SF-98-30-1

Field Measurements of Interactions Between Furnaces and Forced-Air Distribution Systems

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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 discusses 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 the 1996 ASHRAE Handbook—HVAC Systems and Equipment, showing how distribution system efficiencies calculated using Standard 152P can be incorporated into the Handbook.

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 decrease in furnace efficiency increases 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, California. 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 the *1996 ASHRAE Handbook —HVAC Systems and Equipment*, Chapter 28 (ASHRAE 1996). In addition, the delivery effectiveness of the distribution system calculated using the standard is compared to measured data.

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.

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The field measurements were performed in two parts:

- Diagnostic tests of building and duct system characteristics.
- 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 et al. (1996). The following is an overview of the test procedures that provides a context for the experimental results.

Diagnostic Tests

The following measurements were performed for the diagnostic testing:

- *House pressurization test to determine exterior envelope leakage.*
- *Register airflows.*
- *Fan flow.*
- *Air leakage flow*: This was determined by pressurizing the ducts to 25 Pa (0.1 in. water) and measuring the leakage flow. This is not a direct measurement of leakage flows at operating conditions, but it does indicate the changes due to the retrofits.
- *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.
- *Equipment characteristics*: Heating/cooling capacity, location within the building.
- *House characteristics*: Number of stories, floor plan.

Two-Week Measurements

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

- *Register air temperatures*: Used together with the measured register airflows to calculate energy supplied to the house.
- *Plenum air temperatures*: Used together with measured fan airflows to calculate energy output by the furnace and input to the ducts.
- *Ambient air temperatures*: Outside air temperature and the temperature of air surrounding the ducts.
- *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.

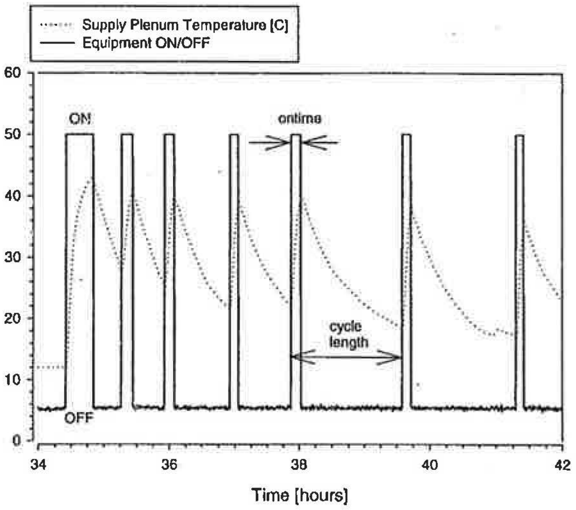

Figure 1 *Illustration of system on-time and cycle length.*

Data Binning Procedure and Cyclic Analysis

The measured field data were 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 airflow 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 while 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°C 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 track variations with weather conditions and those due to retrofits.

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

Delivery effectiveness (η_{del}) 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. (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 conductive losses.) 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 conductive 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.

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 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 is 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. 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 manufacturer "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/h], which is equivalent to 27% of the rated output). No apparent reason was found for this result. The measured outputs of the furnaces show that these furnace energy consumption values are correct for these systems; if the furnaces consumed the energy specified by the manufacturer, then we would see very poor equipment efficiencies in Table 3.

TABLE 1
Diagnostic Test Results and House Specifications

House	Stories	Floor Area, m^2 (ft^2)	Fan Flow m^3/h at 20°C (cfm)		Supply Leakage, m^3/h (cfm) @ 25 Pa		Return Leakage, m^3/h (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/h)	
			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 and Crawl Space	Closet	Sheet Metal	Sheet Metal	24 (80)	16 (55)
E2	Attic	Attic	Sheet Metal	Flex Duct	9 (30)	11(37)

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TABLE 3
Pre- and Post-Retrofit Two-Week System Test Results

House	T_{out}, °C (°F)	Number of cycles	Equipment Efficiency, η_{equip},%		Delivery Effectiveness, η_{del},%		Fractional supply leak loss,%		Fractional supply conduction loss[1],%		Fractional return loss[1],%	
			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

[1]These fractional supply and return losses are fractions of capacity, not airflow.

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 more than 150 cycles.

