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Distribution Effectiveness and Impacts on Equipment Sizing for Residential Thermal Distribution Systems

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Executive Summary

This report builds on and extends our previous efforts described in "Leakage Diagnostics, Sealant Longevity, Sizing and Technology Transfer in Residential Thermal Distribution Systems- CIEE Residential Thermal Distribution Systems Phase VI Final Report, December 1998". This report concentrates on a new area of work: the interaction between distribution system effectiveness and equipment sizing. This issue focuses on the ability of downsized equipment with a good distribution system to deliver the same cooling to conditioned space as a typical Heating, Ventilating and Air Conditioning (HVAC) system. The cooling of the conditioned space is evaluated by looking at the concept introduced in the previous phase of this study: “Tons At the Register" together with comfort issues, such as how quickly a house is cooled (“pulldown time”), and the distribution of cooling throughout the house.

The key outcomes of this study are:

- This investigation yielded a new duct leakage test called DeltaQ.
- The existing ASTM Standard (E1554) for measuring duct leakage has been rewritten and submitted to the ASTM standards review process.
- A draft ASTM standard for longevity testing of duct sealants was developed. A draft was submitted to ASTM subcommittee E06.41 for balloting and comment. The comments on the draft resulted in changes to the test method and apparatus. A new test apparatus was constructed with funding from the Department of Energy (DOE).
- Simulations of summer temperature pulldown time have shown that duct system improvements can be combined with equipment downsizing to save first cost, energy consumption, and peak power and still provide equivalent or superior comfort.
- Air conditioner name plate capacity ratings alone are a poor indicator of how much cooling will actually be delivered to the conditioned space. Duct system efficiency can have as large an impact on performance as variations in Seasonal Energy Efficiency Ratio (SEER).
- Installation of high SEER units can reduce energy consumption with no apparent drawbacks
- Duct efficiency calculations are included in the Low-Rise Residential Alternative Calculation Method Approval Manual for 1998 Energy Efficiency Standards for Low-Rise Residential Buildings” (CEC (1999)).
- Procedures for HVAC System Design and Installation (for Home Energy Raters) have been updated.
- Field testing has shown that standard flowhoods can be poor for measuring residential register flows.

Results from this study were used by the California Energy Commission (CEC) in the formation of the current Energy Efficiency Standards for Low-Rise Residential Buildings (CEC, (1998)), often referred to as Title 24.

Current information on ducts and thermal distribution research can be found at http://ducts.lbl.gov

Introduction

Previous studies (including earlier phases of this research project) have shown that losses from residential thermal distribution systems have significant energy and comfort implications. This study looks at the potential for improvement of thermal distribution systems and the possibility of reducing equipment size as a result. These distribution system and equipment interactions were examined through field testing and computer simulation. In addition, this report outlines our efforts to transfer the results of this research to the marketplace so as to reduce energy losses and improve thermal comfort. This study describes the results of efforts made during the Transitional Phase of this Residential Thermal Distribution Systems research. Results of earlier Phases were described in Walker et al. (1997 and 1998).

1. Duct Leakage Diagnostics

The objectives of this task were:

- **Improve duct leakage test methods.**
- **Update the American Society for Testing and Materials (ASTM) Standard E1554 – "Determining External Air Leakage of Air Distribution Systems by Fan Pressurization"**

Summary of duct leakage diagnostics in previous phases

In Phase V of this work we performed field evaluations of several diagnostic techniques for measuring duct leakage:

- House Pressure Test (HPT).
- Nulling Pressure Test (NPT).
- Duct and house pressurization with separate supply and return leakage.
- Duct only pressurization with combined return and supply leakage.
- Tracer gas.

These tests were evaluated in terms of ease of use, time requirements and the bias and precision errors associated with each test by using the tests in several houses. The results of the testing indicated that none of these methods were ideal (hence our continuing work on improving duct leakage diagnostics). However, for screening of low leakage levels for compliance testing the duct leakage diagnostic of choice is the fan pressurization test of total duct leakage (test 4). The reasons for this are:

- Robustness. The fan pressurization test has almost no restrictions on the type of system it can be used on, or the weather conditions during the test.
- Repeatability. Combining the results of both the phase V and VI reports together with the field experience of other users showed that the repeatability of the pressurization testing was found to be very good.
- Precision. The uncertainty in leakage flow will be small if the allowable leakage is set to a low number because the uncertainties for the pressurization test scale with the amount of leakage.
- Simplicity. It is easy to interpret the results of fan pressurization without having to perform many (or any – with the appropriate hardware) calculations. This allows the work crew to evaluate the ducts during the test and also allows the work crew to ensure that the test has been performed properly because they can see if the results make any sense.
- Familiarity. Work crews that have performed envelope leakage tests are familiar with the test method for ducts, because envelope testing uses a similar apparatus and calculation/interpretation methods.

The biggest drawback with this test is the requirement of covering all the registers which can be time consuming. In addition, this precision of this test is reduced at higher leakage levels that might be found by home energy raters in existing construction, rather than the low leakage levels required in compliance testing. Because this test measures the total leakage and not just the leakage to outside it will overestimate the leakage required for energy loss estimates, however, from a compliance testing point of view, this error is in the right direction because it means that the true losses will be less than those indicated by the test. In other words, a system whose total leakage passes a leakage specification is guaranteed to have the leakage to outside be less than the specification.

In Phase VI we extended the duct leakage measurements to include separate measurements of the boot and cabinet leakage because these were thought to be two main leakage sites. The measurement results confirmed this idea: combining these two leakage sites together accounted for about three quarters of all duct leakage. The average leakage to the outside was about 25 cfm for the boots and about 34 cfm for the cabinets.

A New Duct Leakage Test: DeltaQ

In order to find a duct leakage test that is better than those discussed above, a duct leakage measurement workshop was held as part of the ASHRAE Standard 152P (ASHRAE, 1999) committee meetings in January 1999. We have prepared a summary of this workshop, and it is included as Appendix 1. In addition, we discussed potential innovative measurement techniques with other researchers throughout the US and Canada.

The result of these discussions is a new technique for measuring duct leakage that we have evaluated using a pilot study of local homes. This new technique is called the DeltaQ test because it measures changes in flow (Q) caused by distribution system operation. This new test method has several features that give it the potential for success:

- It has simple equipment requirements. Only a blower door and some pressure sensors are required to perform the test. The blower door is a common item that most building diagnosticians already have and are familiar with its operating principles. Some existing tests require less common equipment, for example specialized combined fan/flowmeters for pressurization tests, or tracer gas analysis equipment.
- It directly measures the value that we want from the test: the leakage to outside at operating conditions of the supply and return separately. Other existing tests require conversion from measured pressures to operating pressures, or they require complex balancing of house and duct pressures to obtain leakage to outside rather than total duct leakage.
- It is quick. There is no requirement for blocking off all the registers or blocking between the supply and return parts of the system.
- It is robust. Our field testing has shown that the DeltaQ test is not as sensitive to wind induced envelope pressure fluctuations as the House Pressure Test, or Nulling Pressure Test.
- It does not have the detailed assumptions (that lead to additional uncertainties) about the house envelope that the House Pressure Test requires.

The DeltaQ test works by using a blower door to maintain the same pressure across the building envelope with the duct system fan on and off. The flow with the system on and off is measured over a range of envelope pressures. This results in pairs of flow data (one with the system fan on and one with the system fan off) at several pressures. As the blower door pressurizes (or depressurizes) the house relative to outside, the pressures in the ducts will also change relative to outside by the same amount. Because the pressure across the leak changes, the flow through the leaks changes and this change in leakage flow appears as a change in envelope flow through the blower door. In addition the operating pressures in the ducts when under normal operating conditions are also measured. These operating pressures are measured at the plenums because this gives the biggest and most repeatable pressure signal and avoids the uncertainties of register pressure measurements. Combining the measured system pressures and the pairs of blower door flow data together with the algebraic analysis of the changes in duct leakage flow allows the calculation of the supply and return leakage coefficients and pressure exponents. Appendix 2 gives more details of the derivation and application of the test method.

So far, only six houses have been DeltaQ tested and more houses will be tested in the near future. Of these six houses, one test was at low wind conditions and gave results that closely matched other measurement techniques. The second test was on a windy day, but still managed to give reasonable results based on visual observation of the duct system, i.e., it showed that the ducts were not very leaky. This is a significant result because other tests that use envelope pressures (HPT, NPT and duct and house pressurization) have not given reasonable results under windy conditions. The third house was tested on a very windy day (wind speeds > 20 m.p.h. and highly variable) and the DeltaQ test did not give satisfactory results under these extreme conditions. These three tests have shown that the DeltaQ test is more robust than most of the existing tests but still fails at the very high wind speeds. Under extremely windy conditions the only test that can be used on ducts is the duct pressurization test because it does not require envelop pressure measurement or measured flow through a fan flowmeter between the house and outside. Future work will apply the DeltaQ test to more houses and include repeatability studies.

ASTM Duct Leakage Standard (E1554)

The existing test procedure in E1554 is called the blower door subtraction method and is no longer used by many researchers due to the poor results obtained from the test. This standard is currently due to be revised by ASTM so we have prepared a revision of E1554 (ASTM (1999)) that incorporates the DeltaQ test together with the combined house and duct fan pressurization test from proposed ASHRAE 152P. In addition to revising the standard, we have also been performing administrative tasks such as attending ASTM meetings and collaborating with ASTM staff to produce this revised standard. This revision of E1554 will be evaluated by an ASTM Task Group in October 1999. After initial review by the Task Group, it will take a year or two for the revised draft to become a test method. This time allows us and other potential users to evaluate the revised procedures in more homes. At the ASHRAE 152P meetings in June 1999 the ASHRAE 152P committee members were given copies of the test procedure and asked to use it and report back to us in order that we can build up a consensus of experience with this test method in as broad a range of homes and test conditions as possible.

Duct leakage diagnostic outcomes:

- **This investigation yielded a new duct leakage test called DeltaQ.**
- **The existing ASTM Standard (E1554) for measuring duct leakage has been rewritten and submitted to the ASTM standards review process.**

2. Duct Sealants and Longevity Testing

The objective of this task was to:

- **Develop and introduce a draft ASTM standard for longevity testing of duct sealants**

The development of the longevity test method and preliminary results have been discussed in previous phases (Walker et al. 1997 and 1998). The final results and details of the experiment were given in "Can Duct Tape Take the Heat" - LBNL report # 41434 and its companion Home Energy Article (Home Energy, Vol. 15, No.4, pp. 14-19. http://www.homeenergy.org/898ductape.title.html).

The results of work in previous phases of this study have been included in California's Residential Energy Code (usually referred to as Title 24). In the Alternative Calculations Manual of Title 24 no cloth backed rubber adhesive duct tape is allowed as a duct sealant on systems obtaining credit for energy efficient duct systems. This has caused some consternation on the part of HVAC installers and duct tape distributors; however, we have been able to show these concerned parties that the test results are real and that there are viable alternatives. In addition, some of the duct systems that were tested for phase VI of this study, and other systems we have observed over the last six months have been sealed in accordance with Title 24 requirements and our leakage measurements have shown them to be well sealed systems.

The longevity test method (ASTM (1999b)) was prepared in ASTM standard format and submitted to ASTM Subcommittee E06.41 for consideration. The ballot results had only one significant technical comment that the high temperatures were too low because some attics can be at a higher temperature than those used in the test (150 °F surface temperatures). This comment says that in order for the longevity test to be "accelerated" the attic temperatures should be at least as high as measured peak temperatures and possibly higher. However, the evidence for extreme attic temperatures higher than 150°F is poor. A literature search of attic temperature studies was undertaken to find evidence of higher measured temperatures. Much of the existing literature does not address peak or extreme values because the studies were interested in estimating energy savings where longer time average values are needed. However, a few studies were found that gave explicit attic peak temperature information, and those with the highest reported peaks are discussed here. Some of the following studies tested several houses but we discuss the results from the hottest attic only. Carlson et al. (1992) measured peak attic air temperatures of 155°F. Parker et al. (1997) measured attic temperatures of 134°F in a house in Florida, however, this was an average of the hottest 2.5% of the summer hours, so peak temperatures would be expected to be higher. The tests discussed in Phase VI of the current study had a peak attic air temperature of 151°F for a house in Sacramento.

An additional parameter that changes duct temperatures is the radiant exchange between the ducts and the roof deck surfaces that are hotter than attic air under peak conditions. For example, Wu (1989) measured attic floor temperatures 7 °F hotter than the attic air. The upper exterior surfaces of ducts in attics are heated by a similar amount. It is important to consider this increase in surface temperature due to radiation because duct sealants are generally applied from the outside and will experience these elevated exterior surface temperatures. Combining the existing peak temperature field data with the radiation effect results in a temperature of about 160°F being a reasonable target temperature.

Another point of view is to look at the duct temperatures experienced by heating systems which can be higher than those for cooling systems. Field studies have found that many furnaces are operating on their high-limit switches – usually set at about 200°F. The Uniform Mechanical Code (ICBO (1994)) has a limit of 250°F (121°C) for furnace and duct heater controls. The Canadian Natural Gas Installation Code (CGA (1995)) gives the same limit of 250°F (121°C) for forced air systems, but includes a higher limit of 350°F (175°C) for gravity furnaces. This indicates that the 160°F high temperature limit from the peak attic temperatures would be too conservative for heating systems. However, the furnace high-limit temperatures will not be used because there is an additional high temperature limit constraint imposed by duct tape manufacturers of 200°F for some of their products. A reasonable compromise is to be half way between the upper limit for cooling (160°F) and the limit set for tape (200°F) resulting in an upper temperature of 180°F. This compromise was chosen to be far enough away from the upper limit for tape that we can be reasonably sure that the tape does not exceed this limit during testing because the temperature control in the experimental apparatus has some uncertainty.

The major difference between the heating and cooling values is that the extreme temperatures for cooling ducts in hot attics occur with the system off and the heating ducts have their extreme with the system running, so the heating limit may be a more realistic scenario. In addition, the temperature gradient across the duct (hot inside air, cool surroundings) is the correct situation for the heating duct case. However, it is the explicit duct surface temperatures that are the temperatures experienced by the duct sealant and so not too much importance should be placed on the direction of heat transfer through the duct walls.

Based on the above temperature limit changes and other feedback from ASTM members and other interested parties, in addition to our own research, we have begun development work on a revised longevity test apparatus funded by the Department of Energy. This new apparatus will allow us to test different procedures for evaluating longevity. The procedures include:

- The existing procedure of alternating between hot (150°F) and cold (0°F) air flows with a pressure difference across the seal.
- Changing the temperatures in the alternating temperature test to be more extreme (180°F).
- Splitting the test into a hot test and a cold test. The hot test will vary the temperatures in the range of 70°F (close to room temperature) to 180°F. The cold test will have the range from 70°F down to 0°F. This narrower temperature range will allow for more rapid cycling and greater temperature extremes (e.g., we could possibly go lower than 0°F). For both the hot and cold test there will be a pressure difference maintained across the leakage sites.
- Having no cycling of temperature and maintaining a steady hot (180°F) or steady cold (0°F) temperature. Unlike the previous baking tests there will be airflow through these samples and pressure differences across the leaks.

