RECEIVED

DEC 05 1995

OST

BNL -62346

Informal Report

CONF-960254--1

**ELECTRIC CO-HEATING IN THE ASHRAE STANDARD METHOD
OF TEST FOR THERMAL DISTRIBUTION EFFICIENCY:
TEST RESULTS ON TWO NEW YORK STATE HOMES**

J. W. ANDREWS, R. F. KRAJEWSKI, AND J. J. STRASSER

OCTOBER 1995

**Prepared for:
Office of Building Technologies
Building Equipment Division
U.S. Department of Energy
Washington, DC 20585**

**Energy Efficiency
and Conservation Division
DEPARTMENT OF APPLIED SCIENCE**

MASTER

**BROOKHAVEN NATIONAL LABORATORY
UPTON, LONG ISLAND, NEW YORK 11973**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, nor any of their contractors, subcontractors, or their employees makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency, contractor, or subcontractor thereof.

**ELECTRIC CO-HEATING IN THE ASHRAE STANDARD METHOD
OF TEST FOR THERMAL DISTRIBUTION EFFICIENCY:
TEST RESULTS ON TWO NEW YORK STATE HOMES**

J. W. ANDREWS, R. F. KRAJEWSKI, AND J. J. STRASSER

OCTOBER 1995

**Prepared for:
Building Equipment Division
Office of Building Technologies
U.S. Department of Energy
Under Contract No. DE-AC02-76CH00016**

**Brookhaven National Laboratory
Upton, New York 11973**

Electric Co-heating in the ASHRAE Standard Method of Test
for Thermal Distribution Efficiency: Test Results
on Two New York State Homes

Abstract

Electric co-heating tests on two single-family homes with forced-air heating systems were carried out in March 1995. The goal of these tests was to evaluate procedures being considered for incorporation in a Standard Method of Test for thermal distribution system efficiency now being developed by ASHRAE. Thermal distribution systems are the ductwork, piping, or other means used to transport heat or cooling effect from the building equipment that produces this thermal energy to the spaces in which it is used. Furthering the project goal, the first objective of the tests was to evaluate electric co-heating as a means of measuring system efficiency. The second objective was to investigate procedures for obtaining the distribution efficiency, using system efficiency as a base. Distribution efficiencies of 0.63 and 0.70 were obtained for the two houses.

Introduction

Electric co-heating tests were conducted during March 1995 on two single-family homes in New York State, as part of an evaluation of procedures for measuring thermal distribution system efficiency. Thermal distribution systems are the ductwork, piping, or other means used to transport heat or cooling effect from the building equipment that produces this thermal energy to the spaces in which it is used. Electric co-heating is the use of electric heaters within the heated space to maintain a given regime of indoor temperatures; the measured thermal energy delivered per unit time, divided by the average indoor-outdoor temperature difference, yields a parameter indicative of the thermal integrity of the building envelope under steady-state conditions. This parameter, which we call the Heating Load Coefficient (HLC), has dimensions of energy over (temperature X time).

Three quite different methods have been used to quantify the effect of thermal distribution on overall heating (or cooling) system efficiency. By far the most common method has been to measure the individual factors that contribute to energy losses in ductwork, such as conductive losses, air leakage, and fan-induced infiltration.[Caffey 1979, Cummings et al. 1989, Gammage et al. 1984, Matthews et al. 1990, Modera 1989, Parker 1989, Palmiter and Bond 1992, Proctor et al. 1992, Robison and Lambert 1989] A second method has been to compare, on a seasonal basis, the heating energy consumption of two large samples of homes, where one sample is heated with electric furnaces and duct systems while the second sample uses electric baseboard heating.[Lambert and Robison 1989] The third method uses electric co-heating. The co-heating system measures the "bare" heating load of a house, which is then compared with the energy used by the ducted heating system. One group has used co-heating in homes with electric furnaces and heat pumps.[Palmiter and Francisco 1994, Olson et al. 1993]

All of these methods have led to the same general conclusion, that duct systems typically lose 25 to 40 percent of the energy output from the space-conditioning equipment. In response to this information, ASHRAE has established Standards Project Committee (SPC) 152P, Standard Method of Test for Determining the Steady-State and Seasonal Efficiencies of Residential Thermal Distribution Systems.

