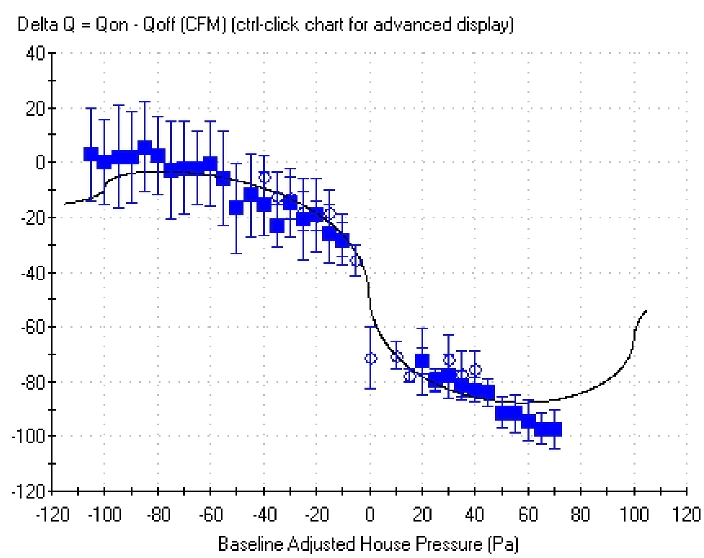
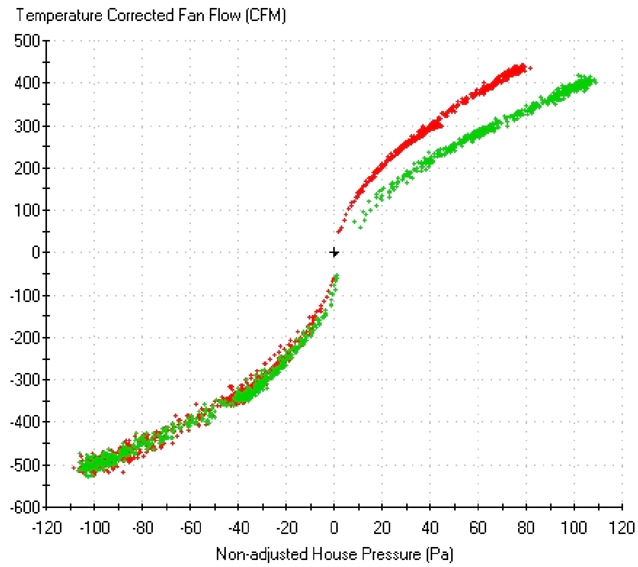


Figure 24. Small Chamber Lennox Furnace DeltaQ test 1



Supply = 38,Return = 82 Pressure = (100, 100)

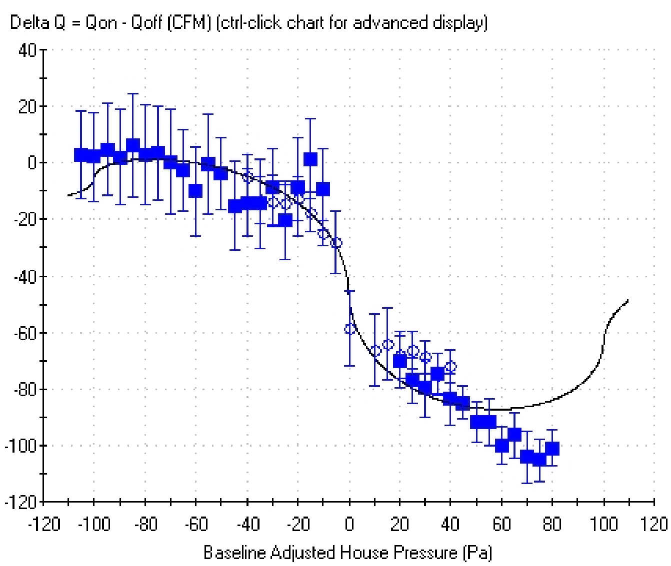


Figure 25. Small Chamber Lennox Furnace Delta Q Test 2

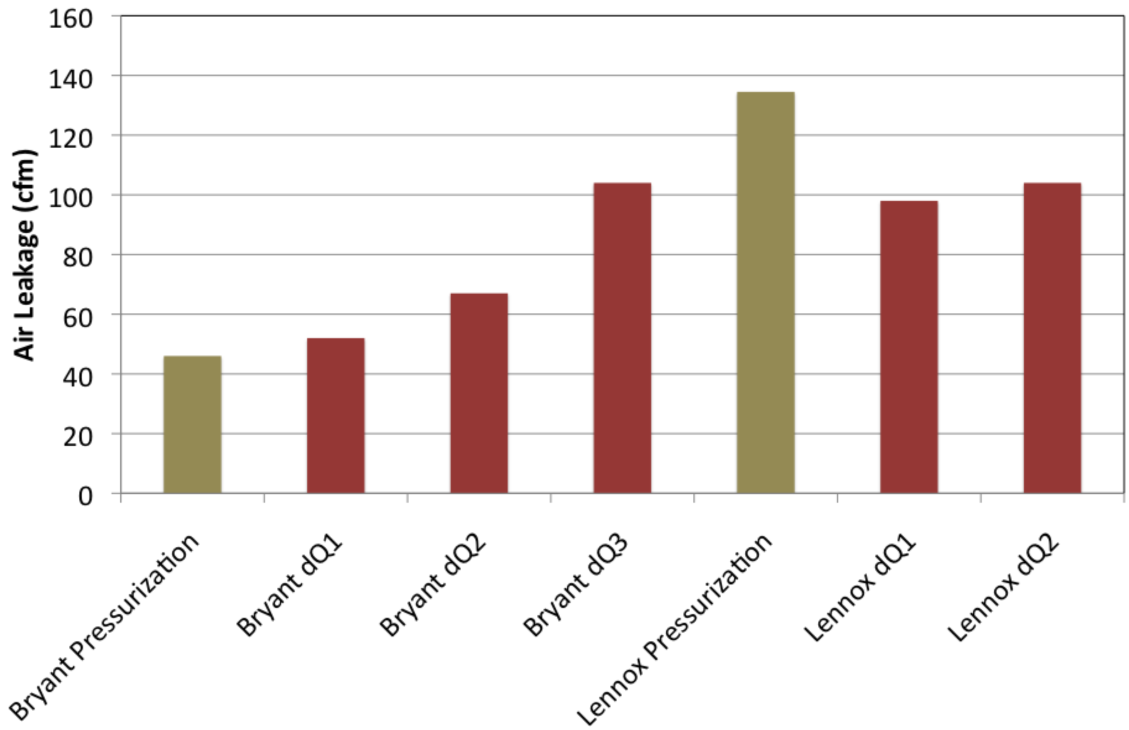


Figure 26. Comparison of pressurization and DeltaQ test results for the Bryant and Lannox furnaces using the small test chamber.