The leakage testing will be the same as in the previous apparatus. Periodically the samples will be removed for individual leakage testing of leakage flows at 25 Pa. The leakage of the samples will be measured before any sealing and immediately after sealing before installation in the test apparatus. The “failure” of a sample will be determined the same as in the previous study, by evaluating the leakage flow as a fraction of the unsealed flow. The failure level is fixed at 10% of the unsealed leakage. The 10% level was chosen by examining test results to determine the point beyond which failure can be rapid and difficult to measure. .This 10% leakage also corresponds to empirical estimates of “unacceptable” leakage for an individual connection.

Duct Sealant longevity testing outcome:

- **A draft ASTM standard for longevity testing of duct sealants was developed. A draft was submitted to ASTM subcommittee E06.41 for balloting and comment. The comments on the draft resulted in changes to the test method and apparatus.**
- **A new test apparatus was constructed with funding from the Department of Energy (DOE) and will be used to evaluate new sealants.**

3. Duct System Interactions with System Sizing

The objectives of this task were:

- **Measure the performance of residential cooling equipment and associated distribution systems.**
- **Compare the REGCAP simulation model to the measured field data.**

In this study the duct system interactions with system sizing were examined using both computer simulations and measured data. The measured data were used to examine field performance of cooling systems and to evaluate and validate the computer simulations but were not used to tune any model coefficients so that the model retains its general applicability. The following sections discuss the field measurements, computer simulations and comparisons between the two.

Field measurements

The cooling system performance was measured in six test houses. Each house was tested in several configurations in order to estimate the effect of duct systems on the capacity, energy performance and comfort. The previous Phase (Walker et al. (1999)) reported the preliminary results considering sensible Tons At the Register (TAR) (“delivered capacity”) and capacities only. The current study looks in more detail at pulldown tests and equipment performance for both latent and sensible cases. The pulldown tests were evaluated by determining the pulldown time (the amount of time to cool down a house) for different parts of the house: at the thermostat, the master bedroom and the kitchen. In addition the temperature in each location at the end of pulldown as indicated by the thermostat was investigated. The differences between these locations indicate the relative comfort for the occupants. e.g., in the summer, a house where the temperature is much higher for the master bedroom when the system turns off (end of pulldown at the thermostat) will not be comfortable when the occupants go to bed. This is a common complaint about air conditioning systems and was specifically mentioned by the people who lived in the occupied house used for this study.

The field measurements included diagnostics to determine building and system characteristics and continuous monitoring over several days to determine pulldown system performance. Six houses were monitored for this project: 2 houses in Palm Springs, CA (sites 1 and 2), one house in Mountain View, CA (site 3), two houses in Sacramento, CA (sites 4 and 5), and a single house in Cedar Park, TX (site 6). All of the houses were new and unoccupied, except for the Mountain View house that had been occupied for less than a month at the beginning of our tests. The houses were tested in their “as found” configuration, then with the duct systems sealed. Houses that did not have very much "as found" duct leakage had holes added.

In two houses, the cooling equipment was replaced with Energy Star® equipment (greater than SEER 13.0). The original cooling equipment in each house was rated at the federal minimum SEER 10. In Sacramento (site 4), just the outside compressor unit and the control system were changed. In the Texas house (site 6), the indoor coil, fan and cabinet (and electric heating system) were also replaced. Details of the HVAC systems and house construction can be found in the report on the previous phase of this work in Walker et al. (1999).

Table 1 summarizes the most significant diagnostic test results for the thermal distribution system and equipment for the six test houses. The air handler flows for these systems were higher than has been found in previous studies (e.g., Blasnik at el. 1996) that suggested that most systems typically had about 15% less than the 400 cfm/ton recommended by manufacturers. In several cases the air handler flow was considerably higher than the 400 cfm/ton benchmark, particularly at site 4 with almost 550 cfm/ton. This high flowrate will limit the ability of the system to handle latent loads. However, site 4 is located in Sacramento, CA and does not have a high latent load, so these high flows are probably acceptable. The leakage expressed as a fraction of fan flow is lower than has been found in previous studies, indicating that these duct systems were better than average installations; in fact, they were some of the least leaky systems we have tested. The exception to this was surprisingly Site 3, where most of the duct system was in interior partition walls or dropped soffits between the first and second floor, with none of the duct system in the attic. A detailed examination of the ducts at Site 3 showed that much of the leakage was at the plenum to duct connections that were in the garage. In addition, the soffits and partition walls were not air sealed with respect to the garage or the attic so that air leaking from the ducts did not leak into the conditioned space but was allowed to escape to outside. This result reinforces the requirement of field testing duct systems for leakage because this system that looks like it is inside conditioned space in engineering drawings and initial visual inspection leaks considerably to outside.

The refrigerant charge was determined gravimetrically each system and compared to the correct system charge. The correct charge was determined by performing superheat tests and tuning the quantity of refrigerant to produce the required superheat. Table 1 also shows that these systems were close to having the correct system refrigerant charge, except for sites 2 and 4, where the systems showed the undercharging that was typical of that found in other studies. Site 2 was the only site where the system charge was an extreme concern because at only 70% of required charge, this system is undercharged to the point where significant equipment damage could occur.

Table 1. Diagnostic Test Results

Site	**Nominal AC Capacity [Tons]**	**Air Handler Flow [CFM/Ton]**	**Supply Leakage Fraction [%]**	**Return Leakage Fraction [%]**	**% of Correct Refrigerant Charge [%]**
1	5	375	4%	2%	98%
2	5	379	4%	1%	70%
3	3.5	491	8%	19%	101%
4	2	547	5%	3%	85%
5	2.5	467	6%	4%	95%
6	3	501	4%	5%	91%

Part of this study examines the possibility of resizing systems in order to reduce HVAC system first cost and peak energy consumption. To provide background information for answering this question, Table 2 contains a comparison of system capacities. For each site, the ACCA Manual J (1986) sensible load was calculated using the measured house dimensions and construction details. This was compared to data from the manufacturer (nameplate capacity), from the ARI (1999) ratings and the measured sensible TAR. The measured TAR were the quasi-steady-state values obtained after the equipment had been operating for some time so as not to include transient effects that are not part of the other ratings. Table 2 shows that the nameplate capacities far exceed the requirements of the ACCA Manual J calculations indicating significant oversizing. The ARI and maximum sensible ratings diminish the oversizing effect and reinforce the overrating in the nameplate capacities. For Site 6 the maximum sensible capacity is actually very close to the Manual J load estimate and this is probably the correct size air conditioner for this house. The measured TAR is even closer to the Manual J estimates and at Site 6 the TAR is less than the Manual J load estimate. The variation in TAR illustrates the impact of the system performance in converting from what is purchased by the homeowner or contractor (nameplate capacity) and is actually delivered to the conditioned space. At sites 1 and 2 the nameplate capacity is the same but the delivered TAR is a ton less for Site 2. Site 3, with a 1.5 ton less nameplate capacity system has almost the same TAR as Site 2. Sites 3 and 4 have TAR that are almost the same as the Maximum Sensible Capacity of the equipment, but the other sites have considerable lower TAR than this maximum. Site 1 is the only site with a significantly higher TAR than the Manual J estimate. Overall, the results shown in Table 2 illustrate that nameplate capacity is a poor way to evaluate the capacity of the equipment (compared to Maximum Sensible Capacity) and the system as a whole (TAR). In addition, the apparently gross oversizing of nameplate capacity compared to Manual J is offset by lower actual equipment performance and thermal distribution system losses.

Table 2. System Capacity Comparisons

Site	Manual J Sensible Load[1] [Tons]	Nameplate Capacity [Tons]	ARI Capacity [Tons]	Maximum Sensible Capacity [Tons]	Tons at the Register [Tons]
1	2.25	5	4.4	3.98	3.6
2	2.18	5	4.4	3.61	2.7
3	1.63	3.5	3.3	2.56	2.5
4	1.02	2	1.9	1.56	1.5
5	1.45	2.5	2.4	2.14	1.7
6	2.28	3	2.9	2.34	1.8

[1]What the capacity should be assuming no duct leakage, 400 CFM/ton airflow, perfect refrigerant charge, no additional safety factor.

Continuous Monitoring

The continuous monitoring used computer based data acquisition systems to store data approximately every 10 seconds. The monitored parameters were:

- Temperatures: at each register, in each room, outdoors, attic, garage, return plenum and supply plenum. The supply plenum temperatures were measured at four points in the plenum to account for spatial variation in plenum temperatures.
- Weather: wind speed, wind direction, total solar radiation and diffuse solar radiation.
- Humidity: outside, supply air, return air and attic (or garage if system located in garage).
- Energy Consumption: Compressor unit (including fan) and distribution fan power.

The measured system temperatures and relative humidities were used to calculate the energy flow for each register (and therefore the total for the system) and the energy change of the air stream at the heat exchanger at each time step.

An overview of all the test data, averaged by a consistent set of test conditions (i.e. amount of duct leakage, refrigerant charge, type of air conditioning unit) appears in Appendix 3. The performance metrics that were calculated are listed in Table 3. Each metric has a sensible and a latent component (reported as a sensible and total) and for all of the metrics except the pulldown time, the value is reported from an average of a minute of data at 5, 30, and 60 minutes from when the pulldown test began. This range of times for evaluation purposes was used due to the large transient changes in system performance between the beginning of a cycle and the quasi-steady-state operation reached later in the pulldown test.

Pulldown time is often very different for each of the three reported locations: thermostat, kitchen, master bedroom. For example, Site 3 was a two-story house with very poor distribution, particularly upstairs to the master bedroom. The upper floor of this house had had a significantly increased load due to several skylights in the space as well as an inadequate return system (there was no return from upstairs). At Site 3 the pulldown time at the thermostat was less than half an hour, but the upstairs took another hour and a half to pulldown to the same temperature. When the thermostat had reached the pulldown temperature it was 3°C (6°F) hotter upstairs than downstairs.

TAR is determined form the airflows and enthalpy at each register. The indivudual register values are combined into a single value for the system. TAR is often negative when the air conditioner first comes on because the hot air inside the duct system is blown into the house. This rapid initial change in temperature (a rapid initial increase followed by a gradual cooling further into the cycle) made analysis of the initial 5 minutes of the cycle difficult because of the response time of the sensors. The response time of the temperature sensors is rapid enough that any time response errors are insignificant. However, the slower response of the relative humidity (RH) sensors increases the uncertainty in the transient latent (and therefore total) TAR estimates. The time lag of the RH sensor compared to the temperature sensor means that the measured RHs are higher than they should be (assuming a reduction in moisture content of the air due to condensation on the coil) which leads to an underprediction of the latent TAR. Alternatively, if there is no moisture removal by the air conditioner, the RH of the air at the register should rise as the temperature drops. The longer time response of the RH sensors means that they read artificially low and this overpredicts the TAR, i.e., it gives the appearance of moisture removal without there being any. Because this time response issue can drive the results high or low depending on the particular operating conditions, it is not possible to estimate a generalized effect that would apply to all the measurements (i.e. a bias), instead it simply adds to the uncertainty of the latent TAR calculations during the start of each cycle. Because of these uncertainties, most of the comparisons made in later sections of this report are based on the 30 minute value when the air conditioner is operating much closer to steady state.

Air Conditioner Capacity is estimated by measuring enthalpy change and air flow through over the evaporator coils. It is a useful way of examining the effect of low evaporator air flow or incorrect refrigerant charge without confounding the impacts of a leaky duct system. It suffers the same sensor response limitations during the initial transient at the start of a cycle as TAR.

Air conditioner COP is a measure of efficiency of an air conditioner, and it is typically around 2-3 for a residential unit. It is estimated from the measured capacity and electricity consumption. Unlike the COPs presented by the manufacturer, the COPs reported here include the energy (and heat generation) of the air handler fan.

System COP is the most inclusive performance measure, and it is a simple ratio of the cooling energy delivered to the conditioned space (TAR) divided by the power consumption of the air conditioner and fan. System COP is affected by changes in the air conditioner capacity as well as any losses/gains in the distribution system.

Delivery efficiency is a simple ratio of the energy of the air that comes out of the registers divided by the energy of the air in the supply plenum. It is a measure of losses that occur in the duct system. The five minute delivery efficiency is almost always higher than the 30 and 60 minute delivery efficiency. This is because after five minutes, the air in the ducts has not cooled down very much and conduction losses (which are proportional to the temperature difference between the ducts and the air around the ducts) are low. By the time the system has reached steady state, the conduction losses have increased and the delivery efficiency drops slightly.

Table 3. Performance Metrics

Pulldown Time	Time that it takes for a zone to reach 24°C. Three pulldown times are reported: for the thermostat (how the house would actually respond), kitchen, and master bedroom. A wide disparity between these times indicates an inadequate distribution system.
Tons At the Register (TAR)	Amount of energy delivered to the space
Air Conditioner Capacity	Capacity of the air conditioner calculated from temperatures and relative humidities measured in the supply and return plenums

Air Conditioner Coefficient of Performance (COP)	Air conditioner capacity divided by power consumed by air conditioner, **including fan energy**
System COP	Tons at the register divided by power consumed by air conditioner, **including fan energy**
Delivery Efficiency	Tons at the register divided by air conditioner capacity

Although these metrics are all intuitive and useful ways to understand cooling system performance, they have limited utility for comparing houses or even for comparing different conditions (amounts of duct leakage, refrigerant charge, etc.) at the same house. All are very strongly influenced by many external factors (in addition to those already discussed such as duct leakage, evaporator airflow, refrigerant charge level, etc.), most notably indoor temperature and humidity, outdoor temperature and conditions, and attic conditions. The tables in Appendix 3 list the average of these parameters at each of the conditions. Because only a few days of measurements were performed at each site condition, there are often large variations associated with each of the mean values for the performance parameters. Also, in some cases only a single day of data was taken, resulting in a single data point being used as the "average". Because of these limitations, the measured results often do not reveal the expected results and make the data difficult to interpret.

For these reasons, this report concentrates on comparing individual sites where conditions are similar. Because weather conditions varied widely during the test period, there were few days that were identical to each other. Limiting comparisons to similar weather conditions means that the comparisons discussed below are often based on a single pulldown test at each condition and should be interpreted with some caution.

Site 1: At this site, there were two similar days of weather that allow us to examine the impact of sealing the ducts. Both the sensible and the total delivery efficiency improved 8 percentage points from 81% to 89%, about a 10% increase. Similarly the system COP improved, but only by 8% (total) and 3%(sensible). The latent improvement may be an overstatement because of uncertainties in the RH measurements, even though the air conditioner had been running for 30 minutes in each case and thus the relative humidity should not have been changing very rapidly. The relatively small improvement in system performance is an indication that inefficiencies of this large (5 ton) air conditioner tended to dominate the system losses, rather than the distribution system losses.

Site 2: Although there were four conditions at this site (as found duct leakage and low refrigerant charge, leaks added and low charge, leaks added and correct charge, ducts sealed and correct charge), only two days had similar enough weather conditions to compare them. Unfortunately, two things changed between the days that had comparable weather conditions: a very low level of refrigerant charge was corrected, which would tend to improve system performance and 212 cfm (12% of fan flow) of leakage was added to the duct system, which would tend to diminish system performance. The added leakage was split almost evenly, with 97 cfm (5.5% of fan flow) added to supply side and 115 cfm (6.5% of fan flow) added to the return side. Paradoxically, correcting the very low refrigerant charge level appeared to very slightly diminish the system performance. However, a close examination of the weather data indicated that, although the outdoor temperatures were very similar, the enthalpy of the outdoor air was significantly greater when the charge was lower which made the unit appear to perform better. Comparisons of delivery efficiency showed a reduction of about 10% due to the added leaks.

