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Journal
ASHRAE Transactions, 104(1B)

Author
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Publication Date
1998

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LBNL-40587
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Environmental Energy Technologies Division

January 1998

Submitted to
ASHRAE Transactions



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This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy of the U.S. Department of Energy, Office of Building Technologies, Existing Buildings Energy Research Program and Building Equipment Program, under contract No. DE-AC03-76F00098

LBNL 40587
ASHRAE Transactions 1998 Vol. 104 Part 1

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Abstract

Measurements on three gas and two electric furnaces have been made to examine the field performance of these furnaces and their interactions with their forced-air distribution systems. The distribution systems were retrofitted as part of this study and the impact of retrofitting on furnace performance is discussed. In addition to field measurements, this paper will discuss how forced-air furnace systems are treated in proposed ASHRAE Standard 152P, and applies the resulting equations to the systems tested in the field. The distribution system calculations in Standard 152P are compared to the current methods employed in the "Furnaces" chapter of ASHRAE's *HVAC Systems and Equipment Handbook*, showing how the distribution system efficiencies calculated using Standard 152P can be incorporated into the handbook.

Key Words

Furnace, standard, air leakage, conduction, duct system, measurement.

Field measurements of interactions between furnaces and forced air distribution systems

Abstract

Measurements on three gas and two electric furnaces have been made to examine the field performance of these furnaces and their interactions with their forced-air distribution systems. The distribution systems were retrofitted as part of this study and the impact of retrofitting on furnace performance is discussed. In addition to field measurements, this paper will discuss how forced-air furnace systems are treated in proposed ASHRAE Standard 152P, and applies the resulting equations to the systems tested in the field. The distribution system calculations in Standard 152P are compared to the current methods employed in the "Furnaces" chapter of ASHRAE's *HVAC Systems and Equipment Handbook*, showing how the distribution system efficiencies calculated using Standard 152P can be incorporated into the handbook.

1 Introduction

The heating system for a house is a combination of a piece of equipment that provides the heating energy (the furnace, boiler or heat pump) and the method used to distribute this energy throughout the house (forced air ducts, hot water radiators, radiant panels etc.). The overall performance on the heating system depends on the interactions between the equipment and the distribution system. For example, an overly restrictive forced air duct system can result in too low a flow over a furnace heat exchanger resulting in the furnace cycling on the high limit switch rather than controlling the indoor temperature. This short cycling increases cyclic losses from the duct system, changes heat exchanger effectiveness and reliability and therefore changes the overall system performance from that intended by the equipment designer. Any increase in distribution system losses or decreases in furnace efficiency increase the energy required to heat the house, resulting in higher energy bills for the homeowner and increased peak demand for utilities. In addition to the energy cost, the desired comfort in the home may not be achieved by a poorly performing system.

In this paper we concentrate on the interaction between forced air duct systems and furnaces. Three gas and two electric furnaces were measured as part of a study to determine the effect of retrofitting duct systems on the duct system and the furnace performance. The retrofit consisted of adding extra insulation to the exterior of the ducts (added insulation was foil backed 50 mm (2 in) thick, nominally RSI 1 (R-6)) and using metal-foil-backed butyl tape and mastic to seal duct leaks. The houses were located in Sacramento, CA. In addition to the field measurements, this paper gives an outline of the forced air furnace sections of proposed ASHRAE standard 152P - "Method of Test For Determining the Steady-State and Seasonal Efficiencies of Residential Thermal Distribution Systems". The calculation methods for the standard are compared to existing procedures in 1996 ASHRAE Systems and Equipment Handbook, Chapter 28.

In addition, the delivery effectiveness of the distribution system calculated using the standard is compared to measured data.

2 Field Measurements

The field tests were used to monitor the system performance by measuring the energy consumed by the system, the energy put into the duct system at the heat exchanger, and the energy delivered to the house through the registers. These field measurements did not take into account losses from the duct system that go into the conditioned space rather than to outside, or changes in the energy lost from the building as a result of duct losses changing the temperature of buffer zones such as attics or crawl spaces.
The field measurements were performed in two parts:

1. Diagnostic tests of building and duct system characteristics.
2. Approximately two weeks of monitoring of characteristic temperatures, weather and HVAC power consumption in both pre- and post-retrofit periods.

More detailed descriptions of the field tests can be found in Jump and Modera (1994), and Jump, Walker and Modera (1996). The following is an overview of the test procedures in order to provide a context for the experimental results.

2.1 Diagnostic Tests

The following measurements were performed for the diagnostic testing:

1. House pressurization test to determine exterior envelope leakage.
2. Register air flows.
3. Fan flow.
4. Air leakage flow. This was determined by pressurizing the ducts to 25 Pa (0.1 inch water) and measuring the leakage flow. This is not a direct measurement of leakage flows at operating conditions, but does indicate the changes due to the retrofits.
5. Duct system characteristics: number and location of registers, duct location, duct shape (round, rectangular), duct material (flex duct, sheet metal or duct board), diameter and length of ducts and air handler location.
6. Equipment characteristics: heating/cooling capacity, location within the building.
7. House characteristics: number of stories, floor plan.

2.2 Two week measurements

Measurements were made for two weeks both pre- and post-retrofit to capture changing weather conditions and system cycling effects.

1. Register air temperatures: Used together with the measured register air flows to calculate energy supplied to the house.
2. Plenum air temperatures: Used together with measured fan air flows to calculate energy output by the furnace and input to the ducts.
3. Ambient air temperatures: outside air temperature and the temperature of air surrounding the ducts.
4. Energy consumed by equipment: electrical power consumed by the air handlers and electric furnaces, and natural gas consumed by gas furnaces. The gas furnace energy

consumption was determined by an electronic flow transducer, calibrated to the gas meter at the house and using the heating energy of the gas obtained from the utility.