3.2.6. Summary of small chamber tests

These test results indicate that for pressurization testing there are significant differences between the two furnaces. The Bryant furnace was typical of other tested furnaces with air leakage between 40 and 50 cfm (20 to 25 L/s). The Lennox furnace was significantly worse at almost 150 cfm (75 L/s). Careful observation of the furnace could not determine where these leaks were located. However, the base/bottom of this furnace was two pieces of loosely fitting sheetmetal with considerable potential for leakage (although the furnace was tested standing on this base).

Figure 26 compares all the small chamber test results where the supply and return leaks have been added to obtain a total leakage value. Conclusions are limited by the small number of tests and the problems found with the DeltaQ testing – particularly for the Bryant furnace. However, it is possible to see that the pressurization and DeltaQ tests generally agreed about which furnace had the most air leakage even though they measured different air flows.

The DeltaQ testing was inconclusive due to doubts about interactions between the test chamber leakage (added holes) and the high velocity air flow inside the chamber when the furnace blower was operating, and issues about hysteresis due to ramping too fast and for some tests the ducts may have moved too much during the testing. Other work performed at LBNL for this study using a much larger test chamber did not show these test oddities. This implies that the test chamber used in the testing at IBACOS was too small. Therefore it is recommended that other DeltaQ testing of furnaces use a larger test chamber than used in the IBACOS testing. In addition, the leaks added to the test chamber need to be placed as far away as possible from the inlet and outlet connections to the ducts and the fan/flowmeter, the ducts should be restrained so as not to move during the test and that sharp turns in the ducting must be avoided.

3.3. Power and Airflow Measurement for Furnace Blowers

Residential furnace blower power consumption is in the range of 100 W to 1000 W using single phase power. Typical power consumption is typically about 500 W based on a wide range of field surveys (summarized in Walker (2007)). The power consumption depends on the selected blower speed, blower capacity, and system air flow resistance. There are two types of motors used in furnace blowers: Permanent Split Capacitor (PSC) and Brushless Permanent Magnet (BPM). PSC's are the most common blower motor (roughly 95% of the market (Sachs and Smith (2003)). BPM's are often referred to as variable speed blowers and are used in higher end premium equipment that often utilize their ability to operate over a wide speed range and gradually increase speed and air flow.

The power and airflow measurements examined in this study include methods for both field and laboratory testing. For laboratory testing a collaboration was undertaken with the Canadian General Standards board who are developing a performance Standard for air handlers in residential space conditioning systems. Information from the current study was used in the development of this CGSB standard. For field testing compromises are made in terms of accuracy and precision due to limits on the type of measurement equipment that can be used. There is also a need for increased simplicity in the field measurements due to the limited time available for testing and the lower level of training and expertise typically available with field technicians.

3.3.1. Using Correlations between rated furnace input capacity and blower power

A simple approach to determine furnace blower power would be to assume a correlation between blower power and furnace input rating. This would only require a reading of the furnace nameplate to determine the power consumption. Unfortunately, there is little correlation between rated furnace input capacity and blower power. Figures 27 and 28 show measured field data on furnaces taken from Pigg (2003). For the two-speed blowers in low speed/low-fire mode (Figure 28) the correlation between blower power and input capacity has an R-squared value of only 0.59 and the blower power increases at one-fifth the rate of increasing input capacity. Therefore, a doubling of input capacity only increases blower power consumption by 20%. Similarly, for BPM blowers on high speed the R-squared value is 0.60 and the blower power increases at only 8% of the rate at which input capacity increases. The PSC blowers have similar results with even less correlation (R-squared = 0.54) and blower power increases at 12% of the input capacity. These results are because many other factors – such as the air flow resistance of the duct system and the selected blower speed have as big an influence as the furnace input capacity. Additional data taken from GAMA Directory and analyzed by Davis Energy Group (2003) showed a similar lack of correlation – even under idealized laboratory conditions. These results imply that rated equipment capacity alone cannot be used to determine blower power consumption.