Site 3. Correcting a slightly low charge level resulted in an increase in air conditioner capacity of about 6% and an improvement in COP of about 8%. Sealing 200 cfm (14% of fan flow) of duct leakage improved the delivery efficiency by 11%. At this site, the ducts leaked into the garage (mostly outside) and into an interstitial space between the first and second floor that was thermally inside, but outside of the pressure boundary. These effects overall combined to improve the system COP by 17%.

Site 4. Many of the performance metrics are harder to interpret at this site because the new Energy Star™ units' energy consumption varied with time (probably due to a variable speed compressor). The air handler capacity of the Energy Star™ unit improved by 5% and the sensible COP improved by 25%. This result suggests that higher efficiency units may improve sensible cooling at the expense of latent cooling; however the uncertainty of RH measurements means that this result requires additional verification. The delivery efficiency dropped by about 4%

after changing the equipment. This drop might have been due to small variations in outdoor and attic temperatures, however, it may have also been caused by the lower supply plenum temperatures (and hence higher temperature differences for conduction losses) of the new unit. Adding 107 cfm (9% of fan flow) of leakage at Site 4 reduced the delivery efficiency by 6%. The extra leakage was greater on the supply side 75 cfm (6% of fan flow) vs. 32 cfm (3% of fan flow) for the return side.

Site 5. Sealing 92 cfm of leakage (7% of fan flow) caused insignificant improvements of the delivery efficiency (between 1 and 2 percent). The overall COP was improved by 3%, however (as with other sites) the uncertainty in the RH and plenum temperature measurements means that this change is about the same as the uncertainty in the measurements and cannot be interpreted as a significant change.

Site 6. The combination of highly variable weather and some data collection problems at site six means that there were no results suitable for comparison. A more detailed analysis of the measured data will be required if any conclusions are to be drawn from the Site 6 measurements.

Computer Simulations

The details of an earlier version of the simulation model (called REGCAP – short for REGister CAPacity) have been given previously by Walker et al. (1998 and 1998b). A flowchart for the simulation program is shown in Appendix 4. The changes made to improve the simulation model for this study are discussed below. This improved model was compared to measured results for validation purposes. In the previous work, the simulations were used to show how pulldown time changed with duct system performance, different weather conditions (a typical design day and a peak day) and with system capacity. The simulations were able to show several key results:

- A good duct system allowed the capacity of the equipment to be reduced by about one quarter: from four tons to three tons nameplate capacity.
- If system nameplate capacity is unchanged, either improving duct systems (to have little leakage) and correctly installing the equipment, or moving the ducts inside results in significant pulldown performance improvements. In these cases pulldown times were reduced by more than an hour and initial tons at the register were approximately doubled.
- The model results also showed the wide range of pulldown times for different duct systems.

Simulation Model Improvements

In our continuing efforts to develop models of HVAC system performance, the model used in the last Phase has been upgraded:

- It includes additional air flow paths through duct leaks when the system is not operating.
- It has a simple moisture balance for use in latent load and equipment capacity calculations.
- Solar load and thermal mass calculations for building load have been improved.
- A simple thermostat model and the ability to make calculations at small timesteps allowed the model to be used for examining cyclic effects.
- Improved equipment modeling accounts for changes in capacity and energy consumption with outdoor weather conditions, fan flow, system charge and indoor air conditions.

The equipment model used to predict the capacity of the air conditioners for the REGCAP simulation is an empirical model developed by John Proctor (Proctor (1999)). This model is the only available model that accounts for refrigerant charge level and is sufficiently general for use in this project. Proctor has used this model in much of his research (see Proctor (1997), (1998) and (1998b)) and continues to update it as he collects new data. Currently, the portion of the model that accounts for deviation from recommended refrigerant charge is taken directly from Rodriguez et al. (1995) and the rest of the model is based on Proctor Engineering Group fieldwork in about one hundred houses.

The model requires the following inputs: nominal (nameplate), capacity, ARI capacity, air flow, outside temperature, indoor (return plenum) enthalpy, refrigerant charge level, and expansion valve type (capillary tube/orifice or TXV (thermostatic expansion valve)). The model predicts sensible capacity and, with the assumption of a sensible heat ratio for the unit, latent and/or total capacity can also be predicted. The comparison of the measured capacities at the six houses (8 air conditioners) in this study indicate that the model overpredicts capacity by about 10%. There is no obvious reason for this consistent deviation from Proctor's data, but a possible reason is that most of the Proctor's verification of the model occurred in very dry climates, rather than the more humid weather that we encountered during the field testing.

Extension of previous simulations

The improved model was used to reexamine the pulldown simulations performed in the previous part of this study. In these simulations, eight different thermal distribution systems are used in the same house for the same weather conditions. Table 4 lists the simulation cases that were examined here. The BASE case is typical of new construction in California. The POOR system represents what is often found at the worst end of the spectrum in existing homes. The BEST system is what could reasonably be installed in new California houses using existing technologies and careful duct and equipment installation to manufacturers' specifications. The BEST RESIZED system looks at the possibility of reducing the equipment capacity using the best duct system. The INTERIOR system examines the gains to be had if duct systems are moved out of the attic and into conditioned space. The INTERIOR RESIZED system examines the system performance if reduced capacity equipment is used together with interior ducts. Lastly, the IDEAL system is an interior duct system that has been installed as well as possible. The IDEAL OVERSIZED simulations were included to examine the difference in pulldown if the IDEAL system were sized using current sizing methods (i.e., still 4 tons).

The difference between the simulation cases listed in Table 4 and those done previously is that the flow used for the "correct flow" cases is 400 cfm instead of 425 cfm. This minor change was done because the equipment used in the equipment model was rated at this flowrate. The large range for pulldown results are illustrated in Figure 1, with each simulation starting at the same time. The better systems were able to pulldown the house in a reasonably short time (under three hours) but the poor systems took over six hours. The longer pulldown times mean that the house would not be comfortable for occupants returning in the afternoon. For example, the house with the POOR system is still not pulled down at 8:00 p.m. For the occupants this would be unacceptable and a better question to ask is: At what time would an occupant have to turn on the air conditioning in order to have the house comfortable upon returning home in the afternoon at 5:00 p.m?

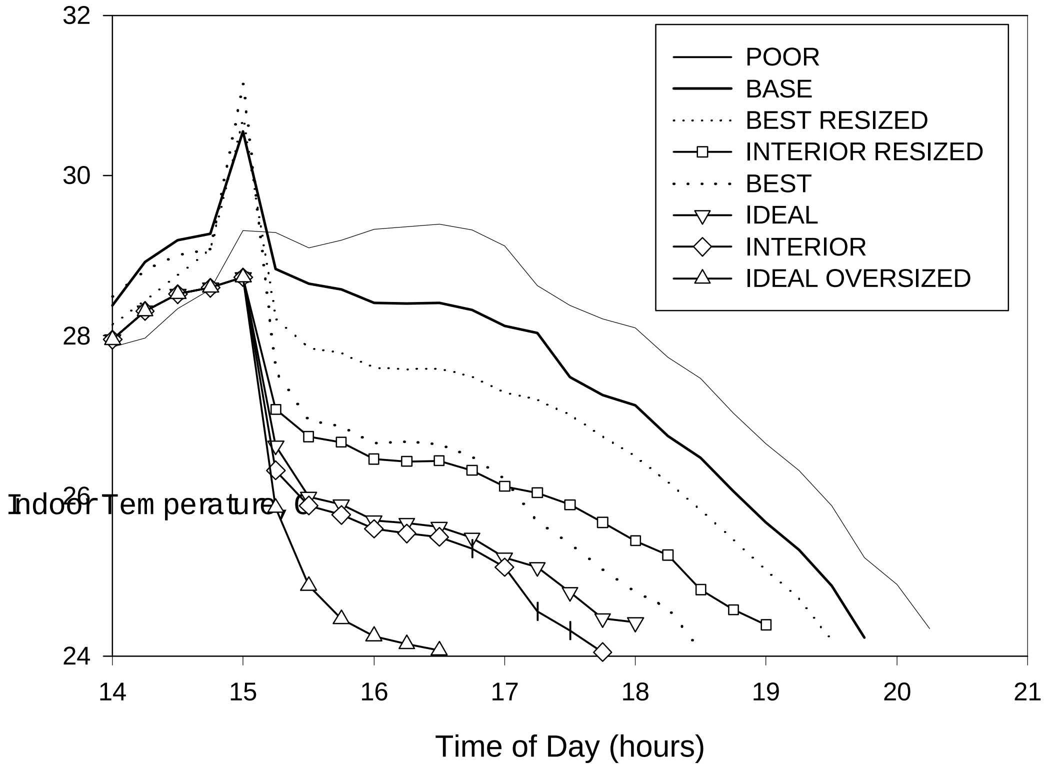

Figure 1. Simulations of Pulldowns from 3:00 p.m. on a Sacramento Design Day.

Table 4. List of REGCAP Simulation Cases					
	System Charge [%]	**Air Handler Flow [CFM/Ton]**	**Duct Leakage Fraction [%]**	**Duct and Equipment Location**	**Rated Capacity [Tons]**
BASE	85	345	11	Attic	4
POOR	70	345	30	Attic	4
BEST	100	400	3	Attic	4
BEST RESIZED	100	400	3	Attic	3
INTERIOR	85	345	0	House	4
INTERIOR RESIZED	85	345	0	House	3
IDEAL	100	400	0	House	3
IDEAL OVERSIZED	100	400	0	House	4

Table 5. Start Time to Pulldown by 5:00 p.m.		
	Rated Capacity [Tons]	**Start Time**
BASE	4	10:30 a.m.
POOR	4	Not possible[1]
BEST	4	2:30 p.m.
BEST RESIZED	3	11:45 a.m.
INTERIOR	4	1:45 p.m.
INTERIOR RESIZED	3	9:45 a.m.[2]
IDEAL	3	12:30 p.m.
IDEAL OVERSIZED	4	2:30 p.m.

1- 37°C at 5:00 p.m., pulldown to 24°C at 9:00 p.m. (drawing in cool outdoor air through return leaks)

2- Although this system is basically on all day, this result is misleading because the indoor temperature never gets above 25°C and a more lenient pulldown criteria drastically changes this result. For example, increasing the setpoint temperature by 1°C (to 25°C) changes the system ontime to 11:00 a.m. and makes it better, not worse, than the base case.

The results in Table 5 show that the time that the systems have to run covers a very wide range from two and a half hours for the BEST and IDEAL OVERSIZED systems to all day for the POOR system. In effect, looking at pulldown this way has further exaggerated the differences between the systems. This is mostly because the systems are now operating more during the heat of day rather than the cooler evening and night time. As with the results reported previously, this table shows that resized systems with good ducts can be as good or better than an existing BASE system and that there are large gains to be had by improving the duct systems. Assuming that the energy consumption scales with system capacity and ontime, and normalizing by the BASE case energy consumption it is possible to calculate the relative energy consumption for each simulation, as shown in Table 6.

Table 6. Relative Energy Consumed in Order to Pulldown by 5:00 p.m.	
	Percent of BASE case
BASE	100
POOR	260
BEST	40
BEST RESIZED	60
INTERIOR	50
INTERIOR RESIZED	85
IDEAL	50
IDEAL OVERSIZED	40

Because the POOR system is on all day, the energy consumption is far greater than for the other systems. All the other systems consume less energy than the base case while providing equal or superior comfort in terms of pulldown time. In particular, the resized systems all consumed less energy than the BASE case for these simulations.

Table 7. Model Delivered Capacity (TAR) Comparison (system on for 1.75 hours)				
	Nameplate Capacity [Tons]	**Tons at the Register [Tons]**	**Tons at the Register / Nameplate Capacity [%]**	**Ratio to Base Case [%]**
BASE	4	1.66	42%	100%
POOR	4	1.51	38%	91%
BEST	4	2.21	55%	133%
BEST RESIZED	3	1.66	55%	133%
INTERIOR	4	1.84	46%	110%
INTERIOR RESIZED	3	1.36	45%	109%
IDEAL	3	1.68	56%	135%
IDEAL OVERSIZED	4	2.28	57%	137%

Table 7 compares the results of the calculated TAR between the simulations. Note that for these calculations the systems have been running for almost two hours and are at quasi-steady-state and do not show the transient capacity reductions at the start of the pulldown. This was done so that the results are as close as possible to the manufacturers rating conditions, and we are not unfairly comparing the nameplate capacity to the transient system performance. In other words, we are being as generous as possible in our comparisons by reporting close to the highest system capacities. All but the POOR ducts are better than the BASE case in terms of delivered TAR and also TAR as a fraction of the nominal (nameplate) capacity of the equipment. All of the resized systems have TAR values closer to their nominal capacity than the BASE or POOR cases. However in all cases (even the ideal situation with correct system charge and airflow and minimal duct losses) the equipment capacities are much less than the nominal nameplate rating that a home owner has paid for.

Comparison of Field Measurements and Computer Simulations

The model was evaluated by comparing predicted temperatures to measured temperatures. Given the same temperatures, other variables used to determine energy flows (e.g., register flowrates) and comfort parameters (e.g., pulldown times) are the same for both modeled and measured data. An essential part of simulation design and use was verifying that the simulation makes accurate predictions. In this case, we were interested in predicting two parameters: tons at the register (delivered capacity) and pulldown time (time to cool down the house). For this purpose, we examined the temperatures of the four air nodes described above (attic, house, supply duct, return duct).

Over 100 days of measured data at 5 sites were used to evaluate the simulation model (4 in California and 1 in Texas). Overall there was very good agreement between the modeled and measured house and attic temperatures and good agreement between the duct air temperatures when the air handler fan was on, but not very good agreement when the air handler was off. In order to illustrate these and other strengths and weaknesses of REGCAP, the modeled/measured comparison is shown for two sites and each of the four modeled temperatures will be discussed individually. There was no attempt to show data that was either particularly favoring or condemning of REGCAP: the following illustrations are included to demonstrate both the strengths and the weaknesses of the model. Appendix 5 contains a preliminary analysis of some of the problems encountered when comparing modeled and measured results due to the sensitivity of the model to measured weather data.

The results for two homes are described in this section (sites 4 and 5). Both homes have floor areas of approximately 140 m^2 (1500 ft^2) and are located in a subdivision in Sacramento, CA. The ducts, air handler, furnace and indoor cooling coil were located in the attic in both homes. Site 4 had supply duct leakage fraction that is 5% of air handler flow, return leakage is 3%. Site 5 had a very tight duct system (both leakage fractions are less then 3%). Site 4 had a 2 ton system with a fixed orifice expansion valve and was found at 85% of manufacturer's refrigerant charge. Site 5 had a nominally 2.5 ton system with a thermal expansion valve (TXV) and was fully charged. For brevity, graphs comparing modeled and measured data are shown for sites 4 and 5 only, and the generalized discussion applies to all the comparisons between measured and modeled data.