2.3 Data Binning Procedure and Cyclic Analysis

The measured field data was analyzed for each system cycle. The cycle time was defined as the period of time from when the equipment switched on to the next time the equipment switched on. Figure 1 illustrates how the cycle length was determined. For each cycle, the power consumed by the equipment was integrated to obtain the system energy consumption. In addition, the measured air flow rates through the fan and registers were combined with the measured temperatures to determine the energy delivered by the equipment to the duct system and the energy delivered by the registers to the rooms. The energy delivered to the ducts and the rooms was integrated whilst the fan was on to determine the cyclic energy flows. With this information the equipment and delivery efficiencies were determined for each cycle.

The cyclic data were then sorted into bins using the measured outdoor temperatures. The bins were 2°C wide and are represented by their middle temperature, i.e., the 20°C bin represents all temperatures between 19° and 21° C. The results for all the cycles falling into each bin were averaged. By breaking down the cyclic averaging results into data bins covering a wide range of temperatures, we were able to separately track variations due to weather conditions and the effect of the retrofits.

For each cycle, the following delivery effectiveness, equipment efficiency and duct losses were calculated:

Delivery effectiveness (DE) is the ratio of energy supplied to the conditioned space through the registers to the energy input to the duct system from the equipment. Note that the energy supplied to the conditioned space is the net energy and includes energy removed by the return side of the system (i.e., it is not just the energy in the air coming out of the supply registers). Equipment efficiency (η_{equip}) is the ratio of energy supplied to the duct system to the energy consumed by the equipment (including fan power). The energy lost due to supply leaks was estimated by assuming that all the leaks are at the plenum[1]. The energy lost due to supply conduction was calculated from the change in temperature between the supply plenum and the registers.

Because the temperature of air leaking into the return ducts was generally unknown, the return leakage and conduction losses are combined into a single term for fractional return losses, such that the total losses plus the energy delivered to the conditioned space by the duct system add up to the energy supplied to the duct system:

2.4 Results

Tables 1 and 2 summarize the house and system specifications, system fan flows and leakage flows. The houses are labeled E1 and E2 for the electric furnaces and G1 through

[1] Jump et al. (1996) showed that the assumption that all supply leaks are at the plenum did not have a large impact on the split between supply leakage and conduction losses.

G3 for the gas furnaces. The duct systems are a mix of flex duct and sheet metal with one system having a platform return under the air handler. A platform return is simply a box with a register connected directly to the air handler cabinet so there is no "duct" system. The majority of the ducts and the air handlers and heating/cooling equipment are in the attics, which is typical of California construction.

The leakage flows were reduced by 65% for supplies and 80% for returns (averaged over the five systems). The leakage was not completely eliminated because some leaks are unreachable for application of tape or mastic. However, these are significant leakage reductions that will have a significant impact on duct system losses (shown later).

The system flows were the same on average before and after retrofitting, with some systems having increased flow and others having reduced flow. When leaks are sealed it is expected that the fan flow will be reduced due to increased system flow resistance. The unchanged fan flows are a result of these test houses having small reductions in leakage. The fan flows may also be insensitive to the change in pressure across the fan, but without explicit fan curves this was difficult to verify. In addition, the changes in fan flow are close to the resolution of the measurement methods and so it would be difficult to see any trend in reduced fan flows due to the duct retrofits.

Table 2 compares the manufacturers "name plate" input capacities to the measured energy consumption. The electric furnaces have measured electricity consumption that closely matches the manufacturer's specification. All the gas furnaces consume considerably less energy than the manufacturer's specification (by an average of 6 kW (18 kBtu/hour) which is equivalent to 27% of the rated output). No apparent reason was found for this result. The measured outputs of the furnaces shows that these furnace energy consumption values are correct for these systems because if the furnaces consumed the energy specified by the manufacturer, then we would see very poor equipment efficiencies in Table 3.

Table 3 summarizes the results of the two week long tests. The data in Table 3 have been chosen so that the pre and post retrofit data for each house are for the same outdoor temperature because the system performance (duct losses, systems loads etc.) will be different at other outdoor temperatures. This allows the comparison of system performance pre and post retrofit. The indoor-outdoor temperature differences are within 0.1°C for pre and post retrofit data, except for G2 where the indoor temperature was 1°C higher for the post retrofit data. The particular results in Table 3 were chosen so as to maximize the number of cycles both pre and post retrofit. The delivery effectiveness, fractional supply and fractional return losses are expressed as a percent of the energy delivered to the duct system (output from the equipment).

The number of cycles is the total of both pre and post retrofit. There are few cycles for E2 due to the mild weather experienced during testing (the outside temperature was 14°C (57 °F)), but G2 had over 150 cycles (mostly due to the furnace cycling on the high limit switch).

The equipment efficiencies for the electric furnaces are lower than expected at 75% and 90% for the two systems tested here (at steady-state an electric furnace should be 100% efficient minus some cabinet losses). These low measured efficiencies are due to several possible factors:

- The system cycles are short (as shown in Table 4) and so the dynamic losses associated with heating and cooling the furnace and air handler for every cycle are a large fraction of the total energy consumed by the furnace. Figure 2 illustrates the short cycling of the furnace in E1. The rapidly changing supply plenum air temperatures show that these cycles are less than steady-state and the system is always in its initial heating transient for the whole cycle.
- Both of these systems were located in the attic and so were in cold surroundings that lead to increased energy losses from the equipment cabinets from both conduction and cold air sucked in through leaks in the fan cabinet and/or plenum.
- Some locations in the supply plenum could have higher and lower air temperatures than those at the four measurement locations we used. If all four measurements were too low then the calculated equipment output would be too low resulting in reduced equipment efficiency.
- Similarly, the return plenum air temperatures may be too high. For example, system E1 had return plenum average temperature of 22.6°C (73°F), compared to an indoor air temperature of 18.2°C (65°F). If we assume that the measured return plenum air temperature is too high and use the measured indoor temperature instead, then we obtain an equipment efficiency of 91% rather than 75%. The measured return plenum temperature may be too high because we only measure at a single location and others may be lower, or the there was inadequate radiation shielding on the temperature measuring probe and the probe was heated by radiation from the heating coils.