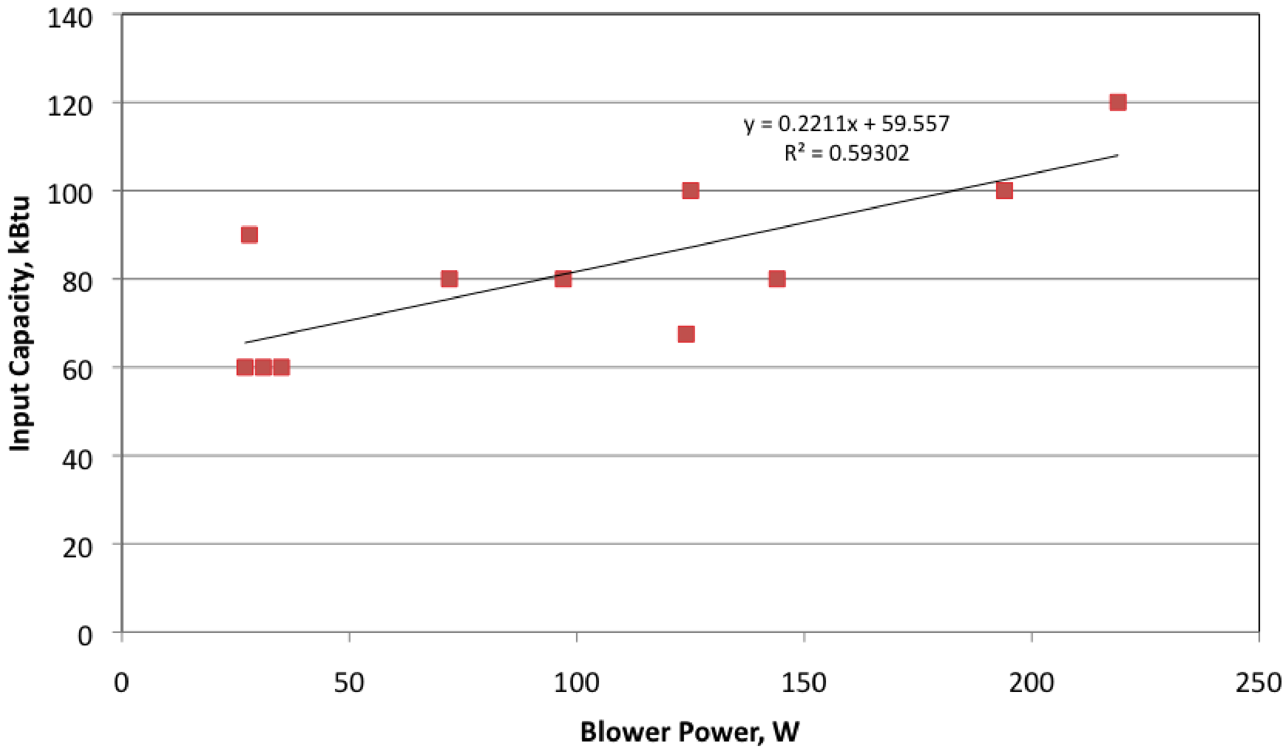


Figure 27. Correlation between furnace input capacity and blower power for two-speed furnaces with BPM motors in low speed/low fire rate operation.

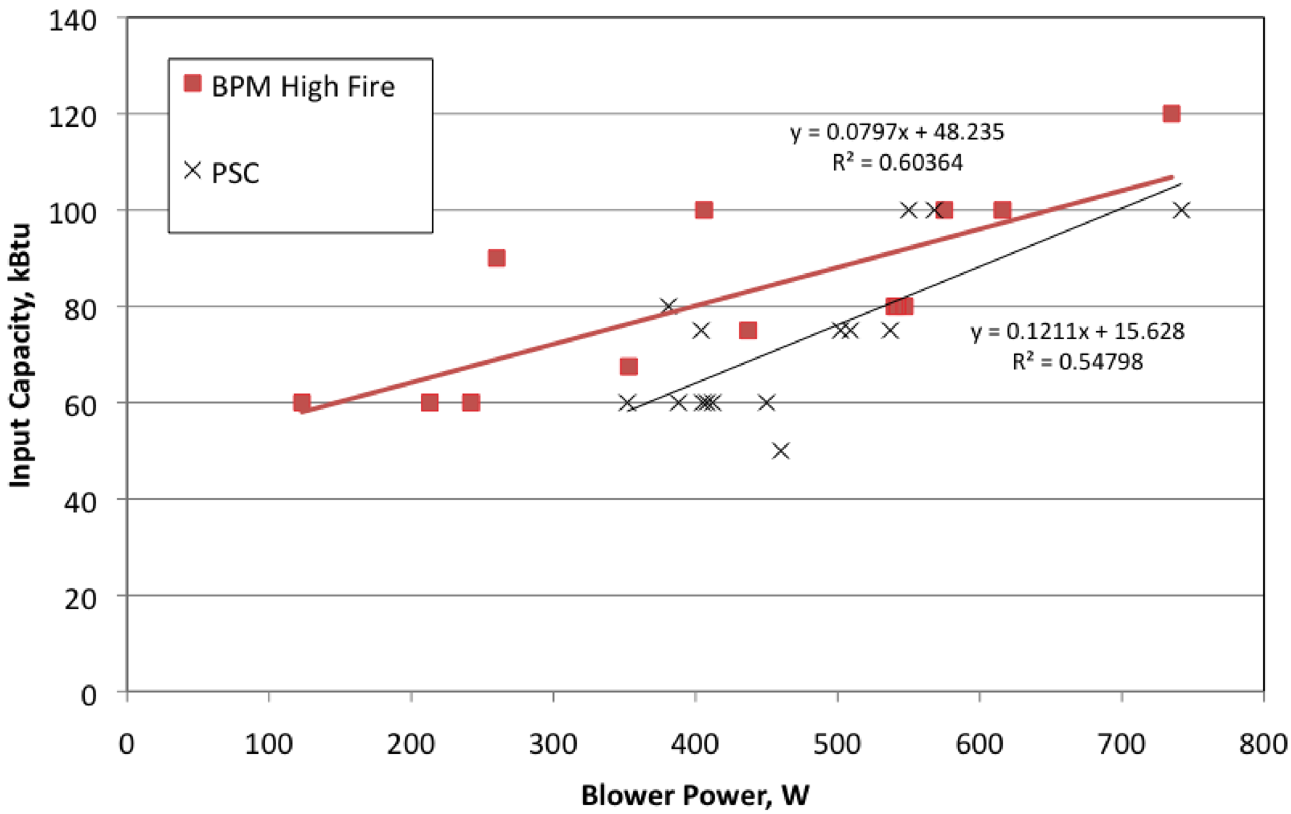


Figure 28. Correlation between furnace input capacity and blower power for two-speed furnaces in high speed/high fire rate operation with BPM blowers and PSC blowers.

3.3.2. Using Manufacturer's data and measured static pressures.

Some manufacturers publish data on air flow and power consumption based on static pressures. If these data are available, then a measurement of static pressure will allow both air flow and power consumption to be known without additional measurements. This allows for cheaper and quicker testing. This has the appeal of being a useful diagnostic for other system problems – e.g., overly high static pressure. Many contractors will have a pressure measuring tool already and they don't have to learn how to use another tool. They also understand the concept of static pressures for duct systems. It is also a simple test – not requiring additional equipment or training for electrical and air flow measurement.

The biggest problem with this approach is that most of the time the information may not be available either because the manufacturer does not provide it, or it is not available to the field technician. One possibility is that the Energy Commission could make it a requirement that all new furnaces be sold with this information on a label attached to the furnace – preferably on the inside of the blower access panel/door. For existing furnaces this is more difficult as the data may never have existed and if it did it is no longer available from manufacturers, distributors or installers.

Another issue with manufacturer's published data is that the data are provided for a furnace with no filter. Therefore any field or laboratory testing using this method would need to remove the filter and seal the filter opening. Removing the filter leads to lower system static pressures and therefore incorrect (higher) air flows in field testing. In laboratory testing dampers and auxiliary fans can be used to control airflow and static pressures so this is not an issue.