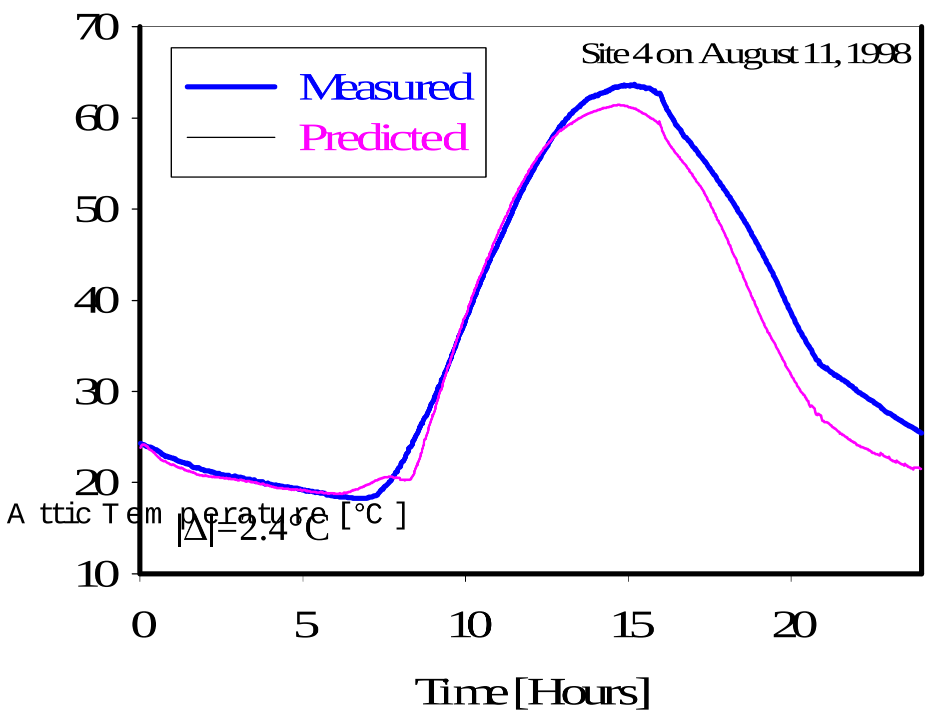

Figure 2: Modeled and Measured Attic Temperatures at Site 4 on August 11, 1998

Attic Temperature

These two houses show excellent agreement between the modeled and the measured attic temperature over the whole day. The agreement at site 4 is near perfect for the first half of the day and then the predicted temperature drops slightly below the measured temperature (Figure 2).

The average absolute difference in temperatures is 2.4°C (4.3°F). There are several hypotheses that explain this small discrepancy: the most plausible is a problem with the measured solar radiation input data (the dip in the data when the sun comes up is an indication of this) or, perhaps, the ducts are too strongly coupled with the house so that when the air conditioner comes on the duct leakage cools the attic more in the modeled case than in the measured case. Another possible problem is the fact that the radiative transfer involving the attic endwalls and the combined mass of wood in the attic was neglected. The modeled data at site 5 overpredicts the temperature for the first half of the day and then underpredicts it for the last half, but the overall average absolute temperature difference is 1.9°C (3.4°F), smaller than the difference at site 4.

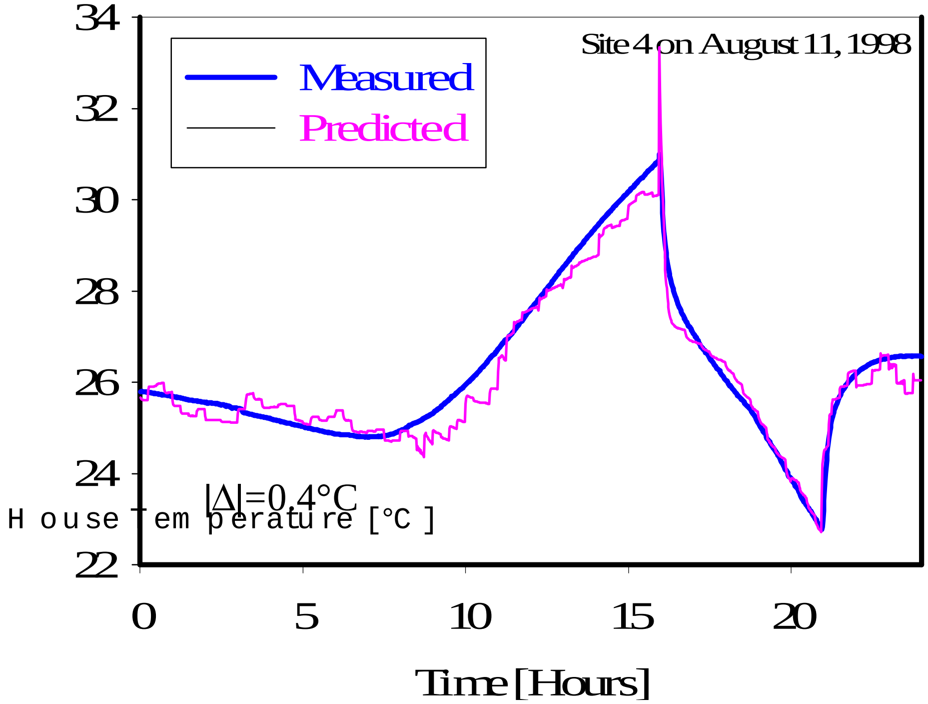

Figure 3: Modeled and Measured House Air Temperatures at Site 4 on August 11, 1998

House Temperature

The comparison of house temperatures at site 4 is shown in Figure 3. The average absolute difference between the modeled and the measured values is 0.4°C (0.7°F). The modeled house air responds very quickly to changes in climatic conditions. This may be due to insufficient coupling between the house air and the house mass. The agreement at site 5 is not as good, with an average absolute temperature difference of 0.6°C (1°F): examination of the weather data collected on the day of test indicates very strong winds from about 11am until 6pm. This is a failure of the model to deal with extreme conditions and is probably the cause of the wide temperature swings evident in the measured data. Both modeled houses have a single spike in the temperature when the air conditioner come on. This is an artifact of the ducts pushing hot air into the house that doesn't seem to be evident in the measured data (which was collected every 10 seconds, a finer resolution than the minute long timestep of the simulation). Despite these discrepancies, both sets of simulated data seem to reflect the overall shape of the temperature curve in each house. An improved house load model, such as Suncode,™ will probably increase the accuracy.

One problem with the house model is that the thermal mass of the house seems to be very weakly coupled to the house air. There are two most likely causes of this problem: the first is that the convection heat transfer coefficient for the house mass is biased towards natural, rather than forced, convection. This is an issue when there are strong winds (which lead to larger pressure differences and air velocities in the house), and when the air handler is on. This is a good example of where reducing the input value (i.e., no average air velocity in the house) leads to a less accurate predicted result. The second is that the surface area active in heat exchange between the thermal mass of the house and house air is too small in the model. Future work will further investigate this thermal mass issue.

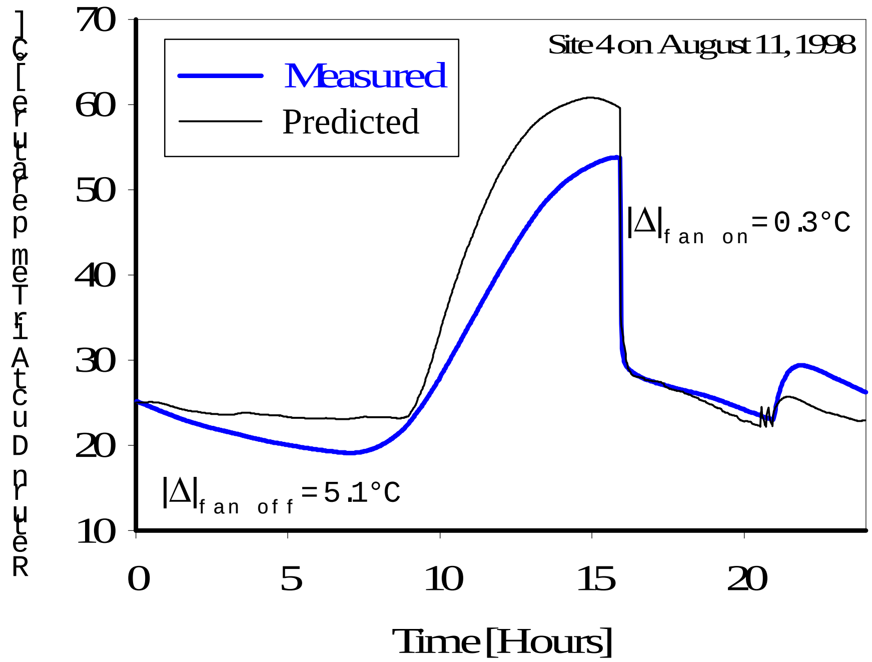

Figure 4: Modeled and Measured Return Duct Air Temperatures at Site 4 on August 11, 1998

Return Duct Air Temperature

The return duct agreement is quite good at site 4 (Figure 4) when the air conditioner is on (absolute difference of only 0.3°C). When it is off, the predicted duct temperature is much hotter than the measured temperature (absolute difference of 5.1°C). A very similar pattern occurs at site 5, with the same average absolute difference between the modeled and measured. The strong winds in the middle of day again affect the simulation quite strongly. Overall, REGCAP does an adequate job of prediction the temperature plots at both sites when the air handler fan is on.

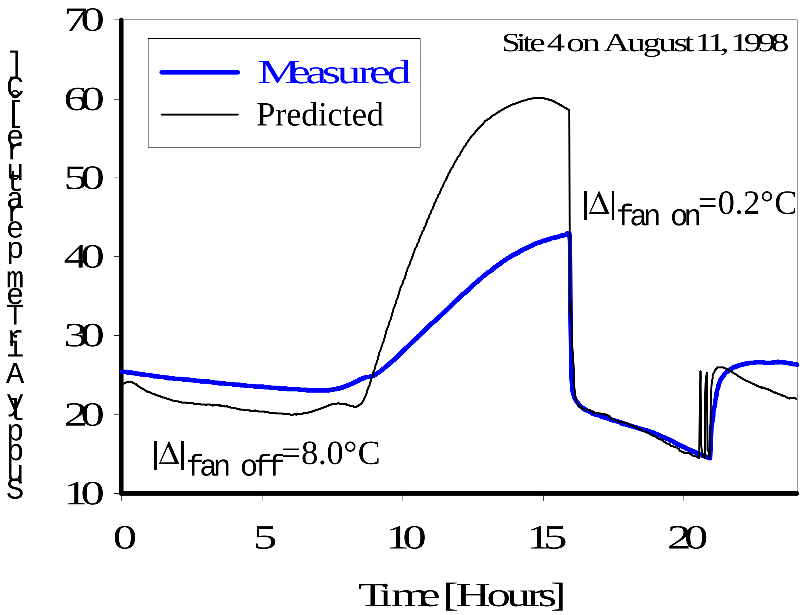

Figure 5: Modeled and Measured Supply Duct Air Temperatures at Site 4 on August 11, 1998

Supply Duct Air Temperature

The supply duct air temperature has a very similar pattern at both sites (Site 4 is shown in Figure 5). Like the return duct, the temperature shows good agreement when the air handler fan is on, but poor agreement when the air handler fan is off. When the air handler is off, the modeled supply duct temperature is very strongly influenced by the attic temperature and radiation exchange with the interior attic surfaces. The fact that the agreement is not very good for the duct air temperatures when the air handler is off may seem surprising because the model explicitly calculates the mass flow through these ducts when the air handler is off. However, there is a subtle distinction: REGCAP calculates the mass flow of air passing from the attic to the house (or the house to the attic) through the ducts, but does not calculate thermosiphon flows. Thermosiphon flows occur as air moves in one register and out another when the air handler is off. These flows are very difficult to calculate because to do so requires extensive information about the geometry of the duct system as well as being able to model flows between and within different rooms in the house.

The lack of air-handler off agreement for the duct temperatures is not particularly significant for the objectives of this study: predicting the pulldown time and the tons of cooling at the register. The only temperatures that are directly needed for these calculations are the house air temperature and the supply duct air temperature when the air conditioning fan is on. For this reason, REGCAP is well suited to calculating the performance parameters that are the focus of this project.

Field Measurement and Computer Simulation Outcomes

- **Improved ducts and system installation can allow the use of a smaller nameplate capacity air conditioner (almost one ton less in the simulations presented here, and at least one ton in more demanding situations) without any comfort penalty in terms of pulldown, and with large energy savings (roughly halving energy consumption).**
- **If system nameplate capacity is unchanged, either improving duct systems and correctly installing the equipment, or moving the ducts inside results in significant pulldown performance improvements.**

- **Simulations confirm field test results regarding delivered capacity and equipment and distribution system performance.**
- **Comparisons of computer simulation results to measured field data show that the simulations predict the equipment attic and house performance with sufficient accuracy to be a useful prediction tool.**
- **Field measurements of delivered cooling capacity are considerably less (20% to 50%) than nameplate & ARI ratings.**
- **Nameplate and ARI capacity ratings of equipment installed in houses exceed those indicated by ACCA Manual J load calculations.**
- **Thermal distribution system losses and poor equipment installation combine to reduce delivered capacity. Measured delivered capacities are close to those indicated as necessary by ACCA Manual J load calculations.**
- **Improving ducts by reducing leakage can lead to significant energy efficiency gains in addition to cooling the house faster.**
- **Efficient systems can still have problems satisfying occupant comfort even though the total delivered capacity for the system is correct due to room-to-room variations in delivered capacity for each room. The room-to-room variations result in large temperature variations throughout the house.**
- **Using higher SEER units indicated significant peak energy savings of about 25% with no apparent drawbacks in the houses measured.**

4. Support for Title 24 and HERS

The objective of this task was to:

- **Provide technical support to the California Energy Commission (CEC) for updating the "Low-Rise Residential Alternative Calculation Method Approval Manual for 1998 Energy Efficiency Standards for Low-Rise Residential Buildings" (CEC (1999)) and Procedures for HVAC System Design and Installation (for HERS).**

One of the most significant technology transfer activities in this project has been the inclusion of credits for energy efficient ducts in the Low-Rise Residential Alternative Calculation Method Approval Manual for 1998 Energy Efficiency Standards for Low-Rise Residential Buildings (ACM). The changes and additions were made to the Alternative Calculations Manual based on our technical and editorial input. They allow an energy credit to be claimed by having improved ducts that are field tested for leakage and do not use rubber adhesive cloth tape for duct seals.

We have also provided technical support for research sponsored by the California Energy Commission (CEC) on home diagnostics (for HERS). We have worked with Davis Energy Group (DEG) on the development of residential commissioning test protocols for these home diagnostics. This has included measurement of register flows, fan flows and duct leakage. For the register flow measurements, a combined study with DEG, LBNL, CEC and The Energy Conservatory that used flowhoods to measure register flows was undertaken. Eight different flowhoods were evaluated in a new house in Sacramento. The results of this testing (given in detail in Appendix 6) showed that standard flowhoods can be poor at measuring the register flows. This is due to a combination of:

- Low flows result in a small pressure signal from the flowhoods that leads to low precision.
- Poor calibration. Some of the flowhoods had large bias errors for all measurements, indicating a calibration problem.
- Sensitivity to flow asymmetry. The flowhoods are calibrated and designed to be used on registers with a uniform face velocities, but the registers in residential buildings are rarely operated in this manner and have strong flow variations across the face of the register.
- Flow restriction lowering the flow during the measurement. The restriction of flow due to inserting the flowmeters can be significant.