The above comments show that further research is needed on field measurements of furnace energy consumption and measuring the output of furnaces. Improved methods of measuring the temperature rise across furnace heat exchangers for field measurements would reduce some of the uncertainties discussed in the above summary.

Both electric furnaces showed a small drop in equipment efficiency after the retrofits. In E2, the drop is negligible, however, E1 has a 4% change. This is most likely because the attic temperatures post retrofit were cooler (by 3°C (6°F)) in the attic of E1, thus increasing the cabinet losses. The changes in attic temperatures from the pre to post retrofit cases are due to a combination of different solar gains, ventilation rates, temperature histories and duct system losses for the attics even thought the outside temperatures are the same. (Table 5 shows that this can lead to either increases or decreases in attic temperatures). In addition, Table 4 shows that the average ontime decreases significantly post retrofit for these two systems. The shorter cycle is due to reduced duct losses - it takes less time for the house to reach its setpoint because more energy is delivered to the conditioned space. The downside of reduced ontime is that the cyclic losses due to heating up the ducts and equipment are increased (as a fraction of the

total energy consumed during a cycle). Therefore a greater fraction of each cycle is spent increasing the temperature of the equipment post retrofit. This leads to the initial transient in measured supply plenum air temperatures being a greater fraction of the total measured temperature difference across the furnace, thus reducing the cyclic average supply plenum air temperature and therefore reducing the overall equipment efficiency.

In Table 3, data is not presented for G3 post retrofit due to sensor malfunctions, and the pre/post comparisons for gas furnaces are only for systems G1 and G2. The equipment efficiencies for G1 and G2 show increases of 1% and 5%, respectively. This is opposite to the reduction in efficiency for the electric furnaces. Unlike the attic in E1, the attic temperatures for the gas furnaces changed less than 0.5°C (1°F) from pre to post retrofit, so the change in equipment performance is not due to a change in the ambient conditions.

The delivery effectiveness increased from 60% to 70% on average because of the reduced leakage and conduction losses. In other words, 10% more of the energy put into the duct system was delivered by the registers to the conditioned space. To think of this another way, this is roughly equivalent to adding another register's worth of energy output to these systems. The post retrofit delivery effectiveness is still low because the ducts are still located outside the conditioned space and the potential for large losses still exists. In addition, some of the leakage and conduction losses will be to the conditioned space, thus increasing the total energy delivered to the conditioned space (the delivery effectiveness values in Table 3 only include energy flows through the registers). These "regained" losses are not included in this analysis because they are extremely difficult to measure, requiring coheat testing in addition to all the tests performed in the current study (see Sonderegger et al. (1980) for a description of coheating).

The fractional supply leak and supply conduction results in Table 3 show that the major factor in increasing the delivery effectiveness was the reduction in leakage. The conduction losses remain the same post retrofit. The conduction losses do not change because the temperature of the air in the duct systems was increased in most of the systems after the retrofit, which counterbalanced the effect of increased insulation. Note that without the additional insulation from the retrofit, the conduction losses would have increased after sealing the leaks in the retrofit.

The return losses depend on the attic temperatures (this is the temperature of the air drawn into the return leaks). G2, G3 and E2 had the warmest attics (and the warmest outdoor temperatures) with E2 having an attic temperature of 19°C (66°F) post retrofit. These warm attics lead to negative return losses, i.e., the net heat transfer for the returns was from the attic to the duct. The cooler attics for E1 and G1 show positive return losses because their attics are much cooler, e.g., the temperature for the attic in E1 is only 7°C (45°F) both pre and post retrofit.

Table 4 summarizes the average ontime (the length of time the system is on for in each cycle) and the average fractional ontime. The ontime divided by the total cycle time, where the total cycle time is from when one cycle begins to when the next cycle begins,

shown in Figure 1. These results show that when the systems were on, they were on for the same length of time (the ontime did not change), but the systems operated for less time (the fractional ontime went down). The reduction in fractional ontime is due to the systems loosing less energy from the ducts and so less equipment operation is required to meet the building load. Note that these values do not reflect design or seasonal (or annual) conditions, but are at a particular outdoor condition (as given in Table 3). The change in ontime also reflects the reduction in building load due to reduced infiltration loads because of reduced duct leakage - both with the system on and off.

3 Forced air furnace systems in proposed ASHRAE standard 152P : "Method of Test For Determining the Steady-State and Seasonal Efficiencies of Residential Thermal Distribution Systems"

The objective of this method of test is to provide estimates of the efficiency of thermal distribution systems. This efficiency may be used in energy consumption or system capacity estimates. This method of test provides thermal distribution system efficiencies for both heating and cooling systems. In addition, thermal distribution system efficiency is calculated for seasonal conditions (for energy consumption) or design conditions (for system sizing). In the following discussion we will look at the forced air heating sections of the standard only. The comparison of 152P predictions to the measured field data will be done using the delivery effectiveness calculated in 152P and not the distribution system efficiency. The delivery effectiveness is intended to be an interim calculated value in 152P because it does not include regain of duct losses, equipment impacts or the effect on infiltration load due to duct leakage imbalances. However, because the measured data is based on flows and temperatures at registers, it does not include these effects. The delivery effectiveness values based on the 152P procedure were calculated using Equation 11.

3.1 Application of diagnostic, building plan, and default input parameters

152P has three options for determining input parameters used in the distribution system efficiency calculations:

- Diagnostic values of input parameters shall be used in existing buildings or buildings under construction (before the envelope is complete).
- Building plan values of input parameters shall be used for buildings prior to construction.
- Default values of input parameters shall be used where they are unavailable from diagnostic tests or building plans.

3.2 152P Procedure

The calculation procedure for distribution system efficiencies is based on six principal input parameters: climate, duct location, duct leakage, duct insulation, duct surface area and system fan flow.