Experience of the authors (and others) in both field and laboratory testing has shown that static pressure measurement has complications arising from non-uniformity of pressures and flows within the tested equipment. In addition, static pressure probes can be sensitive to orientation with respect to the air flow inside the equipment. Some static pressure probes have several small holes around the perimeter of the sample probe. These small holes will only work as an effective static pressure measurement if the air flow is straight over the probe tip. In practice this is difficult to ensure and it is easily demonstrated that moving the probe around results in significant changes to the measured pressures leading to too much test uncertainty.

A possible alternative to using a static pressure probe is to drill a small hole in the sheet metal of the equipment cabinet - creating a wall pressure tap. Ideally, furnaces would have such pressure taps in place from the manufacturer. This problem could be reduced by having manufacturer's install static pressure ports on equipment that are the same ones that are used during the calibration process used to generate their data sheets. Currently the small hole would need to be drilled by a technician. To connect to a manometer via tubing, a removable magnetic ring with a tubing port attached is placed over the small hole. This wall pressure tap method is used in ANSI/AMCA Standard 210/ASHRAE Standard 51 (2007). In addition, previous laboratory tests at LBNL have found that this works well. It has the advantage of requiring only a small hole through the sheet metal thus reducing the risk of coil/refrigerant piping/drain pan damage associated with the larger hole required for the bigger probe. This concept of using a "wall tap" has been found to be a more reliable and consistent method of measuring static pressures.

3.3.3. Direct Measurement

Variable speed blowers and their associated controls can present a challenge to achieving a fixed operating speed for power consumption testing. Some equipment has controllers that gradually increase blower speed rather than having fixed blower speeds. The proprietry nature of these controllers does not lead to easy specification of test methods and procedures. At this time it is necessary to use approaches specific to each furnace and controller – such as turning on the blower and waiting 5 to 10 minutes for the blower to reach a steady speed. In the future it is recommended that manufacturers include controller settings that allow for fixed speed operation to allow for blower power consumption measurement and rating.

Direct measurement of furnace blower power consumption are somewhat complicated by the electric power consumed by other parts of the furnace such as the operation of combustion air blowers, gas valves and ignitors. Typical power consumption for these ancillary loads are (from Lutz et al. (2006)):

- Controls: 5W (PSC) and 10 W (BPM)
- Combustion air blower: 75 W (reduced to 60 W for a two-stage furnace in low-fire mode)
- Ignitor: 400 W

This implies that care must be taken when inferring blower power consumption from total furnace power consumption. There are two paths to directly measure blower power:

Path 1. Isolate the blower from the other power consuming equipment in the furnace.

This is the more difficult and time consuming path, but provides the measurement of blower power only. To isolate the blower it is necessary to open the furnace and make electrical connections to the blower power wiring. An electric power meter is used that measures the current to the blower motor using a clamp-on current transformer (CT) and simultaneously measures the operating voltage. For a meter with a remote CT, wires from the CT to the meter are fed out of the furnace through an existing knockout or a hole is drilled for this purpose. For direct clamp-on meters (with the CT built into the meter itself) the meter is left inside the blower cabinet during operation – and a meter with automatic datalogging capacity is used to record the power consumption. A voltage measurement is made either in the power supply to the furnace, or within the furnace itself. Accuracy for commercially available measurement instrumentation is typically +/- 2% + 5 digits, so at 500W accuracy is +/- 10W. They cost from $600-$2000 depending on features. The use of these meters may require an electrician.

Figure 29. Examples of clamp-on power meters

The blower can be activated in several ways:

1. Use the “fan only” switch at the thermostat (or on the furnace in some older models). The drawback with this method is that the blower speed settings for fan only mode may not be at heat or cooling speeds. It is possible to check this by measuring the static pressure in the supply plenum in the various operating modes. If the purpose of testing requires that the blower operate at its highest speed and fan only mode is not this highest speed, then the blower settings are changed to make fan only operation be high speed for the purposes of this test (and then reset to the original blower speed after the test).

2. Via temperature changes at the thermostat. The thermostat temperature settings are adjusted so as to turn on either the heating or cooling.   If the test requires the blower to be on its highest operating speed, then this will often be cooling. In field testing, when testing in

cooling mode, the circuit breaker for the outdoor unit could be removed so that the outdoor fan and compressor do not operate. This is particularly useful when testing in cool weather when the air conditioner should not operate in order to protect it from damage.

3. Thermostat bypass. The thermostat is bypassed by connecting the appropriate terminals on the furnace control board/panel inside the furnace. This is the most practical approach for laboratory testing.

    - Red to green = fan only
    - Red to yellow = cooling
    - Red to white = heating

    This allows "fan only" operation directly – even if the thermostat does not have a fan only switch. As with the previous methods – if the testing requires that a certain blower speed to be used (typically the highest operating speed) then the thermostat bypass can be used in place of changing temperatures at the thermostat and/or the jumper settings for "fan only" operation temporarily changed.

Path 2: Measuring the total energy consumption of the furnace.

On this path the blower compartment is not opened to isolate the blower power. Instead the total power for the furnace is measured. If the test is performed with the heating activated then the power measurement will also include power to the controls, gas valve and any combustion air or vent blowers. With heating off, the measurement will include the power to the controls, but not the gas valve or combustion air blower. The power meter is connected to the power supplied to the furnace. Ideally, this would be at a junction box near the furnace or at the furnace power switch – if either of these are convenient. Alternatively, the furnace could be opened and connections made where electric power enters the furnace.

There are two ways to account for non-blower power consumption. The first is to use typical values that have been measured for furnaces for the controls and combustion air blowers and subtract these from the total. It could be possible to subtract these typical values from total power measurement to obtain the blower only power consumption. However, in a Wisconsin study Pigg et al. (2003) measured 30 furnaces and found that the range of power for the combustion air blower was 9 W to 153 W in low fire and 20 W to 114 W in high fire (not all furnaces were two stage – single stage furnace data were recorded as low-fire). This large range means that simple subtraction of a single value for controls and combustion air blowers will give significant errors. Another possibility is to assume that the combustion air blower power scales with furnace input capacity. As with the main blower, the Wisconsin study showed that this was not the case. The average W/kBtu (input capacity) was 4.3 with a standard deviation of 2.8, and as shown in Figure 30, this means that there is little correlation between input capacity and combustion air blower power (R-squared is <0.005). The trend is for combustion air blower power to decrease with increasing input capacity – but this trend is very weak.