All of these problems were reduced by using fan assisted flowhoods. The fan assist is used to balance the pressure in the hood with the static pressure in the room. This was originally done to remove the effect of restricting the flow, but the side benefits are of equal, or greater, importance because the fan assist tends to remove the flow asymmetries and give better results with any remaining asymmetry. The calibrations for the flowmeters are well known and easily checked, and the flowmeter can be adjusted to be sensitive to low flow rates, thus improving the precision of the measurements. Unfortunately, the fan assisted flowhood is extra equipment to carry around a house and is equipment that home testers would have to become familiar with. In addition, there is the added expense of additional equipment purchase.

In addition to individual register flows, the CEC is also interested in requiring fan flow to be measured. The proposed method in ASHRAE 152P (and as used in our field testing) that requires blocking of the return and matching operating pressures is considered too time consuming and difficult. Some alternatives – such as measuring return grille flows with a flowhood and adding an estimate of the return leakage – are insufficiently accurate for use in a rating tool. Future possibilities for measuring fan flow may include the use of device under development at ECOTOPE that attaches in place of the filters in the system and requires less time and effort. We hope to evaluate the ECOTOPE system in the near future.

We are working to improve the ASHRAE 152P method by developing a new fan flow measurement device that utilizes a large powerful (but still portable) fan with a built in flowmeter. This device replaces the small fan flowmeters used in previous studies. It should allow us to replicate the fan flow in most residential systems without having to extrapolate from the measurement point to the system operating point (as required with the small fan flowmeters).

We worked with the staff of the CEC to evaluate an HVAC system performance tool developed by Federal Air Conditioning Technologies (FACT). This tool evaluates both the duct air flow and thermal performance as well as the refrigerant systems. We did comparison tests on a house in Sacramento and in LBNL's Building 51 laboratory with LBNL measurement equipment and the FACT equipment. We found some important measurement differences between the two types of equipment. However, without additional testing, it was not possible to pinpoint the exact reason for the discrepancies.

A collaborative of CIEE, CEC, the California Building Industry Association (CBIA) and the Natural Resources Defense Council (NRDC) developed procedures for improved design, fabrication, installation and testing of HVAC systems. We supported the updating of these procedures to ensure compatibility with the changes incorporated into the ACM for duct energy efficiency credits by reviewing a draft of "Procedures for HVAC System Design and Installation" (Hammon 1999). The draft procedures are in Appendix X. We are collaborating with CONSOL (the contractor who is upgrading this document) through this review process.

The key outcomes of this task were:

- **Duct efficiency calculations are included in the Low-Rise Residential Alternative Calculation Method Approval Manual for 1998 Energy Efficiency Standards for Low-Rise Residential Buildings" (CEC (1999)).**
- **Procedures for HVAC System Design and Installation (for Home Energy Raters) have been updated.**
- **Field testing has shown that standard flowhoods can be poor for measuring residential register flows.**

5. Technology Transfer

The objective of this task was to:

- **Support ASHRAE, ASTM and EPA duct leakage research and interface with realted projects funded by other agencies.**

ASHRAE: Rating of Distribution Systems - ASHRAE 152P

ASHRAE published and distributed a draft version of ASHRAE 152P for public review during May and June 1999. It is expected that the final draft of this standard will be ready by January 2000. We have also developed a web-based tool for performing 152P calculations. This tool can be accessed at http://ducts.lbl.gov. This web tool includes many defaults as guides for the uninitiated user that are taken from the appendices of 152P. These defaults are intended to make this web-tool easier to use.

ASTM: Rating of duct sealants and revising duct leakage measurement methods

We have attended ASTM meetings and corresponded with ASTM to discuss the implementation of an ASTM standard for longevity testing of duct sealants (ASTM (1999b)). A draft of the standard was prepared and voted on by ASTM E6.41 subcommittee members. Several comments were made on this draft which was then revised and will be reballotted later this year.

The current duct leakage test measurement in ASTM E1554 is obsolete. This standard has been rewritten based on the results of duct leakage test evaluations performed for the last three phases of the current research sponsored by CIEE/CEC, together with input from other ASTM members and the members of ASHRAE SSPC 152P. The new draft of the standard has two leakage test methods: the DeltaQ test and duct pressurization. The standard also includes the benefits and drawbacks of the two methods, so that the user can select the most appropriate test method for the test they are performing. For example, the DeltaQ test is better for measuring leaky duct systems for HERS testing, but the pressurization tests are more robust for low leakage compliance testing.

Other Thermal Distribution System Efficiency Support Activities

Several other tasks were performed under the scope of this study that relate directly to thermal distribution systems. The following is a summary of these activities:

Health and Safety Assessment of Aerosol Sealant (EPA)

As with any new industrial material, concern exists over the potential health hazards related to human exposure. Potential health and safety issues regarding the duct seal material were evaluated and discussed in Buchanan and Sherman (1999). This report examines the characteristics of the sealants' individual components as determined from current literature. There are three primary means by which exposure could occur: ingestion, eye/dermal contact, and inhalation. Each of these possibilities is examined. Exposure and safety risks were assessed with regard to the currently known constituents that are believed to pose potential hazards: VAP, VAM, 2EH, and acetaldehyde.

Field Testing of Energy Star® Equipment (EPA)

This field testing was performed in conjunction with the field testing for Phase VI. In one of the Sacramento houses and the Texas house the air conditioning equipment was replaced by higher efficiency Energy Star® equipment, but there was no resizing of the equipment. In both cases we replaced a standard SEER 10 unit for one rated at SEER 13. These additional tests funded, by EPA, added an additional three "systems" (the Sacramento house with SEER 13 plus the Texas house in two systems configurations) to the database for the Phase VI work.

Developing Energy Star Ratings for Duct Systems (EPA)

In addition to previous work on incorporating duct system efficiency in Energy Star Ratings for houses, we have also worked with EPA on developing rating methods, baseline studies and possible duct efficiency improvements for an Energy Star rating system. A preliminary report by Walker (1999) summarizes a sensitivity study performed for EPA that examines variability of distribution system efficiency with geographic location (climate) and duct system parameters (e.g., leakage). Additional ongoing work will determine baseline duct efficiencies throughout the country and estimate how much of the energy losses could be saved. This program is currently aimed at existing houses, but we plan to adapt it in the future for application to new construction.

Public Dissemination of Research Results

During this year we have been developing the Thermal Energy Distribution Web page **-** http://ducts.lbl.gov**.** This is intended to be a central reference point for disseminating information about thermal distribution systems in buildings, and the papers resulting from the work done for the current project will be "published" on this web site. We have also assisted CEC by preparing information for their thermal distribution system web page. We have further assisted CEC by participating in their triennial review process.

The results from work done for this phase of the Thermal Distribution Efficiency research program have been presented (and published) mostly at ASHRAE meetings and at the ACEEE 1998 Summer Study. The following presentations have been given in the last 12 months, some of which were based on work performed for the previous phase of this work. Section 7 lists recent publications associated with this research program.

Walker, I.S., (1999), "Distribution System Leakage Impacts on Apartment Building Ventilation Rates", ASHRAE Trans. Vol. No. (presented at ASHRAE TC 4.10 Symposium, January 1999), LBNL 42127.

Walker, I.S. and Sherman, M.H, (1999), "Assessing the Longevity of Residential Duct Sealants", RILEM 3rd International Symposium: Durability of Building and Construction Sealants, February 2000. LBNL 43381.

Walker, I., Sherman, M., Siegel, J., and Modera, M., (1999), "Comfort Impacts of Duct Improvement and Energy-Star Equipment", EPA Contract Report, LBNL 43723.

Walker, I, (1999), CIEE report on Benefits Estimates for CIEE Residential Thermal Distribution projects

Walker, I.S., (1999), "Sensitivity of Forced Air Distribution System Efficiency to Climate, Duct Location and Duct Leakage", EPA Report, LBNL 43371.

Walker, I.S., and Sherman, M.H., (1999), "Can Duct Tape Take the Heat", LBNL 41434.

The key outcomes of this task were:

- ASHRAE standard 152P for rating distribution systems has been prepared for and submitted to the ASHRAE public review process.
- The ASTM standard for duct leakage testing has begun the review process and a new standard for longevity of duct sealants has been proposed to ASTM.
- Support was provided for several thermal distribution system efficiency tasks sponsored by EPA.
- Several reports and papers have been published to allow public dissemination of research results.

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Walker, I.S. 1999. Distribution System Leakage Impacts on Apartment Building Infiltration Rates. *ASHRAE Trans. Vol. 105 Part 1.* (presented at ASHRAE TC 4.10 Symposium, January 1999), LBNL 42127.

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8. Appendices

Appendix 1: ASHRAE SP152P Duct Leakage Workshop Subcommittee Meeting

The workshop was held in January 1999 at the ASHRAE Winter Meeting in Chicago. Table A1.1 lists the attendees and their affiliations.

Table A1.1 Duct Leakage Workshop Attendance

Name	Affiliation
Iain Walker	LBNL
Paul Francisco	Ecotope, Inc.
Collin Olson	Energy Conservatory
Bruce Wilcox	BSG
Gary Nelson	Energy Conservatory
John Andrews	BNL
Chuck Gaston	Penn State - York
Mark Modera	LBNL
Michael Lubliner	WSU Energy

Three main proposals for alternative duct leakage measurements were discussed at this meeting by Chuck Gaston, Paul Francisco and John Andrews. In each case, the proposals were presented by each of the three proponents, and discussed by the committee.

Summary

There are several test methods that require further investigation, and the use of any individual method may depend on the application it is used for. For example, screening tests for compliance for tight duct systems require high repeatability at the expense of precision, but Home Energy Rating of houses with leaky duct systems require a higher precision so as to improve estimates of potential energy savings. Thus, the test method(s) used in proposed ASHRAE 152P may not be the same as those used by Home Energy Raters, code authorities or utility programs. It would be possible to include more than one test in proposed ASHRAE 152P and then have recommendations based on end use for selecting the appropriate test method. However, in a rating tool (which is what 152P has become) it is preferable to have a single test method so that different users of the test will achieve the same result. These are all issues for Standard Project Committee 152P members to think about and discuss during the public review of 152P. In addition, the ASTM standard E1554 needs to be reviewed and changed where necessary based on the additional experience gained in the past few years regarding duct leakage measurement.

A main conclusion of the discussion that is common to all the test methods (this includes the pressurization test if leakage to outside is measured and the house pressurized to the same pressure as the duct system) is that automated measurements offer a significant increase in the precision and usability (it broadens the range of acceptable weather conditions for testing). The standard(s) could specify that multiple automated pressure and flow measurements are a requirement to force users to improve the precision of their measurements. A caveat is that any test requiring envelope pressure measurements can be very difficult to perform on houses with leaky envelopes common to many older houses in the US housing stock.

The following four tests need to be evaluated over the coming months:

Inverse Test: This test was developed into the DeltaQ test that we are currently evaluating.

Nulling Pressure Testing: The applicability of this test is much improved if an automated measurement system is used, without this the test is really too difficult to perform. This test still requires blocking inside the duct system to separate supply and return leakage and determine total leakage.

Hybrid: This uses part of the HPT/Nulling test to determine the difference between supply and return leakage by measuring the envelope pressures, then using pressurization to estimate the total leakage. This test really requires automated envelope pressure measurements to minimize problems arising from weather induced pressure fluctuations (and inherent in all low pressure measurements).

Blower Door Subtraction: The automation of this test method has the same benefits as for the other tests in that the envelope pressure measurements are vastly improved. The other recommendation regarding multipoint testing requires more detailed investigation.

Chuck Gaston: "Inverse" Test

In this test, a blower door is set up the same way as for an envelope leakage test. A blower door test is performed over a range of pressures (going both positive and negative with respect to outside) for two cases:

1. with the distribution system fan off.
2. with the distribution system fan on.

Pressure (across the envelope) and flow data needs to be taken at as many pressure differences as possible. In theory, only as many data points as unknowns are required, extra data points overspecify the system but allow for fitting to remove the effects of noise in the signal caused primarily by wind pressure fluctuations. The multiple data points contain all the information required to determine the supply and return leakage flows at operating conditions (in the form of pressure exponents and flow coefficients). In addition, the envelope leakage is also determined. For the general case we have what is termed an "inverse" problem. This method has a great appeal because the test procedure and equipment is very simple. The analysis is very complex, but in use this complexity would be hidden from the user because a computer will be required to perform the analysis. As with all the tests that measure envelope pressure, it was recommended that the uncertainties can be reduced by automating the pressure measurement procedure to take large numbers of pressure readings until a certain variance limit is met. The use of curve fitted data that can be applicable in this test will also act to reduce the influence of pressure fluctuations

Paul Franciso: Nulling Pressure Testing

In this test, a fan and flowmeter are used to return the house envelope to its distribution fan off pressure difference when the system fan is on and therefore measure the leakage flow imbalance. Essentially, this acts to remove the uncertainties about envelope leakage from the existing house pressure test (HPT). Past experiments with this test procedure showed that it was difficult to adjust the fan precisely enough to get to exactly the same pressure as with the system fan off, particularly with wind induced fluctuations of envelope pressure. Paul discussed how this problem had been much reduced by using automated measurement software developed by Collin and Gary. Paul also showed how the supply leakage could be measured separately from the return leakage also using a nulling technique by using two fan/flowmeters. The combination of the measured supply leakage and the measured difference can then be used to calculate the return leakage. In the supply leakage measurement, the return is blocked off before the air handler and Fan/flowmeter (1) installed so as to blow air through the system. Fan/flowmeter (1) is then set up to produce the same flow through the system as at operating conditions by matching static pressure between the supply plenum and the conditioned space with that measured during normal system operation (this is the same technique as is currently in SP152P for measuring air handler fan flow). Fan/flowmeter (2) is connected across the building envelope and is then used to return the envelope pressure to the same value as with fan/flowmeter (1) off. The flow through fan/flowmeter (2) is then the supply leakage at operating conditions. As discussed above, using an automated system for pressure measurement (and possibly fan control) is an important part of this test.

Paul pointed out that this supply leakage test works best when fan/flowmeter (1) can be connected to draw air from inside the conditioned space. A second way this test can be done is with fan/flowmeter (1) drawing air from outside and using a blower door to depressurize the house. The flow through the blower door when the pressure across the envelope is the same as with the air handler off is then the supply register and supply leakage to inside. The difference between the blower door and fan/flowmeter (1) is the supply leakage to outside. This second method has some disadvantages that make the first method preferable:

1. You need to determine the difference between two large measured numbers whose uncertainty can be the same as the number you are looking for **-** this leads to large uncertainties in supply leakage flow.
2. The air drawn into the house will be at a different temperature than indoor air and some temperature corrections will have to be made to the measured volumetric flows.