The following additional parameters are also used in the calculation procedure: building volume, building floor area, venting condition of attics and/or crawlspaces, insulation in all parts of the building structure, number of stories, number of return registers, ACCA

Manual D (ACCA (1997)) system fan flow, and manufacturers specification of system fan flow.

These input parameters are determined using the diagnostic (for existing buildings), building plan (for unbuilt buildings) and default value procedures described in the proposed standard. In this paper the measured (diagnostic) values were used as input.

3.3 152P Input

3.3.1 Diagnostic Input

The diagnostic inputs are determined for existing buildings and new construction where the building envelope and air handling and conditioning equipment installation is complete. The diagnostics include observation of various duct characteristics and measurement of duct leakage and system fan flows. The following diagnostic procedures are used:

- measure duct system leakage separately for supply and return.
- measure system fan flow.
- measure the duct system and calculate the area of the ducts outside the conditioned space for supply (A_s) and return (A_r) and the total duct areas ($A_{s,total}$ and $A_{r,total}$). Note that these areas include the areas of the plenums.
- note the location of ducts outside the conditioned space (attic, crawlspace, garage etc.). This location determines the ambient temperature conditions to which the ducts are exposed.
- note the insulation level for the supply (R_s) and return (R_r) ducts outside the conditioned space. If the insulation level is not the same for all the ducts, then an area weighted average for each duct location is used.
- sketch the floor plan, including dimensions. The volume of the building (V) shall be calculated from these measurements.

3.3.2 Building Plan Input

Using input from the building plan provides distribution system efficiency values associated with the design of the distribution system. It allows the system to be evaluated before construction is completed. The efficiency of the distribution system is based upon what is written into the builder's plans for that distribution system. Those plans can include a specification for testing during construction, e.g., duct leakage testing by fan pressurization. If specifications for the distribution system are not in the plans, or are not specific enough within the plans, the distribution-system efficiency is based upon default values for the various parameters influencing that efficiency. The building plan specification must include: duct location, duct leakage, duct insulation, duct surface area, and system fan flow. Due to the uncertainty of the construction process, the specified values of these parameters must be tested during or after construction in order to apply the efficiencies calculated using the standard.

3.3.3 Default Input

When input values are unavailable from diagnostic tests or from building plans the default values provided in the standard are used.

3.4 Duct Location

Duct location determines the external temperature for duct conduction losses, the enthalpy of the air drawn into return leaks, and the regain of duct losses.

3.5 Climate and Duct Ambient Conditions for ducts outside conditioned space

The temperature conditions for different duct locations are obtained from a table of default values. These values are derived from a combination of field measurements (see Forest and Walker (1993) and Parker (1997) for attic temperatures) or simple energy balances. The heating dry bulb design temperatures (and mean coincident wet bulb temperatures) are taken from ASHRAE Fundamentals Handbook (1993), Chapter 24, using the 97.5% design values. The seasonal outdoor dry bulb temperature is design temperature plus 9°C (16°F).

The temperatures of the duct zones outside the conditioned space are determined in for design and seasonal conditions and for heating and cooling. If the ducts are not all in the same location, the duct ambient temperature for use in the delivery effectiveness calculations are determined using a weighted average of the duct zone temperatures:

3.6 Duct Wall Thermal Resistance

The diagnostic value of duct wall thermal resistance is determined by direct observation of the ducts. If the ducts have visible manufacturer's specification of duct wall thermal resistance, then this specification is used. If the ducts are unmarked, then the insulation thickness is measured and the type of insulation (e.g., glass fiber) noted. The duct wall thermal resistance is then calculated based on the insulation type and thickness using calculation procedures from ASHRAE Fundamentals (1993), Chapter 23, or equivalent. Air film resistance of 0.25 [Km^2/W] (1 [h ft^2 °F/BTU]) is added to the insulation thermal resistance to account for external and internal film resistance.

Duct-system thermal resistance may also be taken from building plans (using manufacturers specifications) or the observed value from diagnostic testing.

3.7 Duct Surface Area

The duct surface includes the surface area of plenums, and the supply and return duct surface area are calculated separately. If the supply or return ductwork is in more than one location, the area of the ductwork in each location is calculated separately. Duct surface area is calculated from measured duct lengths and external diameters (for round ducts) or perimeters (for rectangular ducts).

Default duct surface areas are based on the results of field tests (Jump, Walker and Modera (1996) and Andrews (1996)). The surface area of the ductwork is determined based on the conditioned floor area of the building.

3.8. System Fan Flow

The diagnostic system fan flow is determined by measuring the flow through the system with a fan and a flowmeter. The system operating point is determined by the pressure difference between the supply plenum and the conditioned space

If the duct system is designed according to ACCA Manual D, it is assumed that the flow across the heat exchanger is equal to that specified by the equipment manufacturer. If no duct-system layout and design calculations are provided, the fan flow is 85% of the manufacturer's specification.

The default system fan flow is 0.003 m^3/s per m^2 building floor area [0.6 cfm/ft^2 building floor area]. This default value is based upon field measurements of system fan flows (e.g., see Jump, Walker and Modera, 1996).

3.9 Duct Leakage Flow Rate

Diagnostic supply and return leakage flows, Q_s and Q_r, are determined using either a fan pressurization test in or a house pressurization test. If Q_s and Q_r are specified in building plans they are verified using a simplified duct pressurization procedure.

The default total leakage is 17% of the total fan flow for both supplies and returns. These defaults are based on the results of numerous field measurements (e.g., Jump, Walker and Modera (1996), Downey and Proctor (1994), Modera and Wilcox (1995), Cummings, Tooley and Dunsmore (1990) and Modera and Jump (1995)). As a default, the fraction of this total leakage that is to outside is the same as the ratio of duct area outside the conditioned space to total duct area. However the default assumes that the minimum amount of duct outside (for the leaks) is one half of the total.