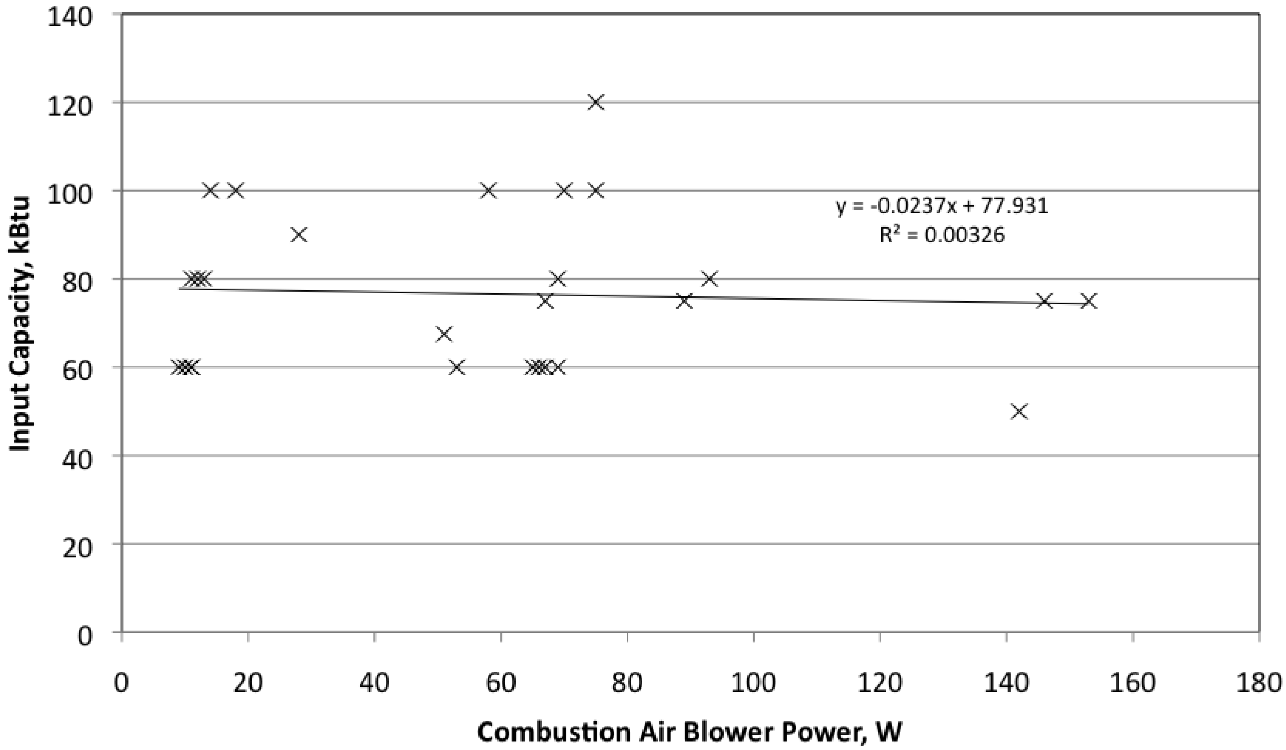


Figure 30. Lack of correlation between furnace input capacity and combustion air blower power.

This large furnace-to-furnace variability and lack of correlation with capacity means that it will be impossible to account for combustion blower power by simple subtraction from total power measurements.

A second method is to measure the individual components of electricity use of the furnace using the furnaces built-in controls. With the furnace not calling for heating or cooling, the power meter measures the standby electric use.

When the furnace calls for heat it goes through a set pattern. First the combustion air fan is activated. The igniter is powered (unless there is a pilot light) and the gas valve opened. After the burners are lit, the igniter is turned off. After the air in the furnace has reached a set temperature, the blower will be activated. The power consumption is measured and recorded at each step in this process. The two key measurements are the power consumption after the igniter is off before and after blower activation. The pre-blower activation measurement will include the controls and combustion air blower. The combustion air blower and control power can then be subtracted from the total to obtain the blower power. An exception to this is for variable speed BPM blowers with “smart” controls. These systems activate the blower in its lowest speed setting as soon as the thermostat calls for heat. In these furnaces the combustion air blower and the central blower are activated at the same time so that their measured power consumption cannot be separated.

A third method is to connect the power meter to measure total furnace power, but use the "fan only" operating condition for the furnace. This method allows for the measurement of blower power only – with out gas valve or combustion fan operation. However – it still includes any power for controls.

If a furnace or air handler uses a standard 120V plug for electric power, then a plug-through watt meter can be used. These are available relatively inexpensively (for as little as $40) and are very simple to use, do not require training and an electrician is not required to use them. Examples shown below are the "Kill-a-Watt" and "Watts-Up" meters. They are plug-through meters where the appliance (in this case the furnace) plugs into the meter which then plugs into the electrical outlet. The power draw of the furnace is read off the front panel/screen of the meter. Questions have been asked about the accuracy of these inexpensive plug-through meters. To examine this issue the meters are evaluated in laboratory tests on a furnace at the LBNL duct testing facility.

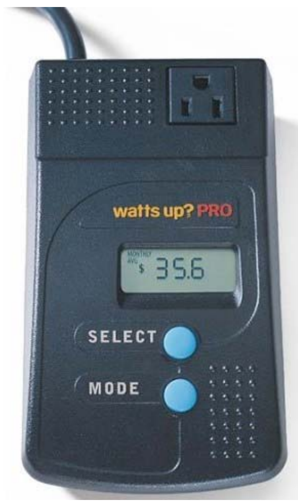


Figure 31. Examples of "plug-through" power meters

Repeatability of furnace power measurements

The blower power consumption of 31 furnaces field tested in a Wisconsin study (Pigg (2003)) were measured twice: before and after a monitoring period. The average difference was 10 W for low fire, 24 W for high fire and 23 W for continuous fan operation. This variability is about 5% of the blower power. The results for the variable speed blowers gave differences of 7 W, 22 W and 24 W indicating that there is no change in repeatability for different blower motors.

For the combustion air blowers, the average power consumption was 57 W and the average difference between the two measurements was 5 W for low fire and 4 W in high fire.