John Andrews: Combined HPT and Pressurization (hybrid)

John has suggested that the first part of the HPT (House Pressure test) that is used to estimate the difference between supply and return leakage should be used in conjunction with a duct pressurization tests of the whole duct system. This procedure removes the time consuming and difficult task of physically separating the supply and return portions of the duct system. Although most of those present felt that eliminating the need to block the supply from the return is a significant time and effort savings, the sensitivity of the envelope pressure measurements to wind induced pressure fluctuations may create large errors in estimating the imbalance flow. Paul Franciso estimated that the uncertainty of the supply-return split using this method would be in the order of +/- 100 cfm. As with all the other envelop measurements the uncertainty can be significantly reduced if suitable time averaging of the pressures is performed using an automated data acquisition system.

Alternatively, if the separation were installed and the pressurization tests used to separate the supply and return duct leakage, using the HPT result for supply/return imbalance could be used to provide a simple additional check. This would be useful in systems where the estimate of operating pressure is poor (because all the leaks are concentrated in a single location), and the resulting flow calculated from the pressurization tests is incorrect by a large amount.

John also reported (and all those present agreed) that the current procedure in 152P of using pressure pans to estimate register pressures can be biased too high. There was a brief discussion regarding balancing the reduction in precision uncertainty against the increased biased errors inherent in the use of the pressure pans. This discussion also included removal of pressure measuring requirements altogether (except for systems with simple returns (and with filters at the return grille) where a single pressure measurement will be a good indicator of pressure across any return leaks). The measured flow at 25Pa would then be used as the leakage flow at operating conditions. In this case the test method is trading an increase in repeatability (everyone doing the test will use the same pressure difference) and simplicity against a possible decrease in precision. However, given the uncertainty in the measurement of pressures so that the resulting calculated flow is the flow out of the leaks (this requires a leakage flow weighted pressure) it is difficult to estimate the reduction in precision using this single pressure method. If the test is being used as a rating tool (particularly if a pass/fail criteria is set at a low leakage level) then this single pressure approach has merit. This approach has been taken by the California Energy Commission in the new Title 24 Residential Building Energy Code and has been used previously in utility and local government programs, in which repeatability is a key factor because the credit obtained for having good ducts should depend on the test personnel as little as possible. The opposite may be true for Energy Ratings of existing houses that have leaky duct systems where the exact leakage at operating conditions is a major factor in determining the energy use of the building and/or the energy benefits that would occur if the duct system were repaired/sealed.

The efficiency changes using a fixed pressure rather than attempting to use the measured register pressures (or plenum pressures) has been calculated using proposed ASHRAE 152P. The leakage flows were calculated using the limits of the register and plenum pressures as well as the fixed pressure (see details of these calculations below). It was difficult to reach a single conclusion from these example calculations, but they show that in many cases the selection of a single pressure may not introduce undue penalties. This is less true with leakier systems, where the change efficiency becomes more sensitive to leak pressure selection.

Gary Nelson: Improved Blower Door Subtraction

The current ASTM standard (E1554) for measuring duct leakage has not proved to be very popular. Gary Nelson has been working on improving this test procedure by focussing on automated data sampling to improve precision. Gary also discussed the possibility of doing multipoint tests, then using regression results in a modified version of E1554.

ASHRAE 152P Efficiency Limits Due to Extremes of Duct System Pressure Variation

At the recent ASHRAE 152P Duct Leakage Diagnostic Subcommittee Workshop, a question was raised regarding the maximum uncertainty in distribution system efficiency that arises from changing the assumptions about pressures across duct leaks. In order to investigate this, measured register and plenum pressures combined with measured pressure exponents were used to determine the change in leakage flows for measurements made at 25 Pa (as required in the standard). The pressures and exponents were measured in six test houses last summer. The houses were all new construction, with only one house already occupied. Two houses were in Sacramento, CA., two in Palm Springs, CA., one in Mountain View, CA., and the last house was in Cedar Park, TX (near San Antonio). The houses were tested in a total of 14 conditions that included sealing and adding leakage. The register pressures averaged 5 Pa and the plenum pressures averaged about 60 Pa. There was significant house to house variation (expressed as standard deviation) of these pressures of ±3 Pa for registers and ±30 Pa for plenums. These variations depended on many duct installation factors and there were significant differences even between houses with the same duct layout and equipment. It should be noted that the register pressures were not measured using a pressure pan, but were measured with carefully located pitot-static pressure probes inserted at the edges of the registers so as to be away from the main flow field. Based on our field experience (and that of other researchers) we can assume that pressure pan measurements would have yielded higher register pressures, thus leading to less extreme pressures and flows. Therefore the results of the calculations presented here are biased towards extreme values and using pressure pan measurements would have lowered the spread of calculated efficiencies. The leakage tests were performed at both 25 Pa and 50 Pa so that an estimate could be made of the pressure exponent. For these houses the average pressure exponent was 0.6. This value has been measured before in other multi-pressure duct leakage tests by LBNL.

Two base leakage levels at 25Pa were chosen (expressed as fractions of fan flow): the 22% total used as the default value in California T24 energy Code and the 6% total used in T24 for duct efficiency credit. This is a reasonable range of values for new construction. Note that the 22% and 6% are assumed to be evenly split between supplies and returns, i.e. 11% supply and 11% return for the 22% total case. Using the 5 Pa and 60 Pa limits and the mean measured pressure exponent of 0.6 gives the following leakage flow multipliers:

$$\text{All @ register leakage multiplier} = \left(\frac{5}{25}\right)^{0.6} = 0.38$$

$$\text{All @ plenum leakage multiplier} = \left(\frac{60}{25}\right)^{0.6} = 1.7$$

Combining the two base leakage levels with the three pressures (25 Pa, register and plenum) yields the following six leakage cases:

- 6% @ 25 Pa, 2% @ registers and 10% at plenums
- 22% @ 25 Pa, 8% @ registers and 38% at plenums

We can see from these simple calculations how the effect of the leak location assumption is much less at lower leakage levels.

For the 152P distribution system efficiency calculations, the following house was modeled:
2000 ft^2, 2 Story, Attic Ducts (10% regain), ACCA D design, single speed heating and cooling equipment, R4 flex duct with surface area equal to 152P defaults (assuming two return registers). The fan flow and capacity were determined using the defaults used in T24: 0.5 cfm/ft^2 for heating, 0.7 cfm/ft^2 cooling, heat exchanger temperature change was 55°F for heating and 20°F for cooling. This results in fan flows of 1000 cfm for heating and 1400 cfm for cooling. The capacities are 17.3 kW for heating and –8.7 kW (2.5 tons) for cooling. In Sacramento, the

cooling design temperature is 98°F and the seasonal temperature is 73°F. For heating, the design temperature is 32°F and the seasonal temperature is 48°F. Entering all this information (plus the cooling humidity information) into the 152P calculations resulted in the following table:

Table A1.2 Sacramento, CA ASHRAE 152P Distribution System Efficiency, %						
	Efficient Ducts			Typical Ducts		
Total Leakage	Base **6%**	Registers **2%**	Plenum **10%**	Base **22%**	Registers **8%**	Plenum **38%**
Heating Design	84	86	81	74	82	66
Heating Seasonal	85	87	83	76	84	68
Cooling Design	72	78	67	51	69	33
Cooling Seasonal	80	83	78	70	79	60
Mean	80	84	77	68	79	57

Table A1.2 shows that for the efficient ducts, the assumption about leak location changes the seasonal efficiencies by only a couple of percentage points. The design efficiencies are more sensitive and changed relative to the Base case by more than 5 percentage points for cooling. The acceptability of these results depends on what they are used for. With annual energy calculations (energy codes and home energy ratings) the seasonal results appear satisfactory, however, if one is interested in peak demand (design) calculations, then the five percentage point cooling differences may be too large. Bear in mind that the above results are extreme values, i.e. as bad as it could be, and so a five percentage point maximum error may still be acceptable for load and peak demand calculations.

The typical duct system showed much greater variation, with a 10 percentage point change relative to the Base case even for seasonal calculations.

It is reasonable to ask if using extreme results is a good indicator of the range of performance of real duct systems. To answer this we can look at recent tests performed by LBNL that separated the register boot and plenum/cabinet leakage from the total. In the seven houses tested, the combined plenum/cabinet and boot leakage accounted for 25% to 75% of the total system leaks, with an average of about 74%. This result indicates that the above results are extreme and would be unlikely to occur in a real house. Looking at the 25 Pa leakage results for registers and plenums/cabinets separately: the registers averaged 53% of leakage to outside (range of 20% to 60%, standard deviation of 22%) and plenums/cabinets averaged 21% of leakage to outside (range of 5% to 50%, standard deviation of 18%). Summarizing these results, it looks like the extreme case is to have about half of the system leaks at either the plenum/cabinet or the registers, while a typical system has about one half the 25 Pa leakage at the registers and about 20% at the cabinet/plenum. To first order, this implies that you would expect about half the variation indicated in Table A1.2. This means that all the results would be within a plus or minus five percentage point band.

Another aspect of the results is that they are for ducts in an extreme location – the attic. For another type of system – a basement heating system in Fargo, ND., Table A1.3 shows that the leakage induced variations in efficiency are small because the total losses are small, even for very leaky ducts. In this case, I used the same house as above, but placed the ducts in an "unconditioned" basement with insulated walls (regain of 75%), and a design temperature of –18°F and a seasonal temperature of 14°F.

Table A1.3 Fargo, ND ASHRAE 152P Distribution System Efficiency, %						
	Efficient Ducts			Typical Ducts		
Total Leakage	Base **6%**	Registers **2%**	Plenum **10%**	Base **22%**	Registers **8%**	Plenum **38%**
Heating Design	93	94	91	87	92	81
Heating Seasonal	93	94	92	87	92	82

Appendix 2. Delta Q duct Leakage test

Procedure:
1. Install blower door and envelope pressure difference tubing/sensor.
2. With blower door fan opening blocked, blower door off and system off measure pressure difference across envelope with blower door off ΔP_{zero}. ΔP_{zero} is subtracted off all the envelope pressure measurements (or remeasured at the end of the test and some average used).
3. Turn on the system and measure the pressure across the envelope, ΔP_{env} (at Q=0, where Q is the flow through the blower door).
4. Measure the plenum operating pressures - ΔP_s for supply and ΔP_r for return – relative to the conditioned space. Note that both pressures are recorded as positive numbers for use in the analysis, i.e., the return pressure is NOT negative.
5. Turn on the blower door until there is 5 Pa across the envelope. Record ΔP_{env}, and Q_{on}.
6. Turn off the system fan and adjust the blower door fan to obtain the same pressure ΔP_{env} across the envelope. When the pressures are matched, record Q_{off}.
7. Repeat steps 5 and 6, but with the envelope pressure, ΔP_{env}, incremented by about 5 Pa each time. At each ΔP_{env} there will be a pair of flows Q_{on} and Q_{off}.
8. Subtract ΔP_{zero} from each ΔP_{env} to obtain ΔP.
9. Calculate ΔQ_i at each P_i by subtracting $Q_{off,i}$ from $Q_{on,i}$.
10. Do a non-linear fit of the P and ΔQ pairs to:

$$\Delta Q(P) = Q_s\left[\left(1+\frac{P}{\Delta P_s}\right)^n - \left(\frac{P}{\Delta P_s}\right)^n\right] - Q_r\left[\left(1-\frac{P}{\Delta P_r}\right)^n + \left(\frac{P}{\Delta P_r}\right)^n\right]$$

to find supply leakage: Q_s, return leakage: Q_r, and the pressure exponent for duct leaks: n.

Note that all envelope pressures are measured relative to outside – i.e. $P_{in} - P_{out}$, so that pressurization of the house is a positive pressure. Similarly, flows into the house through the blower door are also positive.

Derivation of DeltaQ test

Nomenclature:
C_{env} = flow coefficient for building envelope
C_r = flow coefficient for return duct leaks
C_s = flow coefficient for supply duct leaks
n_{env} = envelope pressure coefficient
n_r = return leak pressure coefficient
n_s= supply pressure coefficient
Q_{on} = measured flow through blower door with A/H fan on
Q_{off}= measured flow through blower door with A/H fan off
Q_s = supply leak flow at operating conditions to outside
Q_r = return leak flow at operating conditions to outside
ΔP = pressure difference across envelope (in-out)
ΔP_s = pressure difference across supply leaks at operating conditions.
ΔP_r = pressure difference across return leaks at operating conditions (note that this is a positive number for flow into ducts, so Q_r is positive)

With the A/H fan off we have:

$$Q_{off}(\Delta P) = C_{env}(\Delta P)^{n_{env}} + C_r(\Delta P)^{n_r} + C_s(\Delta P)^{n_s}$$

With the A/H fan on we have:

$$Q_{on}(\Delta P) = C_{env}(\Delta P)^{n_{env}} + C_r(\Delta P - \Delta P_r)^{n_r} + C_s(\Delta P + \Delta P_s)^{n_s}$$

"DeltaQ" is the difference between these two:

$$\Delta Q(\Delta P) = Q_{on}(\Delta P) - Q_{off}(\Delta P) = C_s\left[(\Delta P + \Delta P_s)^{n_s} - (\Delta P)^{n_s}\right] + C_r\left[(\Delta P - \Delta P_r)^{n_r} - (\Delta P)^{n_r}\right]$$

Defining the supply and return leakage flows:

$$Q_s = C_s(\Delta P_s)^{n_s} \qquad Q_r = C_r(\Delta P_r)^{n_r}$$

and $C_s = \dfrac{Q_s}{(\Delta P_s)^{n_s}} \qquad C_r = \dfrac{Q_r}{(\Delta P_r)^{n_r}}$

Substituting C_s and C_r into the deltaQ equation, we get:

$$\Delta Q(\Delta P) = Q_s\left[\left(\frac{\Delta P + \Delta P_s}{\Delta P_s}\right)^{n_s} - \left(\frac{\Delta P}{\Delta P_s}\right)^{n_s}\right] + Q_r\left[\left(\frac{\Delta P - \Delta P_r}{\Delta P_r}\right)^{n_r} - \left(\frac{\Delta P}{\Delta P_r}\right)^{n_r}\right]$$

This equation can be solved for Q_s, Q_r, n_s and n_r given the measured plenum pressures, ΔQ's and ΔP's. However, it is easier (and more robust) if we fix the duct leakage pressure exponents. Experiments to characterize the pressure exponent have shown that a value of 0.6 is suitable for most duct systems. The variability in this exponent is between 0.5 and 0.7. If we fix the value of n, and do alittle algebraic manipulation we get a form that gives DeltaQ in terms of a difference between the supply and return leaks and is a little clearer to interpret (e.g., it is easier to see that when ΔP=0, then ΔQ is the difference between supply and return leaks).

$$\Delta Q(\Delta P) = Q_s\left[\left(1 + \frac{\Delta P}{\Delta P_s}\right)^{n} - \left(\frac{\Delta P}{\Delta P_s}\right)^{n}\right] - Q_r\left[\left(1 - \frac{\Delta P}{\Delta P_r}\right)^{n} + \left(\frac{\Delta P}{\Delta P_r}\right)^{n}\right]$$

Uncertainty Estimate for exponent and duct pressure assumptions

Using plenum pressures assumes that these pressures characterize the pressure across the leaks. The uncertainty associated with fixing the value of n and using plenum pressures has been investigated parametrically by using an actual DeltaQ test and varying n and the supply and pressures. The following table contains the results of this parametric study. In Table A2.1, the pressures were varied over a range that captures the variation we expect to find in a duct system. If the leaks were all at the registers, then we need to use a low pressure: 5 Pa in this case, and if we change the pressure measurement location (and orientation of the pressure probe) in a plenum we find that pressures can change by a factor of two: as shown by the increased pressures in the table. Note that in this table we have used the worst case values in order to bound the problem. An estimate of typical uncertainty would be less than the variation shown here.