3.10 Duct Thermal Mass

The thermal mass of the duct system (this includes the ducts, system cabinet and heat exchanger if they are outside conditioned space) has the effect of trapping energy used to heat the ducts outside the conditioned space when the system turns off. At the beginning of the system cycle, the ducts are warmed and the temperature of air leaving the supplies will be cooler. Thus the furnace output is split between heating the ducts and heating the conditioned space. When the system fan turns off the remaining heat in ducts outside the conditioned space is lost (some of this is regained by having the fan remain on after the gas burners or electric heating elements are turned off). An estimate of how much energy is left behind in the duct system after the fan has turned off was made using computer simulations by Modera and Treidler (1995). They determined the fraction of energy put into the ducts during the cycle that was lost due to these thermal mass effects (the cyclic loss factor). Typical results indicated a cyclic loss factor for sheet metal ducts is 0.05, and is 0.02 for non-metallic ducts.

3.11 Delivery effectiveness calculations

The steady state delivery effectiveness calculations are based on a simplified model of duct systems developed by Palmiter and Francisco (1996). The supply and return conduction fractions, B_s and B_r, are calculated as follows:

For SI:

$$B_s = \exp\left(\frac{-A_s}{Q_e \rho_{in} Cp R_s}\right) \quad (1)$$

$$B_r = \exp\left(\frac{-A_r}{Q_e \rho_{in} Cp R_r}\right) \quad (2)$$

For IP:

$$B_s = \exp\left(\frac{-A_s}{60 Q_e \rho_{in} Cp R_s}\right) \quad (3)$$

$$B_r = \exp\left(\frac{-A_r}{60 Q_e \rho_{in} Cp R_r}\right) \quad (4)$$

where: A_s is the supply duct surface area, A_r is the return duct surface area, R_s is the thermal resistance of the supply duct insulation, R_r is the thermal resistance of the return duct insulation, ρ_{in} is the density of indoor air and Cp is the specific heat of air and Q_e is the system fan flow.

The duct leakage factors for the supply and return ducts are calculated using:

$$a_s = \frac{Q_e - Q_s}{Q_e} \quad (5)$$

$$a_r = \frac{Q_e - Q_r}{Q_e} \quad (6)$$

The temperature rise across heat exchanger, Δt_e, is calculated based on the capacity of the equipment, E_{cap}, and the system air flow, Q_e.

For SI:

$$\Delta t_e = \frac{E_{cap}}{Q_e \rho_{in} Cp} \quad (7)$$

For IP:

$$\Delta t_e = \frac{E_{cap}}{60 Q_e \rho_{in} Cp} \quad (8)$$

The difference between the inside (t_{in}) and the ambient temperature surrounding the supply ($t_{amb,s,}$) and return ($t_{amb,r}$), is calculated as follows:

$$\Delta t_s = t_{in} - t_{amb,s} \quad (9)$$

$$\Delta t_r = t_{in} - t_{amb,r} \quad (10)$$

The steady-state delivery effectiveness is calculated using:

$$DE = a_s B_s - a_s B_s (1 - B_r a_r)\frac{\Delta t_r}{\Delta t_e} - a_s (1 - B_s)\frac{\Delta t_s}{\Delta t_e} \quad (11)$$

3.12 Distribution System Efficiency

The distribution system efficiency takes the steady-state delivery effectiveness and includes changes in equipment efficiency, duct system and building envelope load interactions, and thermal regain of duct losses that are not directly to outside (e.g., losses to attic spaces). The measurements discussed here do not include these factors and therefore the comparisons are made between measured data and the delivery effectiveness calculated using Equation 12.

4. Comparison of measured delivery effectiveness to those calculated using ASHRAE 152P

The inputs to the standard 152P calculations were determined as follows:

- Outdoor temperatures – from field data
- Attic temperatures – from 152P table for design conditions (outdoor temperature + 5° C (9°F))
- Duct location - determined from direct field observation
- Fan flow and leakage flows – from field measurements. The duct leakage used in for this comparison is the total duct leakage under operating conditions (measured using the duct pressurization procedures given in 152P). This is because the measured delivery effectiveness only includes air flows from the registers and does not include any duct leakage that is to the conditioned space. Tests were also done to estimate the fraction of the total duct leakage that was to outside. Using leakage to outside results in calculated 152P delivery effectiveness that is 3-4% higher than the values given later.
- Insulation value – from field observation
- Duct surface area – from field measurements
- System capacity – from field measurements
- Indoor temperature - from field measurements

Because the actual attic temperatures were measured for this study, it is possible to compare the predictions of the standard to the measurements to see how well the standard predicts attic temperatures for these houses. Table 5 summarizes the measured and predicted (by the standard) attic temperatures. The results in Table 5 indicate that the default attic temperature calculation in the standard is surprisingly good (for such a simple method) and is within 1°C (2°F) of the measured values on average. The worst

case has a difference of 4°C (7°F).

The predicted 152P delivery effectiveness values are given in Table 6. Comparing these results to the measured results in Table 4 shows that the 152P calculations tend to overpredict the delivery effectiveness by an average of 8%. The average difference between measurements and predictions is about 11%. These significant differences are due to:

1. the simplified nature of the 152P calculation procedure that does not account for all the details of each individual duct installation. For example, all of the supply duct is assumed to be exposed to the same temperature and to have the same thermal resistance. However, in real duct installations, the plenums may be uninsulated and the proportionally higher air temperatures in the plenum lead to a real system having higher conduction losses than the simplified system in the 152P calculation.
2. measurement uncertainties. Possible sources of measurement uncertainties have been discussed earlier.
3. differences between measurement conditions and calculation conditions. Perhaps the most significant issue for the comparison of predictions to measurements is that the measured data are for systems that are cycling, but the calculated delivery effectiveness is for steady-state and does not include any cyclic losses. If the duct thermal mass correction (discussed earlier in section 3.10) of 2% to 5% is subtracted from the 152P delivery effectiveness, then the bias would be reduced by a similar amount. Using the 5% reduction (Figure 1 and Table 4 show that the systems studied here tended to have relatively short cycles, with the systems operating far from steady-state) would reduce the 152P bias to only 3%, a much more acceptable figure.