The results indicate that the repeatability of the furnace blower power measurements is about 5% under field conditions using simple plug-through meters. Under more controlled laboratory conditions it is expected that repeatability would be improved – particularly if more accurate power meters are used and efforts are made to control the input voltage supplied to the furnace.

3.3.4. Laboratory testing of furnace power meters

The purpose of these laboratory tests is to evaluate not just the meters - but also some measurement techniques in order to make recommendations for blower power measurement. Power measurement techniques and meters are evaluated using a single furnace. Two different blowers are used: a Permanent Split Capacitor (PSC) four speed blower controlled and powered via an external speed switch, and a Brushless Permanent Magnet (BPM) controlled by the furnace's own control board.

The following power meters were evaluated (in the order of increasing cost/complexity): Kill-A-Watt, Watts-Up Pro, WattMan (demo model), Fluke 41 clamp-on (an older meter) and an Elite-Pro (the only one that needs a computer to function). The first three meters are plug-through devices that measure the total power consumed by the furnace. To isolate the PSC blower and provide speed control these laboratory tests used a control box separate from the furnace controls with its own independent power cord that measured blower power directly without including any ancilliary loads. The other two meters are more adaptable to measuring blower only power because they use a combination of a CT for current and voltage probes for voltage. The CTs could be placed around the blower power wires in the blower compartment - thus isolating the blower power. This requires opening the blower compartment and finding a way to have the CT leads come outside the cabinet safely. In these experiments, this is done through the existing path for the blower compartment wiring. All the meters measured real power (Watts). All but the Kill-A-Watt reported Power Factor (PF). All but the Kill-A-Watt and WattMan reported apparent power (VA). The laboratory test results are summarized in Table 5.

Table 5. Comparison of electric power meter measurements

Carrier 58CTA090		Kill-A-Watt			Watts-Up-Pro			Watt-man			Fluke 41			Elite Pro		
PSC Blower Motor*		**W**	VA	*PF*	**W**	VA	*PF*	**W**	VA	*PF*	**W**	VA	*PF*	**W**	VA	*PF*
Standby		**8**	-	-	**8.7**	26	*0.33*	**8.2**	-	*0.35*	**9**	22	*0.40*	**7**	21	*0.36*
Blower Off		**0**	-	-	**1.3**			**1.2**	-	*0.68*	**0**	2	*0.25*	**0**	0	-
Blower Only	Low	**483**	-	-	**483**	553	*0.86*	**487**	-	*0.87*	**460**	520	*0.88*	**507**	572	*0.88*
	Medium	**577**	-	-	**566**	640	*0.88*	**572**	-	*0.89*	**540**	600	*0.90*	**592**	658	*0.90*
	Medium-High	**663**	-	-	**649**	720	*0.9*	**653**	-	*0.91*	**620**	680	*0.91*	**671**	737	*0.92*
	High	**750**	-	-	**740**	815	*0.91*	**744**	-	*0.92*	**700**	750	*0.92*	**755**	804	*0.93*
Combustion Blower + Controls		**84**	-	-	**85**	194	*0.44*	Note 1- Not measureable due to furnace control board error because the meter reverses polarity			**42**	124	*0.34*	**95**	192	*0.49*
* has customized separate power connection for blower for remote speed selection																
BPM Blower Motor using furnace controls																
Blower – High		**353**		*0.68*	**356**	530	*0.67*	**355**	-	*0.68*	**340**	500	*0.67*	**354**	529	*0.67*
Blower – Low + Combustion Blower		**134**		*0.58*	**136**	235	*0.58*	See Note 1			**100**	180	*0.55*	**155**	237	*0.56*
Standby		**10**		*0.42*	**11**	25	*0.43*	**11**	-	*0.45*	**10**	22	*0.48*	**9**	20	*0.45*

The key observations from the results in Table 5 are:

1. Combustion blowers and controls contribute significantly to the total power - roughly 85W for this furnace. Therefore, this must be accounted for this in any "total" power measurement that would be made with a plug-through meter. Depending on the thermostat and other control wiring it may be possible to have the furnace operate in such a mode that the two blowers can be isolated. For example, “fan on” mode on the thermostat should activate the main blower without turning on the heating or cooling and therefore not activating the combustion blower. One drawback to this approach is that the “fan on” mode may activate the blower in low speed rather than at normal heating or cooling. For the BPM blower, the controls slowly ramp up the blower speed to prevent blowing cold air. In an attempt to isolate the combustion blower power consumption, the furnace with the BPM blower was activated using the thermostat setpoint (as would be done for a thermostat with no ”fan on” switch or a furnace with these wires not connected) . The experiments reported here did not have gas connected to the furnace, so there was no heat and the blower only ever operated at its lowest speed when activated by the thermostat setpoint – thus defeating the purpose of the test. These controller complications will need to be well understood by technicians performing any laboratory or field testing for compliance or credit in California building energy codes.

2. Standby power is 8 to 10 W and the PFs are low - typically 0.35. Standby power can be a significant amount of energy over a year. 7000 hours of standby operation is 70 kWh for these furnaces.

3. Some meters measure a small power draw of about one Watt for the PSC blower alone when it is "off". Note that for the PSC measurements all the controls were bypassed so the results are just for the blower motor.

4. It appears that the Fluke 41 meter needs recalibrating. It consistently gave lower power values than the other meters with bigger differences at lower power factors (alternatively, all the other meters may be biased high, but this seems unlikely). Issues of periodic meter calibration need to be addressed in both laboratory and field testing protocols.

5. The WattMan reverses polarity. This is potentially hazardous, and for this furnace it put the furnace into "error" mode. The furnace controls detected the reversed polarity and would not turn on the combustion blower (but would allow operation of the main ECM blower).

6. The inexpensive simple plug through meters are acceptable (errors less than 5%) - except for the BPM at low speed (with its lower power factor) where they tend to under read the real power (Watts) by more than 10%. They have the advantage of being very simple and safe to use - so long as there is a plug on the furnace. For laboratory testing, a plug is attached to the input power wiring for the furnace. Without a plug on the furnace being field-tested it would be necessary to open a junction box and install a plug and socket.

One issue with any additional measurement requirements in California building energy codes is the affordability of doing the measurements. The following is a summary of typical costs for the meters evaluated in this study:

Kill-A-Watt: $39.95

Watts-Up Pro: $125

WattMan: unknown

Fluke: $1600

Clamp-ons Similar to Fluke: $170 (+/- 5% PF<0.5) to $600 (+/- 2%) depending on specification.