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 in 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 or lower air temperatures than those at the four measurement locations we used. If all four measurements were too low,

TABLE 4
Pre- and Post-Retrofit System Cycling

House	Average On Time (Minutes)		Average Fractional On Time (%)	
	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

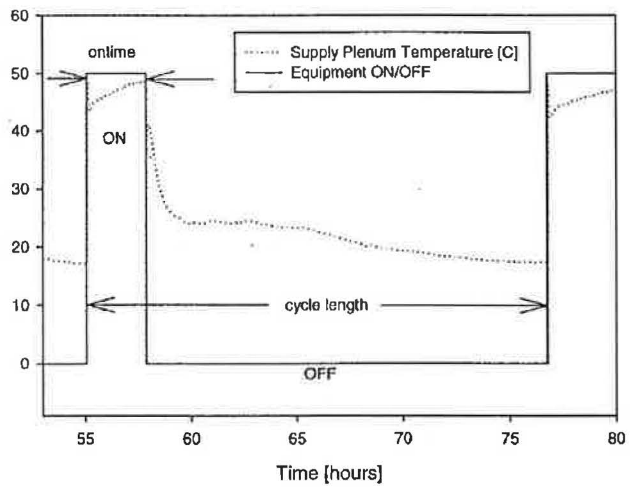

Figure 2 Short cycles in system E1.

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 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 gas 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 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. In addition, Table 4 shows that the average on-time 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 on-time 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 supply plenum air temperatures, thus reducing the cyclic average supply plenum air temperature and, therefore, reducing the overall equipment efficiency.

In Table 3, data are not presented for G3 post-retrofit due to sensor malfunctions, and the pre/ post-retrofit 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 conductive 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 conductive losses will be to the conditioned space, thus increasing the total energy delivered to the conditioned space (the delivery efficiencies 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).

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 conductive losses remain the same post-retrofit. The conductive losses do not change because the temperature of the air in the duct systems was increased in most of the systems after the retrofit. Note that without the additional insulation from the retrofit, the conductive 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 on-time (the length of time the system is on for in each cycle) and the average fractional on-time. The on-time divided by the total cycle time, where the total cycle time is from when one cycle begins to when the next cycle begins, is shown in Figure 1. These results show that when the systems were on, they were on for the same length of time (the on-time did not change), but the systems operated for less time (the fractional on-time went down). The reduction in fractional on-time 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 (given in Table 3). The change in on-time also reflects the reduction in building load due to reduced infiltration loads because of reduced duct leakage—both with the system on and off.

FORCED-AIR FURNACE SYSTEMS IN PROPOSED ASHRAE STANDARD 152P

The objective of the method of test in proposed ASHRAE Standard 152P, "Method of Test for Determining the Steady-State and Seasonal Efficiencies of Residential Thermal Distribution Systems," 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 efficien-

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cies 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 only at the forced-air heating sections of the standard.

Application of Diagnostic, Building Plan, and Default Input Parameters

Proposed Standard 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.

The 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 crawl spaces, insulation in all parts of the building structure, number of stories, number of return registers, ACCA *Manual D* 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.

Input for 152P

Diagnostic Input. The diagnostic inputs are determined for existing buildings and new construction where the building envelope and air-handling and air-conditioning equipment installation are 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, crawl space, 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.

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.

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

Duct Location

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

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 et al. [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* (ASHRAE 1993), Chapter 24, using the 97.5% design values. The seasonal outdoor dry-bulb temperature is the design temperature plus 9°C (16°F).

The temperatures of the duct zones outside the conditioned space are determined 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 effectiveness calculations is determined using a weighted average of the duct zone temperatures.

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Duct Wall Thermal Resistance

The diagnostic value of duct wall thermal resistance is determined by direct observation of the ducts. If the ducts have a 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* (ASHRAE 1993), Chapter 23, or an equivalent source. Air film resistance of 0.25 [Km^2/W] (1 [$h \cdot ft^2 \cdot °F/Btu$]) 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 manufacturer's specifications) or the observed value from diagnostic testing.

Duct Surface Area

The duct surface includes the surface area of plenums, and the supply and return duct surface areas 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 et al. 1996; Andrews 1996). The surface area of the ductwork is determined based on the conditioned floor area of the building.

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 et al. 1996).