The bias is larger (11%) for the post retrofit data than for the pre retrofit data (5%). The source of the larger post retrofit bias is that the improvement in the inputs used to calculate the delivery effectiveness is not reflected in increased measured delivery effectiveness. The R value of the duct insulation was typically R2 (RSI 0.3) for the pre retrofit case and improved to R8 (RSI 1.3) for the post retrofit case. Combined with an increase in exterior duct surface area of about 40% (a change in diameter from 10" (150 mm) to 14" (210 mm)) due to the increased insulation thickness, we would expect approximately a factor of three reduction in conduction losses. Combined with the factor of two reduction in duct leakage, we would expect to see the improvements indicted in the 152P calculation results, rather than the lesser changes in the measured data. The probable explanation for this result is that cyclic losses (not included in the steady-state 152P calculations) are significant. The leakage reduction and improved insulation may well reduce the duct system losses to the extent indicated by the 152P predictions if the systems were operating at steady-state. It is possible that the conduction and leakage losses could have been reduced by the amount indicated by the 152P results, but the cyclic losses may have changed little. If the cyclic losses are significant and unchanged, then the change in measured delivery effectiveness will not be as great as expected. Note that the cyclic losses did not change much according to Table 4 and other observations of cyclic behavior, e.g., the performance shown in Figure 2 was the same pre and post retrofit.

5. Comparison of Proposed ASHRAE standard 152P to existing distribution system analysis in 1996 ASHRAE Systems and Equipment Handbook Chapter 28

Appendix 1 contains a detailed comparison between the proposed ASHRAE standard 152P and the 1996 ASHRAE Systems and Equipment Handbook. The critical difference between the two approaches is that the handbook lists many factors which require very detailed measured data or a computer simulation model, such as ASHRAE SP43, whereas the standard gives simple algebraic equations, look-up tables, or default values to estimate the parameters required to make estimates of the system performance factors. There are several major differences between the proposed standard and distribution system parameters in the handbook, as follows:

- The handbook defines all the duct system performance parameters on an annual basis. In proposed standard 152P the duct performance is determined for design conditions (for system sizing) and for seasonal conditions (for seasonal energy use). The seasonal and annual performance should be roughly equivalent despite being defined differently.
- The equipment efficiency is not referred to directly in standard 152P, however changes in equipment efficiency due to changing delivery system performance are included. For furnaces these are not large effects. Typical effects of poor duct systems are longer ontimes and greater fractional ontime. Heat pumps and air conditioners are more sensitive to these changes in the cyclic behavior of the system, particularly for multi capacity systems that are forced to operate at high capacity longer with poor duct systems.
- The duct efficiency in the handbook is almost the same as the delivery effectiveness in the proposed standard. The difference is that the handbook duct efficiency only includes energy flows directly out of registers whereas the delivery effectiveness in the proposed standard includes leakage losses to the conditioned space (these losses to conditioned space would have to be included in the miscellaneous gain factor in the handbook.). For example a leaky duct system with all the ducts inside conditioned space would have low duct handbook efficiency, but high 152P delivery effectiveness.
- The induced load factor in the handbook includes cyclic effects and losses to unconditioned space, however it does not (nor does any other factor in the handbook) account for changes in infiltration rate due to return and supply duct leakage imbalances. It does account for extra infiltration due to combustion and open flues when the system is off. In the proposed standard, there is an infiltration load factor to account for the supply-return leakage imbalance. The proposed standard assumes that combustion induced flows or flows through open flues are included in baseline infiltration calculations and therefore they are not in the infiltration load factor in the standard. The standard also has a separate factor to account for changes in losses to unconditioned spaces within the house (e.g., attics, crawlspaces and garages).
- The final numbers produced by the two procedures are - system index (from the handbook) and distribution system efficiency (from the standard). The differences in these two numbers arise from factors that are included in one but not the other, e.g., infiltration changes due to leakage imbalance is included in the standard, but not in the handbook.

6 Summary

The field measurements have shown how furnace performance is highly variable and depends on the complex interaction of the duct system and ambient conditions. These measurements illustrate that the furnace performance will be different from its rated performance unless the furnace is operating under the same conditions it was rated at. For example, the gas furnaces tested here had 27% less energy consumption than the manufacturers rated input. The pre and post retrofit measurements showed that the energy losses from the ducts were reduced by 10% of furnace output, mostly due to the reduction in leakage. The retrofits did not effect the furnace performance on average. However both the delivery and equipment efficiency changes were significantly variable from house to house indicating the complex nature of the system interactions with the building and their environment.

The 152P calculation procedure illustrates how many factors must be accounted for, even with a much simplified calculation procedure. The procedure showed how the proposed standard calculates seasonal and design system performance and how all the input parameters are determined either by diagnostic measurement, design in building plans or from default values provided in the standard (based on field measurements or simplified calculations).

The comparison of 152P predicted delivery effectiveness to the measured values has shown that 152P provides adequate estimates on average (an average of 8% difference – reduced by including a cyclic loss factor estimate to 3%). However, for individual systems, the simplifications required for the 152P calculation means that there can be substantial differences between measured and predicted values. In addition, the 152P delivery effectiveness is a steady-state value but the measurements can be dominated by cyclic losses. The significant cyclic behavior of the systems studied here makes the comparison of the measured and predicted results much more difficult. Future work on the proposed standard will have to look in more detail at accounting for cyclic losses.