Electric co-heating has been proposed as a research tool to be used in Standard 152. Co-heating would be used to obtain system efficiency as the dimensionless ratio of the HLC measured via co-heating to a similar parameter that includes the thermal performance of the building's own heating system. This second parameter is called the Heating System Coefficient (HSC). System efficiency, which accounts for all the thermal losses in both the equipment

and the distribution system, including any interactive effects on the heating load, is important not only in its own right, but also because it serves as the starting point for determining the distribution system efficiency.

To improve accuracy, it was proposed that the tests be done at night (to minimize the effect of solar gains) and that two nights of testing be required. Originally it was assumed that the heating system would be tested on one night, with the coheat test on the following night, but the suggestion was made [Modera 1995a] that both kinds of tests be done for one-half of each night, with changeover to occur about midnight and with the order of the tests reversed on the second night. (We will use the term "flip-flop protocol" for this approach.) The advantage of doing both kinds of tests on both nights would be to minimize error due to weather variation (e.g. wind speed, large temperature changes) between the two tests. Reversing the order of the tests on successive nights would minimize any advantage to one system or the other caused by the response of the building to the local climate (e.g. differential response of thermal mass to different rates of outdoor temperature change typically encountered before and after midnight). There was a concern, however, that the lack of time between tests might give rise to errors associated with transient effects, such as charging or discharging of thermal mass coupled to the conditioned space, caused by an inability of the co-heating system to match precisely the temperature maintained by the building's own heating system.

The first objective of these tests, therefore, was to provide as many of these answers as possible within a very limited time. Specifically, we sought to shed light in the following areas:

- o Do the coheat tests give repeatable results?
- o Can the co-heating system maintain constant temperatures in the living space? Can the home's own heating system maintain room temperatures sufficiently near setpoint using the system's own thermostat, or will it be necessary for a computer to take charge during the system test?
- o Should co-heating and heating system testing be done on different nights, or should the flip-flop protocol be used?

The second objective was to evaluate the process by which thermal distribution efficiency is obtained from system efficiency. The degree to which system efficiency is less than unity is the result of energy losses in the equipment and in the distribution system. Distribution efficiency can thus be obtained by dividing system efficiency by the efficiency of the heating equipment and then making a (generally small) correction for the interaction between the two. Extensive discussion within SPC152P related to whether equipment efficiency should be determined by a) comparing the thermal output of the equipment with its purchased energy input, or b) using other approaches such as steady-state efficiency of a fuel-fired furnace using stack temperature and CO₂ concentration, or use of the electric-resistance elements in a heat pump to permit the assumption that equipment efficiency equals 1.00.

Description of Test Houses

Both houses have forced-air thermal distribution systems and are of average to slightly larger-than-average floor area. Both were fully furnished and unoccupied. In other respects they are quite different.

The Long Island House. The first house tested was a two-story, 2000 ft² (186 m²) single-family structure constructed in 1980 on a national laboratory site in Long Island, New York. Figure 1 shows a plan of the house together with locations of registers, coheaters, and indoor temperature sensors. Built

as a research house, it was intended to demonstrate how energy-efficient construction techniques could be applied to a house of traditional design and appearance. Its passive solar design features include a high level of thermal mass, and it has a basement that is insulated on the outside of the foundation to a value of $10 \text{ ft}^2\text{-F-h/Btu}$ ($1.7 \text{ m}^2\text{-K/W}$). A detailed description of the house and test results obtained shortly after construction are given in Jones et al. 1984.

The Long Island house has two separate duct systems. One system, in the basement, is served by an electric heat pump. It has a main supply trunk with four branches, all insulated to approximately $4 \text{ ft}^2\text{-F-h/Btu}$ ($0.7 \text{ m}^2\text{-K/W}$), and one uninsulated return duct located in the chimney chase. This return duct accepts air from a large grille on the first floor and is also in communication with registers located over the second-floor doors. Supply to the second floor was originally accomplished by passageways in the first-floor ceiling that allowed warmed first-floor air to make its way up to the second floor. After the research program on this house had ended and it began to be used to house personnel, an additional duct system was installed in the attic to provide air conditioning and electric-resistance auxiliary heat to the second floor. It takes return air from the abovementioned over-the-door registers and delivers supply air through registers in the ceilings of the rooms on the second floor. The upstairs and downstairs heating systems are controlled by separate thermostats. The house was tested in an as-found condition, with one exception. To facilitate measurement of the quantities of interest, the heat pump in the basement was inactivated so that both the attic and basement duct systems were served by electric-resistance coils.