Elite Pro: $1000 + laptop computer

The plug-through devices are so low cost and so easy to use that they present no significant barrier to this measurement technique for field testing. Similarly, the cost of the Elite Pro is reasonable for a laboratory that would have multiple uses for this meter.

3.4. Air Flow

3.4.1. Laboratory testing

For laboratory measurements an in-line air flow meter can be used and connected to the furnace with the appropriate ducting. For the LBNL laboratory testing a Brandt flow nozzle (40 cm or 16 in. diameter) is used that has an accuracy of ±1%. Other air flow measurement techniques may be adopted such as those in AMCA 210/ASHRAE 51 (2007). The approach taken in the current draft of CSA 823 is to specify particular apparatus such as those in AMCA 210/ASHRAE 51 and ASHREAE 37. However, in the future it may be more acceptable to specify an accuracy and allow the use of other air flow measuring devices. Table 5 summarizes the proposed accuracy requirement in the current draft of CSA 823.

For standardized laboratory testing it is common practice to express the results at standard conditions that may not be the same as those during the test. The corrections are usually based on a combination of temperature and barometric pressure.

Table 1
Equipment Precision and Accuracy Requirements
(See Clause 5.10 and 6.3)

Item	Instrument accuracy	Instrument Precision*
Atmospheric pressure	± 0.3 kPa (±0.1 in Hg)	± 0.17 kPa (± 0.005 in Hg)
External static pressure	± 0.002 kPa (± 0.01 in w.c.)	± 0.001 kPa (± 0.005 in w.c.)
Airflow	± 2% of reading	
Electrical energy	± 1% of reading	± 0.5% of reading
Time	± 0.5 s/h	± 0.25 s/h
Volume	± 1% of measured volume	± 0.5% of measured volume
Inlet and outlet air temperatures†	± 0.1 °C (± 0.2°F)	± 0.06 °C (± 0.1°F)

**The smallest scale division on an instrument used shall not be more than twice the specified precision.*
†*The time constant at the inlet and outlet water temperature measurements shall be less than* 5 s.

Table 6. Precision and Accuracy requirements from draft CSA 823.

3.4.2. Field Testing

More so than for electricity measurements there is considerable extra difficulty in measuring air flows in field testing compared to laboratory testing. The current state of the art for field measurements consists of two test methods:

1. Pressure matching. In the pressure matching technique, the pressure difference between the supply plenum and the house is measured under normal operating conditions. An air tight blockage is then placed at the entrance to the blower compartment. A fan and flowmeter are attached to the blower compartment access. This fan is turned on and adjusted until the pressure difference between the supply plenum and the house matches that under normal operating conditions. The flow through the fan is then the air flow at operating conditions. In practice, the furnace blower is often turned on in order to match the pressures. The furnace blower must be turned on before the fan and flowmeter being used to make the measurement to ensure that the furnace blower is rotating the in the right direction.

2. Flow plate. In the flow plate technique, the filter is removed from the return system and replaced with a calibrated flow plate. The flow plate has fixed holes that act as a known air flow resistance. The pressure drop across the plate is measured and a calibration used to determine the air flow. To account for any changes in air flow due to the flow

plate being a different air flow resistance than the filter, the pressure difference between the supply plenum and the house is measured before the flow plate is installed and during flow plate use. Any differences in these pressures are used to correct the measured air flow with the flow plate in place to the air flow with the filter in place.

Both these test methods are referred to the current California Building Energy code Reference Appendixes sections RA 3.3, ASHRAE Standard 152, and ASTM E1554. There are no other currently available test methods for field testing of total system air flow that are better than these current methods. Therefore it is recommended that these be retained as the test methods in the Residential Appendixes.

4.0 Project Outcomes

Typical air leakage for forced air system cabinets is about 60 cfm or 5% of total system air flow. A reasonable target for an air leakage limit is 1.5% of the nominal air flow at a pressure difference of 0.5 in. water (125 Pa). This limit should be lowered in the future as manufacturers become better at constructing tight cabinets. Note that this limit is essentially equivalent to the current Florida and California requirements of 2% of nominal air flow at 1.0 in. water (250 Pa)

A new standard laboratory test for measuring HVAC cabinet air leakage has been developed as ASHRAE Standard 193P "Method of test for Determining the Air Leakage Rate of HVAC Equipment". This standard should be complete in early 2010. In addition, a simplified method has been provided to the Commission for use in Building Energy Standards until Standard 193 is publically available.

The laboratory testing of the energy performance of air handlers undertaken for this and previous studies has been used in the development of a new standard in conjunction with the Canadian General Standards Board's (CGSB): C823 "Performance Standard for air handlers in residential space conditioning systems". This standard should be complete in early 2010, at which time it should be used by reference in California Building Energy Standards – potentially as an alternative to field testing and measurement of blower performance.

5.0 Conclusions and Recommendations

Pressurization testing is robust and repeatable and is recommended for use in standardized test methods.

DeltaQ testing may be useful as a test procedure but it requires a large auxiliary test chamber and careful apparatus construction (e.g., non-moveable ducting) for successful testing.

Tracer gas testing is too unreliable for use as a standard test method.

ASHRAE Standard 193 should be used as a reference in building energy codes when it becomes available in 2010.

There are large variations in furnace-to-furnace air leakage and so the laboratory testing of these devices is recommended.

Typical air leakage for furnaces is a significant contributor to thermal distribution system air leakage: typically 60 cfm or about 5 % of total air flow.

Other duct system components (wyes, VAV boxes, splitter boxes)) have individual component leakage under 5 L/s (10 cfm) - however the use of multiple components in a duct system means that their contribution can be significant.

Attempts to seal component air leakage were successful indicating that it is possible to significantly reduce this component leakage.

The use of better cabinet materials (e.g., thicker gauge steel) and seals on access panels significantly reduces furnace cabinet air leakage.

It is recommended that the State of California require air leakage testing of forced air system cabinets (primarily furnace and air handlers) using ASHRAE Standard 193 with a maximum air leakage of 1.5% of maximum air flow for each unit.