Table A2.1 DeltaQ Sensitivity Test					
ΔP_s, Pa	ΔP_r, Pa	n	Q_s, cfm	Q_r, cfm	
9.8	22.4	0.6	14	167	plenum pressures: actual measurements, fixed n
9.8	22.4	0.7	44	187	High value of n
9.8	22.4	0.5	-7	155	Low value of n
20	22.4	0.6	31	194	doubled supply pressure
5	22.4	0.6	-2	151	halved supply pressure
40	22.4	0.6	33	199	quadrupled supply pressure
9.8	10	0.6	1	177	halved return pressure
9.8	40	0.6	21	163	doubled return pressure
9.8	5	0.6	-18	178	quartered return pressure

These results show that this test method is not very sensitive to the assumed pressure exponent or the leak pressures. Note that this result only applies to this particular test, so we will have to do similar sensitivity studies on some more house results before we can say that the test method is insensitive for all situations.

Flow Adjustments for Exact Pressure Matching

The trickiest part of the test procedure is the matching of pressures with the distribution fan on and off. With an automated system that monitors the envelope pressure and can adjust the fan this would be made much easier (particularly if the envelope pressures have the typical fluctuations seen in field tests). Because the pressure and flow pairs will not be exactly matched, we need to have a procedure for determining what the flow difference should be with the pressures matched exactly. This procedure can also be automated as follows. If we take the system fan off as the reference pressure and flow condition we need to match the fan on conditions. By doing a power law fit to all the fan on data we can obtain the pressure exponent for the fan on data. Using this exponent and the ratio of the reference pressure to the actual fan on pressure we can find the fan on flow at the reference pressure:

Let the reference pressure for a given data point be ΔP_{off} and the corresponding flow is Q_{off}. We now take some distribution system fan on data at ΔP_{on} with a corresponding Q_{on}. Although ΔP_{on} and ΔP_{off} are close they are not exactly the same. If we fit to all of the fan on data we can obtain the pressure exponent n_{on}. The on flow can now be corrected to be at the same pressure as the off data:

$$Q_{on,corrected} = Q_{on}\left(\frac{P_{off}}{P_{on}}\right)^{n_{on}}$$

This correction can be applied to all the fan on data so that we have flows at exactly matched pressures. Because we aim to have the measured P_{off} and P_{on} close to begin with, any uncertainties in assuming that the pressure flow relationship is a power law and in evaluating the pressure exponent are small. In other words, because the flow corrections will be small anyway (probably less than 5%), the errors in this interpolation procedure will not be significant.

Comparison to other measurements

The pilot test of the DeltaQ procedure was performed in a house that we have already made many duct leakage measurements in. The following table summarizes the test results for comparison purposes.

Table A2.2 Comparison of duct leakage measurement procedures					
	DeltaQ	Duct Pressurization[1]	Duct Pressurization[2]	NPT[3]	Tracer gas
Q_s, cfm	14	51	30	17	n/a
Q_r, cfm	167	116	95	151	160

1- Converted to operating pressures using pressure pans
2- Converted to operating pressures using plenum pressures
3- NPT = Nulling Pressure Test.

Appendix 3. Summary of field measurement performance metrics

In the following tables, there are some results that are counter intuitive. The main culprit in these cases is the uncertainty in the relative humidity measurements. For example, in Table A3.1 there are some cases where the "total" TAR is less than the "sensible" TAR. This is particularly evident in the 5 minute results for Site 2, where the leaks sealed case gives the lowest TAR. Detailed examination of the measured data has shown that these anomalies are due to poor RH measurements. Improved RH measurements (and plenum temperatures) are needed to reduce the incidence of these results. For this reason, we are now performing improved RH calibrations on the RH sensors and also performing period field recalibrations.

Table A3.1 Tons At the Register

	5minutes		30 Minutes		60 minutes	
	Total	Sensible	Total	Sensible	Total	Sensible
condition	Mean	Mean	Mean	Mean	Mean	Mean
Site 1 as found	1.8	1.6	3.0	3.0	2.8	2.9
Site 1 sealed	3.5	3.0	3.0	2.9	2.7	2.6
Site 2 as found	3.9	2.6	2.6	2.7	2.6	2.8
Site 2 leaks added	4.1	2.6	2.3	2.5	N/a	N/a
Site 2 leaks added, correct charge	4.8	2.6	3.1	2.4	2.2	2.4
Site 2 sealed, correct charge	2.6	2.8	2.2	2.6	2.2	2.6
Site 3 as found	3.3	2.1	2.9	2.1	2.8	2.1
Site 3 as found correct charge	3.4	2.2	3.1	2.3	3.1	2.3
Site 3 sealed, correct charge	4.1	2.4	3.7	2.6	3.6	2.7
Site 4 as found	3.1	1.1	3.0	1.2	2.9	1.3
Site 4 as found, new compressor	1.9	1.3	1.7	1.3	1.6	1.3
Site 4 Leaks added	1.4	1.0	1.3	1.0	1.4	1.2
Site 5 as found	1.8	1.3	1.6	1.2	1.6	1.4
Site 5 sealed	1.8	1.3	1.6	1.4	1.6	1.4
Site 6 as found	2.8	1.2	2.0	1.3	1.9	1.4
Site 6 as found, new compressor	n/a	n/a	2.2	1.4	2.0	1.4
Site 6 leaks added	2.8	1.4	2.1	1.4	2.0	1.4

Table A3.2 Capacity at the indoor coil						
	5 minutes		30 minutes		60 minutes	
	Total	Sensible	Total	Sensible	Total	Sensible
Condition	KW	kW	kW	kW	kW	kW
Site 1 as found	7.4	6.7	6.2	5.8	12.5	12.7
Site 1 sealed	13.6	12.2	12.45	11.7	11.6	11.2
Site 2 as found	18.2	12.3	11.9	12.3	11.9	12.7
Site 2 leaks added	18.4	12.4	11	12		
Site 2 leaks added, correct charge	20.9	12.3	14.4	11.8	10.9	12
Site 2 sealed, correct charge	10.9	11.6	10.1	11.6	9.8	11.2
Site 3 as found	13.7	8.6	12.1	8.8	11.386	8.4
Site 3 as found correct charge	13.2	8.2	12.2	8.56	12	8.7
Site 3 sealed, correct charge	14.5	8.2	13.6	9.5	13.2	9.7
Site 4 as found	12.3	4.8	11.7	5	11.5	5.2
Site 4 as found, new compressor	7.6	5.17	6.9	5.4	6.73	5.5
Site 4 Leaks added	6.1	4.53	5.6	4.7	5.49	4.67
Site 5 as found	9.9	6.9	8	7.1	7.8	7.1
Site 5 sealed	7.96	6.1	7.1	6.3	5.6	5
Site 6 as found	12.1	6	9.5	6.9	9.2	7.1
Site 6 as found, new compressor			9.94	7.17	9.7	7.4
Site 6 leaks added	11.8	6.6	9.3	6.7	8.97	6.8

Table A3.3 System Power consumption				
	5 minutes	30 minutes	60 minutes	fan as fraction of compressor
Condition	kW	kW	kW	
Site 1 as found	5.6	5.1	5.1	0.1
Site 1 sealed	5.4	5.3	6.1	0.1
Site 2 as found	5.6	5.7	5.4	0.1
Site 2 leaks added	5.3	5.2		0.1
Site 2 leaks added, correct charge	5.7	5.7	5.6	0.1
Site 2 sealed, correct charge	6.1	6.6	6.5	
Site 3 as found	4.6	4.5	4.7	0.2
Site 3 as found correct charge	4.4	4.2	4.2	0.2
Site 3 sealed, correct charge	4.2	4.2	4.2	0.2
Site 4 as found	2.9	2.9	2.8	0.2
Site 4 as found, new compressor	2.4	2.4	2.3	0.3[1]
Site 4 Leaks added	2.1	2.1	2.1	0.3
Site 5 as found	3.1	3.1	3.1	0.2
Site 5 sealed	3.2	3.2	3.1	0.2
Site 6 as found	3.7	3.7	33.7	0.2
Site 6 as found, new compressor		3.7	3.7	0.2
Site 6 leaks added	3.8	3.9	3.9	0.15

1 – Large variation indicating a variable speed compressor see above

Table A3.4 Key Temperatures and Enthalpies for calculating system performance									
	5 minutes			30 minutes			60 minutes		
condition	Tout[1] (°C)	hreturn[2] (kJ/kg)	Tattic[3] (°C)	Tout (°C)	hreturn (kJ/kg)	Tattic (°C)	Tout (°C)	hreturn (kJ/kg)	Tattic (°C)
Site 1 as found	29.3	44.4	35.4	26.9	36.4	30.3	26.7	35	37.3
Site 1 sealed	30.7	44.5	37.7	31.6	41.3	37.3	36.7	43.1	40.5
Site 2 as found	28.8	41.4	33.6	30.7	40.3	36.7	27.5	36.3	29.9
Site 2 leaks added	26	40.3	28.4	25.5	37.6	27.2			
Site 2 leaks added, correct charge	30.3	41.9	32.6	29.8	38.7	31.5	29.3	37	30.3
Site 2 sealed, correct charge	33.2	43.4	40.1	36.9	43.7	41.1	36.2	41.6	39.9
Site 3 as found	28.9	56.2		29.5	52.6		32.9	54.9	
Site 3 as found correct charge	26.3	55		25.6	50.7		24.4	48.5	
Site 3 sealed, correct charge	24.7	50.7		24.5	45.6		24.1	43.1	
Site 4 as found	36.9	61.4	60.2	36.5	56.1	57.5	35.8	53.6	55
Site 4 as found, new compressor	32.2	57.4	56	32.02	52.2	53.4	31.9	49.6	50.6
Site 4 Leaks added	32.82	55.7	53.6	32.6	51.8	50.9	31.9	49.6	48.1
Site 5 as found	31.6	52.4	44.5	31.7	47.1	40.3	31.3	44.8	38.6
Site 5 sealed	34.7	52.6	47.3	33.4	46.9	42.4	33.1	44.8	40.6
Site 6 as found	33.7	54.5	53.8	33.96	48.9	54.4	34.3	46.7	53.1
Site 6 as found, new compressor			38.8	33.4	48.8	48.9	33.8	46.3	50.6
Site 6 leaks added	27.4	41.8	37.1	27.1	46.4	37	27.2	43.6	35.8

1- outside air dry bulb temperature
2- enthalpy of air in return
3- attic air dry bulb temperature

Table A3.5 Temperature at different locations in the house during pulldown tests									
	5 minutes			30 minutes			60 minutes		
condition	Thermostat [°C]	Master BR [°C]	Kitchen [°C]	Thermostat [°C]	Master BR [°C]	Kitchen [°C]	Thermostat [°C]	Master BR [°C]	Kitchen [°C]
Site 1 as found	26.1	27.7	27.2	23.2	24.1	21.9	22.1	23.2	20.7
Site 1 sealed	25.8	27.1	25.1	24	25.3	23	23.1	25.2	22.7
Site 2 as found	25.8	25.7	23.8	23	24.7	22.3	22.5	22.9	19.9
Site 2 leaks added	24.5	24.3	22.6	22.5	22.6	20.4			
Site 2 leaks added, correct charge	24.7	25	23	22.9	23.3	20.9	22.1	22.8	20.3
Site 2 sealed, correct charge	25.2	26.3	23.9	24.8	26	23.3	23.8	25.2	22.3
Site 3 as found	25.3	27.7	25.1	23.6	26.6	23.64	23.9	27.4	24.2
Site 3 as found correct charge	25.7	28.2	25.2	23.8	26.7	23.2	23	26	22.6
Site 3 sealed, correct charge	25	27.7	24.6	23.2	25.5	22.7	22.2	24.7	21.9
Site 4 as found	30	26.9	28.1	28.5	26.7	25.9	27.6	26.1	25
Site 4 as found, new compressor	28.6	27.5	26.8	27	25.5	24.7	26.1	24.8	23.8
Site 4 Leaks added	28.1	27	26.9	26.7	25.1	24.9	26	24.4	24.1
Site 5 as found	28.4	31.1	27.6	25.4	29	24.8	24.6	28.6	23.9
Site 5 sealed	29.2	30.8	27	25.9	28.5	23.8	24.9	28	22.96
Site 6 as found	26.6	30.1	28	27.8	28.6	25.5	27.3	28.2	24.7
Site 6 as found, new compressor				27.6	28.4	25.2	27	27.7	24.4
Site 6 leaks added	24.4	26	23.6	23.6	24.5	21.4	22.9	23.7	20.7

Table A3.6 Delivery Effectiveness						
	Sensible			Total		
	5 minutes	30 minutes	60 minutes	5 minutes	30 minutes	60 minutes
Site 1 as found	0.95	0.81	0.80	0.95	0.81	0.80
Site 1 sealed	0.88	0.85	0.82	0.88	0.85	0.83
Site 2 as found	0.76	0.76	0.77	0.83	0.78	0.76
Site 2 leaks added	0.73	0.74		0.79	0.73	
Site 2 leaks added, correct charge	0.74	0.72	0.72	0.81	0.75	0.70
Site 2 sealed, correct charge	0.85	0.79	0.80	0.85	0.78	0.78
Site 3 as found	0.87	0.86	0.89	0.85	0.85	0.87
Site 3 as found correct charge	0.95	0.94	0.94	0.90	0.90	0.91
Site 3 sealed, correct charge	1.05[1]	0.98	0.98	0.99	0.96	0.97
Site 4 as found	0.85	0.85	0.84	0.88	0.89	0.88
Site 4 as found, new compressor	0.86	0.83	0.85	0.88	0.85	0.87
Site 4 Leaks added	0.78	0.77	0.90	0.80	0.79	0.90
Site 5 as found	0.64	0.68	0.70	0.70	0.71	0.72
Site 5 sealed	0.72	0.77	0.79	0.77	0.80	0.80
Site 6 as found	0.69	0.67	0.67	0.82	0.75	0.74
Site 6 as found, new compressor		0.78	0.74			
Site 6 leaks added	0.77	0.75	0.75	0.85	0.80	0.80