The Watertown House. The second house tested was a 1585 ft^2 (147 m^2) double-wide manufactured home fabricated in two sections in December, 1994 and shortly thereafter set in place over a well-vented above-ground crawl space on a site in Watertown, New York, near the Canadian border. The one-story plan (Figure 2) is an open, contemporary style with wide, doorless passages between the kitchen, living, and dining areas grouped at one end of the house. In contrast to the Long Island house, the Watertown house has no basement, no passive solar design, and no added thermal mass.

Heat is provided by a natural gas-fired, forced-air furnace rated at 95 000 Btu/h (27.8 kW) input and 76 000 Btu/h (22.3 kW) output, for a nominal steady-state efficiency of 80%. The input rate was measured by timing the sweep hand on the gas meter while the burner was on, and using the higher heating value for delivered gas at Watertown, given by the gas utility as 1019.4 Btu/ft^3 (38.0 MJ/m^3), and found to equal 97 150 Btu/h (28.4 kW). The rated Annual Fuel Utilization Efficiency (AFUE) for this furnace is 75%.

The furnace, located within the conditioned space (in the utility room), is a downflow type with a counterflow heat exchanger. Combustion air from the crawl space mixes with natural gas at the bottom of the furnace; the products of combustion flow through a heat exchanger and a double-wall vent pipe through the roof. Heated air flows through an enclosed galvanized sheet-metal duct system, which is housed in the floor cavity. The duct system has two main trunk lines, one serving each half of the building; these trunk ducts are connected by a 12-inch (0.3 m) diameter flex duct. Branch lines serve 14 supply registers.

Return air from the interior is unducted. The return side of the furnace has a louvered door that accepts air from the utility room. This room is open to the public areas of the house. The bedrooms and bathrooms have small louvered openings over the doors to allow air flow from these rooms into the main hallway and thence to the return side of the furnace. Ventilation openings to the outside, including fireplace doors, range hood, and an intake vent from the outside to the return side of the furnace, were taped shut during the test.

System Efficiency

System efficiency η_{system} was defined above in general terms as the ratio of HLC to HSC. The motivation for this definition is that it expresses the fraction of purchased input energy that effectively reaches the conditioned space. The algorithm for calculating HLC and HSC, described in detail below, serves as the operational definition of η_{system} .

The flip-flop protocol, also mentioned in the Introduction, was used in testing both houses. A complete test covered two nights. A night's testing included a duct-system test and a coheat test, each test consuming approximately 6 hours, one of them in the evening hours before midnight and the other in the early morning hours after midnight. The time order of the tests was reversed on the second night. In the case where the heating system was tested first, the registers were sealed as quickly as possible after midnight. (Generally this took about 45 minutes.) The coheat system was started within three minutes of midnight, i.e. we did not wait for the register taping to be completed before starting the coheaters. During the second night, when co-heating was done first, register covers were removed immediately before midnight so that the heating system could be activated within three minutes of the target time.

In order to reduce to a minimum the effect of solar gain as a variable, the windows on the east, south, and west sides of each house were covered with reflective foil. The assumption is that the efficiency of the thermal distribution system is not a strong function of solar gain, but it is more easily measured if transient effects such as solar gain are minimized.

Two complete flip-flop protocols (4 nights) were performed in the Long Island house. In the Watertown house, one flip-flop protocol (2 nights) was performed.

Calculation of System Efficiency. Computation of system efficiency used the following steps:

1. Regardless of the order in which the tests were conducted, compute the heating load coefficient (HLC) for the coheat test using the following equation:

$$HLC = \frac{H_{\text{coheat}} + H_{\text{ig-coheat}}}{(T_c - T_a)_{\text{coheat}} t_{\text{coheat}}} \quad (1)$$

where H_{coheat} is the heat deposited into the house by the coheaters; $H_{\text{ig-coheat}}$ is the sum of internal energy gains to the house, other than from the coheaters, during the coheat period; $(T_c - T_a)_{\text{coheat}}$ is the average indoor-outdoor temperature difference during the coheat period; and t_{coheat} is the duration of the coheat period. $H_{\text{ig-coheat}}$ includes auxiliary electrical energy used within the house as well as metabolic energy from occupants.

2. Compute $H_{\text{ig-system}}$, the internal energy gain to the building during the test of the system under evaluation, including metabolic heat and electric energy delivered directly to the conditioned space (and not used to run the equipment). The internal gain rate (IGR) is then calculated (in units of energy per unit time) as the ratio of $H_{\text{ig-system}}$ to the system test duration t_{system} .