For air handler power use it is recommended that field verification be required (as is currently implemented in the Residential ACM) for blower performance due to the dependence on duct system air flow resistance. Once the Canadian standard for blower performance is completed it could be referred to by California Building Energy Codes with appropriate performance specifications such as a minimum of 3 cfm/W. Builders and retrofitters of very energy efficient homes have reported that there is a difficulty meeting the tight duct specification in Title 24 for very small systems and is seen as a barrier for installing correctly sized systems. Specifying a maximum allowable air leakage for air handlers will improve this situation. In addition, it is recommended that the tight duct specification be altered so as to have fixed lower limit: e.g., 4% of total air flow or 20 cfm (10 L/s) – whichever is greater.

Lowering cabinet air leakage from 5% to 1.5% of system air flow will result in a reduction in the energy used for space conditioning by about 5%. The same reduction, or more, is also applied to peak heating and cooling loads. Another benefit is that tighter cabinets will allow the smaller HVAC systems installed in energy efficient homes to meet tightness specifications.

The cabinet air leakage reductions will mean that more homes are capable of achieving the California State Building Energy Code requirements for a tight duct system. Currently, it is reported by implementers that they can install tight duct systems – but the residual leakage of the furnace or air handler is so great that they cannot achieve the tightness levels specified for the tight duct credit and this is a barrier to duct sealing because the credit is not obtained. Therefore tighter cabinets will encourage more users to pursue duct sealing and the tight duct credit.

Providing uniform ratings and test methods for blower power will allow designers to specify and installers to select equipment that uses less blower power. Currently this information is not available. Annual energy savings that can be made using more efficient blowers for a typical three-and-a-half ton air conditioner are 45 kWh. Peak demand reductions are 50 W per system, or about 0.13 GW statewide if all blowers were replaced. The numbers improve significantly for duct systems that match manufacturers' rating points. For such systems, annual energy savings increase to 153 kWh, and peak demand reductions to 70W per system, or about 0.18 GW statewide when all blowers are replaced (Walker (2006)).

Both the air leakage and blower power reductions contribute to the California Long-Term Energy Efficiency Strategic Plan goals of net zero energy new homes by 2020, as well as the shorter term goals of 35% reductions relative to current state building energy codes. Eliminating furnace cabinet leakage would result in an approximate 5% reduction in heating and cooling energy use which would contribute significantly to these goals.

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7.0 Appendix A: Test Procedure for Air Handler Leakage

Introduction

This test procedure is to be used to determine the air leakage of air handlers to qualify for the low-leakage air Handler credit in the 2008 Residential ACM Manual, Section 3.12.5 Duct/Air Handler Leakage. This test procedure depressurizes the air handler and measures the air flow to maintain a static pressure of one inch of water (250 Pa).

Air Handler Test Apparatus and Accuracy Specifications

The air handler shall be depressurized using an air moving blower or equivalent device capable of controlling and maintaining air flow with the air handler depressurized relative to its surrounding of 1.0 in. water (250 Pa). The flow required to maintain this pressure difference shall be measured using an air flow measuring device capable of measuring volumetric air flow with an accuracy of 2 cfm (1 L/s) or 5% of measured flow, whichever is larger.

The pressure difference between the inside and outside of the air handler shall be measured using a meter with accuracy equal to or better than 0.004 in. water (1.0 Pa). The temperature of air flowing through the air flow meter shall be measured with an accuracy equal to or better than 2°F (1° C). Barometric pressures shall be measured with an accuracy equal to or better than 0.15 in. Hg (0.5 kPa).

Test procedure

Step 1. Background Air Leakage

The background air leakage is a measurement of air leakage of the test apparatus that is subtracted from the total air leakage with the air handler in place.

1.1 The inlet to the test apparatus shall be sealed and a static pressure tap shall be connected to the air handler at the air outlet. The meter used to record the pressure difference between the inside and outside of the test apparatus shall be connected to this pressure tap.

1.2 The device used to depressurize the air handler shall be turned on and adjusted to provide 1.0 in. water static pressure (with the inside depressurized relative to ambient conditions) between the inside and outside of the test apparatus.

1.3 The air flow through the measurement device shall be recorded (Q_{bg}) together with the temperature of air flowing through the device (T) and the barometric pressure (P_{baro}). Q_{bg} is the background air leakage. The measured temperature and barometric pressure shall be used to perform any calibration corrections required by the manufacturer of the air flow measurement device.

1.4 The background air leakage shall be converted to standard conditions using Equation 1 (for SI units) and 2 (for IP units).

$$Q_{bg,std} = Q_{bg}\sqrt{\frac{P_{baro}}{101.325}\frac{293}{(T+273)}} \quad (1)$$

$$Q_{bg,std} = Q_{bg}\sqrt{\frac{P_{baro}}{29.92}\frac{460}{(T+460)}} \quad (2)$$

Step 2. Test set up

The device used to depressurize the air handler shall be connected to the air inlet of the air handler together with the air flow measuring apparatus. All other air inlets, air outlets and condensate drain port(s), when present, shall be sealed during the test. A static pressure tap shall be connected to the air handler at the air outlet. The meter used to record the pressure difference between the inside and outside of the air handler shall be connected to this pressure tap.

Step 3. Depressurize air handler

The air moving device shall be turned on and adjusted until there is a pressure difference of 1.0 in. water (250 Pa) between the inside and outside of the air handler, with the inside of the equipment depressurized relative to ambient conditions.

Step 4. Record air flow

The air flow through the measurement device shall be recorded (Q_{meas}) together with the temperature of air flowing through the device (T) and the barometric pressure (P_{baro}). The measured temperature and barometric pressure shall be used to perform any calibration corrections required by the manufacturer of the air flow measurement device.

Step 5. Conversion to standard conditions

The air flow shall be converted to standard conditions using Equation 3 (for SI units) and 4 (for IP units).

$$Q_{meas,std} = Q_{meas}\sqrt{\frac{P_{baro}}{101.325}\frac{293}{(T+273)}} \quad (3)$$

$$Q_{meas,std} = Q_{meas}\sqrt{\frac{P_{baro}}{29.92}\frac{460}{(T+460)}} \quad (4)$$

Step 6. Subtract background leakage

The background air leakage shall be subtracted from $Q_{meas,std}$ to determine the air handler air leakage, Q_{leak}.

$$Q_{leak} = Q_{meas,std} - Q_{bg,std} \tag{5}$$

Step 7. Record Test Results

Record and report the following information:

Date of test.

Name of test organization.

A description of the equipment under test including manufacturer, make, model number, and serial number.

The air leakage test result, Q_{leak}.