Duct Leakage Flow Rate

Diagnostic supply and return leakage flows, Q_s and Q_r, are determined using either a fan pressurization test 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 et al. [1996], Downey and Proctor [1994], Modera and Wilcox [1995], Cummings et al. [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.

Duct Thermal Mass

The cyclic loss factor for sheet metal ducts is 0.05 and is 0.02 for nonmetallic ducts (based on the work of Modera and Treidler[1995]).

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:

$$B_s = \exp\frac{-A_s}{Q_e \rho_{in} Cp R_s} \tag{1}$$

$$B_r = \exp\frac{-A_r}{Q_e \rho_{in} Cp R_r} \tag{2}$$

For IP:

$$B_s = \exp\frac{-A_s}{60 Q_e \rho_{in} Cp R_s} \tag{3}$$

$$B_r = \exp\frac{-A_r}{60 Q_e \rho_{in} Cp R_r} \tag{4}$$

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

$$a_s = \frac{Q_e - Q_s}{Q_e} \tag{5}$$

$$a_r = \frac{Q_e - Q_r}{Q_e} \tag{6}$$

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

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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

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

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, any comparisons are made to delivery effectiveness.

COMPARISON OF MEASURED EFFICIENCIES 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 for this comparison is the total duct leakage under operating conditions (not the duct leakage to outside as required in 152P). This is because the measured delivery effectiveness only includes airflows from the registers and does not include any duct leakage that is to the conditioned space. Using leakage to outside results in 152P delivery effectiveness that is 3% to 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.

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)

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

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

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results in Table 4 shows that the 152P calculations tend to overpredict the delivery effectiveness by an average of 0.08. The average difference between measurements and predictions is about 0.11. These significant differences are due to the following:

- *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 ducts are 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 may lead to a real system having higher conductive losses than the simplified system in the 152P calculation.
- *Measurement uncertainties.* Possible sources of measurement uncertainties were discussed earlier.
- *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) 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 would reduce the 152P bias to only 3%, a much more acceptable figure (Table 4 shows that the systems studied here tended to have relatively short cycles, with the systems operating far from steady state).

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 in. [150 mm] to 14 in. [210 mm]) due to the increased insulation thickness, we would expect approximately a factor of 3 reduction in conductive losses. Combined with the factor of 2 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 factors other than steady-state performance are critical for the systems studied here. 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 conductive 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.

COMPARISON OF 152P TO EXISTING DISTRIBUTION SYSTEM ANALYSIS IN HANDBOOK

Appendix A contains a detailed comparison between proposed ASHRAE Standard 152P and the *1996 ASHRAE Handbook—HVAC Systems and Equipment.* The critical difference between the two approaches is that the Handbook lists many factors that require very detailed measured data or a computer simulation model, 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:

- 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). Seasonal and annual performance should be roughly equivalent despite being defined differently.
- The furnace 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 on-times and greater fractional on-time. 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 duct efficiency only includes energy flows directly out of registers, whereas the proposed standard includes 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 efficiency but high 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 standard, there is an infiltration load factor to account for the supply-return leakage imbalance. The 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, crawl spaces, 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 are included in the standard but not in the Handbook.

SUMMARY

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 furnace performance will be different from its rated performance unless the furnace is operating under the same conditions at which it was rated. For example, the gas furnaces tested here had 27% less energy consumption than the manufacturer's 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 affect 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 the environment.

The 152P calculation procedure illustrates how many factors must be accounted for, even with a much simplified calculation procedure. The paper 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 mean 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.

The 152P calculation procedure has been compared to the existing method in the *1996 ASHRAE Handbook—HVAC Systems and Equipment*, 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 that require very detailed measurements or the use of a computer model, 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.

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 A

Comparison of Existing ASHRAE Definitions to Those Proposed in ASHRAE 152P

ASHRAE Definitions, *1996 ASHRAE Handbook—HVAC Systems and Equipment*, 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

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

ASHRAE DEFINITIONS, *1996 ASHRAE HANDBOOK—HVAC SYSTEMS AND EQUIPMENT*, 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 and equipment losses to unconditioned spaces

F_{LM} = load modification factor
$= 1\text{-}F_{IL}$

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

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

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

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

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

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

152P DEFINITIONS

F_{load} does not include off-cycle infiltration due to duct leakage. It is assumed that base building load is 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.