The 152P calculation procedure has been compared to the existing method in 1996 ASHRAE Systems and Equipment Handbook Chapter 28. The two methods are attempting to estimate the same performance parameters so that energy consumption may be calculated for a house. However, the two methods have many detail differences (e.g., which factors have their own explicit terms and what assumptions are made to simplify the calculation procedures) and have a fundamentally different approach. The handbook lists many factors which require very detailed measurements or the use of a computer model such as ASHRAE SP43, whereas the standard gives simple algebraic equations, look-up tables, or default values to estimate the parameters required to make estimates of the system, performance factors.

7 Acknowledgments

This work was supported by the Assistant Secretary for Conservation and Renewable Energy of the U.S. Department of Energy, Office of Building Technologies, Existing Buildings Research Program, under Contract No. DE-AC03-76SF00098.

The research reported here was funded in part by the California Institute for Energy Efficiency (CIEE), a research unit of the University of California. Publication of research results does not imply CIEE endorsement of or agreement with these findings, nor that of any CIEE sponsor.

This work was also supported by the Sacramento Municipal Utility District, which identified all of the houses, interacted with the homeowners, acquired the retrofit contractors, and paid for the retrofits.

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Appendix 1. Comparison of Existing ASHRAE Definitions to those proposed in ASHRAE 152P

ASHRAE Definitions
1996 ASHRAE Systems and Equipment Handbook Page 28.10

All the following parameters are defined on **annual** basis.

duct output = energy out of registers
duct input = energy into air from heat exchanger (= equipment output)
total energy input = energy consumed by equipment
total heat delivered ≡ space heating load for base case
system induced load = difference between space heating load for a particular case and the space heating load for the base case - this includes infiltration and regain/recovery effects

Equipment-Component efficiency factors

E_F = furnace efficiency

$$= 100\frac{\text{duct input}}{\text{total energy input}}$$

E_D = duct efficiency

$$= 100\frac{\text{duct output}}{\text{duct input}}$$

Equipment-System performance factors

E_{HD} = heat delivery efficiency

$$= \frac{E_F E_D}{100} = 100\frac{\text{duct output}}{\text{total energy input}}$$

152P Definitions

DE (Delivery Efficiency) and DSE (Distribution System Efficiency) are defined for **seasonal and design** conditions.

Leaks to inside = energy from duct leaks that is to conditioned space, but does NOT flow through the registers
additional infiltration load = additional load due to imbalance in duct system leakage. Can be positive or negative.

Equipment-Component efficiency factors

DE = delivery effectiveness

$$= \frac{\text{duct output} + \text{leaks to inside}}{\text{duct input}}$$

ASHRAE Definitions

1996 ASHRAE Systems and Equipment Handbook Page 28.10

F_{MG} = miscellaneous gain factor

$$= \frac{\text{total heat delivered}}{\text{duct output}}$$

F_{MG} includes losses directly to conditioned space. 152P includes these in DE.

E_S = system efficiency

$= E_{HD}F_{MG}$

Equipment-Load interaction factors

F_{IL} = induced load factor

$$= \frac{\text{system induced load}}{\text{total heat delivered}}$$

F_{IL} includes off cycle effects & equipment losses to unconditioned spaces

F_{LM} = load modification factor

$= 1\text{-}F_{IL}$

$$I_S = \text{system index} = \frac{E_S F_{LM}}{100}$$

$$= \frac{E_F E_D F_{MG} F_{LM}}{10000}$$

$$= \frac{\text{total heat delivered} - \text{system induced load}}{\text{total energy input}}$$

152P Definitions

Equipment-Load interaction factors

F_{load} = infiltration load factor

$$= \frac{1}{1 + \text{additional infiltration load}}$$

F_{equip} = equipment factor

F_{recov} = recovery factor

DSE = distribution system efficiency

$$= \frac{\text{energy used by perfect system}}{\text{energy used by tested system}}$$

$=DE \bullet F_{load} \bullet F_{equip} \bullet F_{recov}$

152P Definitions

F_{load} does not include off cycle infiltration due to duct leakage. It is assumed that base building load is uses an infiltration load including duct leakage because standard blower door leakage measurement techniques include duct leakage in envelope leakage.

F_{equip} includes interaction between equipment and ducts in terms of reduced flow lowering equipment efficiency and two speed equipment effects, but NOT off cycle effects.

F_{recov} accounts for regain of losses to unconditioned spaces.

F_{load}, F_{equip} and F_{recov} are all in F_{IL} except:

- F_{IL} includes system off losses
- F_{equip} explicitly includes equipment derating due to reduced fan flow and system loss impacts one two speed equipment efficiency.

Table 1. Diagnostic Test Results and House Specifications

House	Stories	Floor Area, m^2 (ft^2)	Fan Flow m^3/hour at 20 °C, (cfm)		Supply Leakage, m^3/hour (cfm) @ 25 Pa		Return Leakage, m^3/hour (cfm) @ 25 Pa	
			Pre	Post	Pre	Post	Pre	Post
E1	1	93 (990)	1040 (612)	1040 (612)	253 (149)	46 (27)	49 (29)	12 (7)
G1	1	130 (1380)	698 (411)	678 (399)	116 (68)	65 (38)	105 (62)	46 (27)
G2	2	214 (2270)	1851 (1089)	1756 (1033)	342 (201)	121 (71)	95 (56)	19 (11)
G3	1	155 (1640)	1084 (638)	1234 (726)	163 (96)	65 (38)	581 (342)	70 (41)
E2	1	139 (1480)	1573 (925)	1535 (903)	209 (123)	87 (51)	70 (41)	31 (18)
Mean	-	**150 (1600)**	**1250 (735)**	**1250 (735)**	**127 (75)**	**45 (26)**	**106 (62)**	**21 (12)**

Table 2. Heating system specifications

House	Duct Location	Air handler location	Duct Material		Equipment Input Capacity kW, (kBtu/hr)	
			supply	return	Manufacturers data	Measured consumption
E1	attic	attic	flex duct	air handler platform	10 (34)	10 (34)
G1	attic	closet	sheet metal	sheet metal	24 (80)	16 (55)
G2	attic	attic	flex duct	flex duct	18 (60)	16 (55)
G3	attic & crawlspace	closet	sheet metal	sheet metal	24 (80)	16 (55)
E2	attic	attic	sheet metal	flex duct	9 (30)	11(37)