1- Error in supply plenum temperature from spatial variation and response time

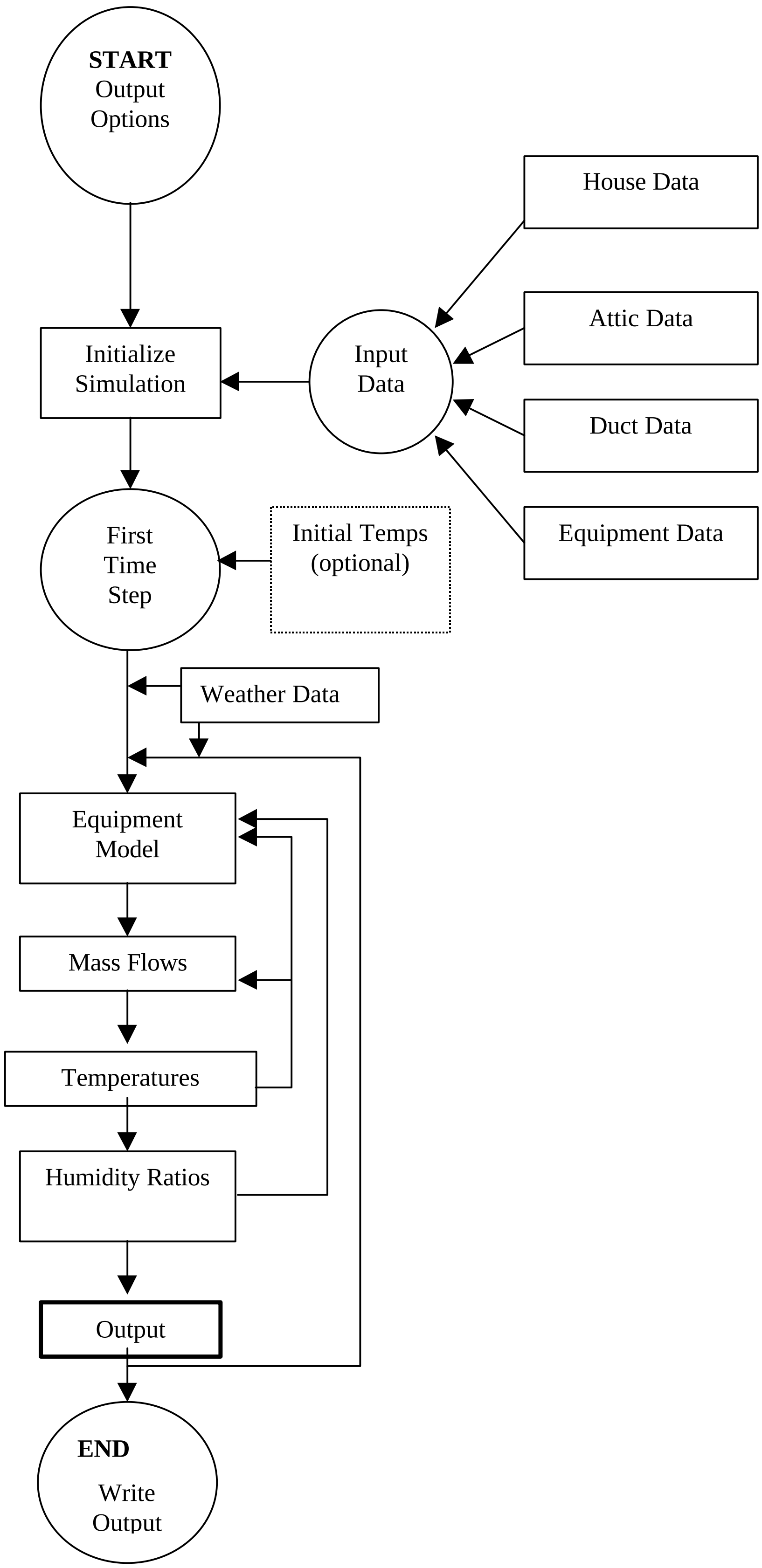
The Delivery Effectiveness is often higher at the start (5 minute values) due to lower conduction losses. A t the beginning of the cycle the air in the ducts is not as cool as later in the cycle.
The sensible DE is a function of both conduction and leakage. The total DE contains the moisture losses that are from leakage only.

Table A3.7 Equipment Coefficient of Performance (COP)						
	Sensible			Total		
	5 minutes	30 minutes	60 minutes	5 minutes	30 minutes	60 minutes
Site 1 as found	0.7	2.6	2.5	0.7	2.6	2.5
Site 1 sealed	2.3	2.1	2.0	2.5	2.1	2.1
Site 2 as found	2.3	2.1	2.4	3.5	2.1	2.2
Site 2 leaks added	2.4	2.3		3.5	2.1	
Site 2 leaks added, correct charge	2.2	2.1	2.0	3.7	2.5	1.9
Site 2 sealed, correct charge	1.9	1.7	1.7	1.8	1.5	1.5
Site 3 as found	1.9	2.0	1.8	3.0	2.7	2.4
Site 3 as found correct charge	1.9	2.0	2.1	3.0	2.9	2.9
Site 3 sealed, correct charge	1.9	2.3	2.3	3.4	3.2	0.3
Site 4 as found	1.7	1.8	1.9	4.4	4.3	4.3
Site 4 as found, new compressor	2.2	2.4	2.4	3.3	3.0	2.9
Site 4 Leaks added	2.2	2.2	2.2	2.9	2.7	2.6
Site 5 as found	2.2	2.3	2.3	2.9	2.6	2.5
Site 5 sealed	1.9	2.0	2.0	2.5	2.3	2.2
Site 6 as found	1.6	1.9	1.9	3.2	2.6	2.5
Site 6 as found, new compressor		1.9	2.0		2.7	2.6
Site 6 leaks added	1.8	1.7	1.8	3.1	2.4	2.3

Table A3.8 Total System Coefficient of Performance (COP)						
	Sensible			Total		
	5 minutes	30 minutes	60 minutes	5 minutes	30 minutes	60 minutes
Site 1 as found	0.7	2.1	2.0	0.7	2.1	2.0
Site 1 sealed	2.0	1.8	1.6	2.2	1.7	1.7
Site 2 as found	1.6	1.6	1.8	2.5	1.6	1.7
Site 2 leaks added	1.7	1.7		2.8	*1.5*	
Site 2 leaks added, correct charge	1.6	1.5	1.5	3.0	1.9	1.4
Site 2 sealed, correct charge	1.6	1.4	1.4	1.5	1.2	1.2
Site 3 as found	1.6	1.7	1.6	2.6	2.3	2.1
Site 3 as found correct charge	1.8	1.9	2.0	2.7	2.6	2.6
Site 3 sealed, correct charge	2.0	2.2	2.3	3.4	3.1	3.1
Site 4 as found	1.4	1.5	1.6	3.9	3.8	3.8
Site 4 as found, new compressor	1.9	2.0	2.1	2.8	2.5	2.6
Site 4 Leaks added	1.7	1.7	2.0	2.4	2.1	2.4
Site 5 as found	1.4	1.5	1.6	2.0	1.8	1.8
Site 5 sealed	1.4	1.5	1.6	1.9	1.8	1.8
Site 6 as found	1.1	1.2	1.3	2.7	1.9	1.8
Site 6 as found, new compressor		1.4	1.4		2.1	2.0
Site 6 leaks added	1.4	1.3	1.3	2.6	1.9	1.9

Table A3.9 Pulldown time and temperature variation in different locations in the house						
	Pulldown Time (minutes)			Temperatures (°C)		
	Thermostat	Master BR	Kitchen	Thermostat	Master BR	Kitchen
Site 1 as found	11	14	22	24.3	25.2	26.8
Site 1 sealed	32	19	50	24.1	23.1	25.4
Site 2 as found	26	10	27	24.5	22.3	24.4
Site 2 leaks added	8	2	7	24.0	21.9	23.8
Site 2 leaks added, correct charge	10	3	13	24.0	22.2	24.2
Site 2 sealed, correct charge	30	10	55	23.9	22.8	25.3
Site 3 as found	29	45	120	24.0	24.1	27.0
Site 3 as found correct charge	28	14	180	24.0	23.4	26.8
Site 3 sealed, correct charge	15	10	95	24.0	23.6	26.6
Site 4 as found	239	122	198	24.1	21.9	23.2
Site 4 as found, new compressor	159	64	107	24.0	21.9	23.1
Site 4 Leaks added	170	75	92	24.0	22.1	22.7
Site 5 as found	87	63	204	23.9	23.5	27.7
Site 5 sealed	107	34	180	24.0	22.2	26.7
Site 6 as found	257	93	266	24.0	21.0	24.2
Site 6 as found, new compressor	118	20	123	23.2	20.3	23.1
Site 6 leaks added	94	8	75	23.6	22.8	25.2

Appendix 4. Flowchart for REGCAP Model



Appendix 5. REGCAP Simulation Sensitivity to input data uncertainty

Additional model verification tests have been completed at sites 1 and 2. The results of these comparisons are less encouraging, but are based on problems with the input (measured) data rather than problems with the model. The average absolute temperature differences for each air node at each site are shown below in Table A5.1. It is clear that the model to measured comparison is not very good at these sites, although it is acceptable in the house and ducts when the air handler is on during the pulldown tests. The model overpredicts the attic temperature difference by a very large margin both when the air handler is on and when the air handler is off. This discrepancy is caused by errors in the solar input data. The solar sensors were poorly calibrated at these two sites and the shading device was very rudimentary and didn't always work. This caused us to overestimate total horizontal radiation and to underestimate direct normal radiation. This in turn causes the model to incorrectly overpredict the solar gain on both the house and the attic (although more significantly on the attic). Other problems with the model that these two sites revealed include the lack of coupling between the solar gain and the house mass and the inadequate heat transfer coefficient between the house mass and the house air. These problems will be corrected when an improved load calculation routine is implemented.

Table A5.1. Comparison of measured and modeled temperatures illustrating problems with measured input data

Site	Date	$\lvert T_{hse} \rvert$		$\lvert T_{attic} \rvert$		$\lvert T_{supply} \rvert$		$\lvert T_{return} \rvert$	
		Whole day	Pulldown only	Whole day	Pulldown only	Whole day	Pulldown only	Whole day	Pulldown only
1	June 6, 1998	4.1	0.4	21.4	14.7	8.7	1.1	9.4	1.7
2	June 10, 1998	5.0	0.4	16.7	12.8	6.2	0.7	5.4	1.2

Appendix 6. Evaluation of flow hood measurements of residential register flows

Eight flowhoods were used in a new single story house in Sacramento CA. The house had 9 supply registers and a single return. The following discussion will concentrate on the supply measurements because none of these flowhoods was large enough to cover the large return grille and correctly measure the return flow. The registers were either high on the wall or in the ceiling. All the ducts in this house were in the attic.

Two of the flowhoods were powered flowhoods that used the flowhood as a flow capture device and measured the flow with a separate flowmeter. The two powered flowhoods used a fan to compensate for the insertion loss of the flow capture hood and the flow measurement device. The compensation is accomplished by balancing the pressure between inside the flowhood and the room so that the flow out of the register is not reduced by placing the flowhood over the register. To test the sensitivity of the powered flowhoods to the setting of the balancing pressure, the tests were repeated with different measurements of balancing pressure. The LBNL fan assisted flow hood was tested in two modes:

1. Balancing the room and flowhood static pressure with flowhood static pressure measured in the corner of the flowhood up against the wall – i.e., as distant from the bulk flow field as possible.
2. Balancing the flowhood total pressure with room static pressure. The flowhood total pressure was measured using the pressure sampling array normally used for direct flowhood measurements.

The Energy Conservatory flowhood was also tested on two modes:

1. Balancing the flowhood and room static pressures with static pressure measured in the corner of the flow capture hood near the wall (the same as the first LBNL test)
2. With the flow capture hood static pressure measured at the other end of the flow capture hood near the fan/flowmeter entrance.

The other six flowhoods were from four manufacturers. The following table summarizes the important information about each flowhood tested.

Table A6.1 Flowhood characteristics

Code for flowhood	**Description/characteristics**
Hood1	Hard glass fiber capture hood with propeller flow measurement (manufacturer A)
Hood2	Standard flow hood with ΔP flow measurement (manufacturer B)
Hood3	Small flow hood with ΔP flow measurement – specifically for low residential flows (manufacturer B)
Hood4	Standard flow hood with ΔP flow measurement (manufacturer C)
Hood5	Standard flow hood with ΔP flow measurement (manufacturer D)
Hood6	Standard flow hood with ΔP flow measurement (manufacturer D – same model as Hood5, but different serial number)
Hood7	LBNL fan assisted flow hood – total hood pressure balance
Hood8	LBNL fan assisted flow hood – static hood pressure balance
Hood9	Energy Conservatory fan assisted flowhood – corner hood pressure balance
Hood10	Energy Conservatory fan assisted flowhood – near fan entry hood pressure balance

The following table summarizes the individual register measurements for each of these flowhoods:

Table A6.2 Comparison of flowhood measurements of supply registers (cfm)										
Register	Hood 1	Hood 2	Hood 3	Hood 4	Hood 5	Hood 6	Hood7	Hood 8	Hood 9	Hood 10
1	184	194	182	205	240	226	197	194	197	203
2	167	181	163	173	232	238	169	186	173	175
3	234	243	275	265	370	341	239	246	244	250
4	156	162	158	164	233	248	151	154	149	164
5	68	69	80	86	110	122	75	75	72	72
6	56	52	55	60	91	66	58	65	53	53
7	103	84	93	102	153	157	98	96	90	95
8	54	43	53	43	83	84	49	53	45	46
9	51	43	52	60	87	82	44	52	47	46
sum	**1073**	**1071**	**1111**	**1158**	**1599**	**1564**	**1080**	**1121**	**1070**	**1104**

These results show that some flowhoods (4, 5 and 6) can give substantially different results from the others. In particular, Hoods 5 and 6 that are from the same manufacturer give flows that are much too high. There are also some significant differences on a register by register basis. If we take Hood 7 to be our measurement standard for comparison purposes, all the flowhoods except Hood 9 have differences for an individual register of 13 cfm or greater. This magnitude of difference may be a concern if the flowhood measurements are used to verify ACCA designs, for example. Another key result is that register 9 was in an interior bathroom and the duct design called for only 5 cfm at this register. Measuring flows this small is very difficult with existing portable flow measurement equipment and would be very difficult to verify. However, in a real duct system it is almost impossible to install it to get this low a flow and, as our results show, the actual flow out of the register is substantially higher than its design flow. This case shows that some interpretation of required design flows is required (for field test verification and compliance testing) because it is probably better to allow the higher than design flow in this case, rather than attempt to restrict the flow to the design value.

Static vs. total pressure balancing.

Comparing the two LNBL tests (Hood 7 – total and Hood 8 -static) shows that the static pressure balancing results in consistently (7 out of 9 registers) lower flow measurements. However, the differences are quite small – less than the specified flow measurement uncertainty (±3% of flow) for 5 of the 9 registers and so this difference is not very significant.

Changing balancing pressure measurement location

Comparing the Hood 9 and Hood 10 results shows that measurement location is not critical. The differences between the two tests are less than the flow meter measurement uncertainty (±3% of flow) for all but two of the registers.

Comparing return measurements

Due to limited capacity not all of the flowhoods had the capability to measure the return flows. Table A6.3 compares the results for the six flow hoods that were used to measure the return flow. These results show good agreement between hoods 4 through 8, but hood 3 gave results that were too low. For hood 3, the return flow measurement was at the upper limit of its measurement range. This flowhood added significant flow restriction to the return, resulting in a lower flow through the register and flowhood.

Table A 6.3 Comparison of flowhood measurements of return register flow (cfm)					
Hood 3	Hood 4	Hood 5	Hood 6	Hood 7	Hood 8
860	995	1028	1055	1037	1057