Table 3. Pre and Post Retrofit Two Week System Test Results

House	Tout, ° C (°F)	Number of cycles	Equipment Efficiency η_{equip}, %		Delivery Effectiveness %		Fractional supply leak loss[2], %		Fractional supply conduction loss[2], %		Fractional return loss[2], %	
			Pre	Post	Pre	Post	Pre	Post	Pre	Post	Pre	Post
E1	7 (45)	41	75	71	50	69	16	1	16	10	21	21
G1	3 (37)	88	74	75	60	68	23	24	10	8	8	0
G2	10 (50)	166	88	93	48	58	23	6	35	41	-4	-2
G3	11 (52)	30	79	n/a	60	n/a	18	n/a	11	n/a	11	n/a
E2	14 (57)	23	90	89	80	86	13	-3	23	26	-17	-12
Mean for E1, G1, G3, E2			**82**	**82**	**60**	**70**	**19**	**7**	**21**	**21**	**2**	**2**

[2] These fractional supply and return losses are fractions of capacity, not air flow

Table 4. Pre and post retrofit system cycling

House	Average ontime (minutes)		Average fractional ontime (%)	
	Pre	post	Pre	post
E1	17	12	32	7
G1	12	7	33	27
G2	6	7	34	6
G3	99	112	22	18
E2	22	17	6	5
mean	**31**	**31**	**25**	**13**

Table 5. Comparison of measured attic temperatures to ASHRAE 152P attic temperatures

House	152 Predicted Attic Temperature, °C [°F]	Measured attic temperatures, °C [°F]	
		PRE Retrofit	POST retrofit
E1	12 [54]	12 [54]	9 [48]
G1	8 [46]	7 [45]	7 [45]
G2	15 [59]	15 [59]	15 [59]
G3	16 [61]	15 [59]	13 [55]
E2	19 [66]	15 [59]	19 [66]
mean	**14 [57]**	**13 [55]**	**13 [55]**

Table 6 Comparison of measured and 152P calculated delivery effectiveness

House	PRE Retrofit		POST retrofit	
	152P Calculated Delivery Effectiveness, %	Measured delivery Effectiveness, %	152P Calculated Delivery Effectiveness, %	Measured delivery Effectiveness, %
E1	73	50	86	69
G1	61	60	82	68
G2	62	48	82	58
G3	60	60	80	n/a
E2	67	80	85	86
mean	**65**	**60**	**83**	**70**

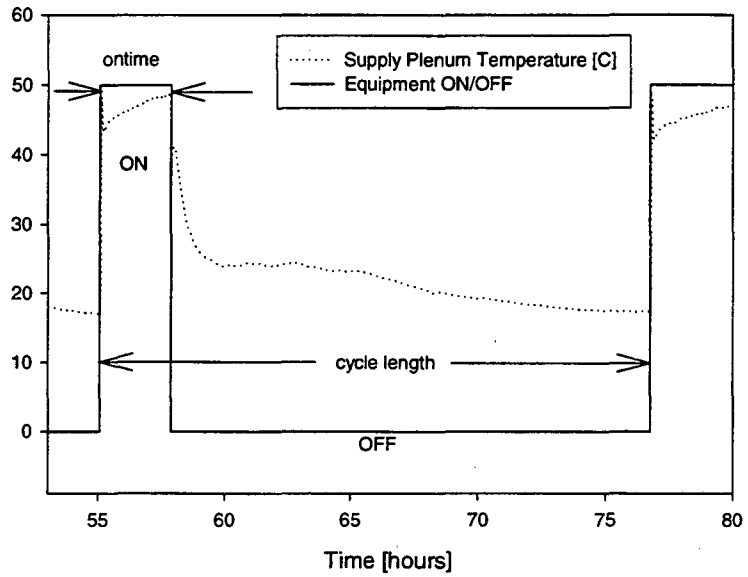

Figure 1. Illustration of system ontime and cycle length

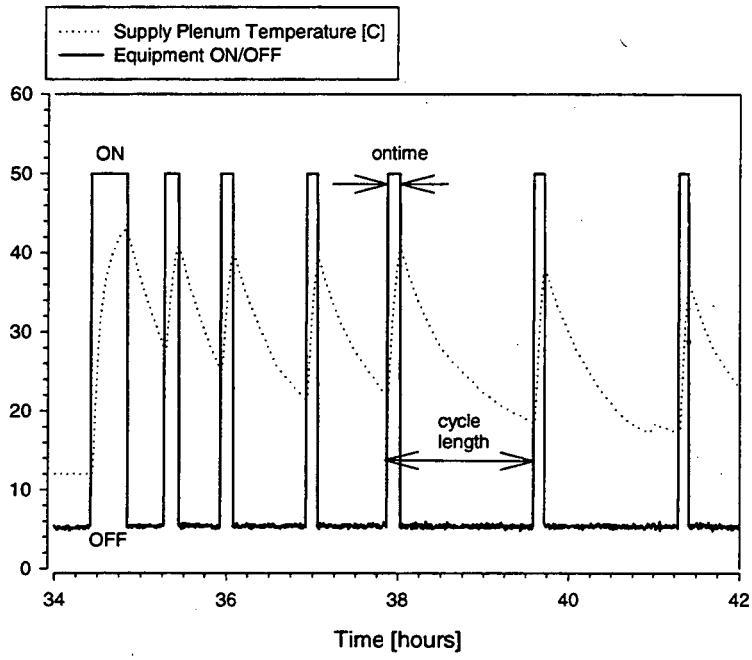

Figure 2. Short cycles in system E1

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Prepared for the U.S. Department of Energy under Contract No. DE-AC03-76SF00098