3. Calculate the balance-point temperature depression ΔT_{bal} equal to the ratio of IGR to HLC. Then compute the adjusted indoor-outdoor temperature difference ΔT_{io-adj} :

$$\Delta T_{io-adj} = (T_c - T_a)_{system} - \Delta T_{bal} \quad (2)$$

where $(T_c - T_a)_{system}$ is the average indoor-outdoor temperature difference during the system test interval.

4. A heating system coefficient (HSC) is calculated as

$$HSC = \frac{G_{system} + E_{system}}{\Delta T_{io-adj} t_{system}} \quad (3)$$

where G_{system} and E_{system} are the purchased fuel and electric energy used by the system under evaluation, respectively.

5. The system efficiency η_{system} for each test night is equal to that night's ratio of HLC/HSC. The best overall value for η_{system} is then the average of the values determined separately for the two nights' tests.

Key data for system efficiency calculations on the four nights' testing in the Long Island house are shown in Table 1. The HLC's average about 100 Btu/F-h [50 W/K] more than they were in the early 1980's for this house. [cf. Jones et al. 1984, p. 43] Comparison of perfluorocarbon tracer gas measurements then and now showed an increase in the air infiltration rate of about 0.4 air changes per hour, which would account for this difference.

In any case, the average of the two values should be a better estimate of the true system efficiency than either value alone, or even than the average of two tests done in the same order. In the tests above, these averages turned out to be:

Test Dates	η_{system} Ducts First	η_{system} Coheat First	η_{system} Average
3/15-16 & 16-17	0.686	0.578	0.632
3/17-18 & 20-21	0.682	0.506	0.594

The two average values differ by 4 percentage points, or 6% of their mean. It should be noted that since electric resistance heat was used in the ducts in these tests, the system efficiency and the distribution efficiency are the same in this case. The average of the two individual averages is 0.613.

Table 1. System Efficiency Calculations for the Long Island House.

Date	$T_{c-coheat}$ F [°C]	$T_{a-coheat}$ F [°C]	HLC B/F-h [W/K]	IGR B/h [W]	$T_{c-system}$ F [°C]	$T_{a-system}$ F [°C]	$\dot{G}_{system}$ B/h [W]	$\dot{E}_{system}$ B/h [W]	HSC B/F-h [W/K]	η_{system}
3/15 S-C	74.1 [23.4]	42.8 [6.0]	456 [240]	2165 [634]	74.1 [23.4]	44.0 [6.7]	0	16 900 [4950]	665 [351]	0.686
3/16 C-S	74.1 [23.4]	44.0 [6.7]	421 [222]	2165 [634]	74.2 [23.4]	44.9 [7.2]	0	17 640 [5170]	729 [384]	0.578
3/17 S-C	74.1 [23.4]	39.9 [4.4]	431 [227]	2165 [634]	74.0 [23.3]	45.0 [7.2]	0	15 160 [4440]	632 [333]	0.682
3/20 C-S	74.1 [23.4]	42.4 [5.8]	463 [244]	2165 [634]	74.0 [23.3]	49.4 [9.7]	0	18 210 [5330]	915 [482]	0.506

Note 1: Date is that in effect prior to midnight.

Note 2: $G_{system} = G_{system}/t_{system}$; $E_{system} = E_{system}/t_{system}$.

Note 3: B = Btu.

Note 4: S-C in column 1 means the heating system test was done before midnight and coheating after; C-S indicates the reverse.

Table 2. System Efficiency Calculations for the Watertown House.

Date	$T_{c-coheat}$ F [°C]	$T_{a-coheat}$ F [°C]	HLC B/F-h [W/K]	IGR B/h [W]	$T_{c-system}$ F [°C]	$T_{a-system}$ F [°C]	$\dot{G}_{system}$ B/h [W]	$\dot{E}_{system}$ B/h [W]	HSC B/F-h [W/K]	η_{system}
3/27 S-C	71.6 [22.0]	29.0 [-1.7]	333 [176]	3190 [934]	71.9 [22.2]	38.3 [3.5]	15 580 [4560]	180 [50]	657 [346]	0.507
3/28 C-S	71.6 [22.0]	37.8 [3.2]	366 [193]	3000 [879]	70.8 [21.6]	30.9 [-0.6]	21 570 [6320]	240 [70]	688 [363]	0.532

Notes: See under Table 1.

Similar calculations for the Watertown house are shown in Table 2. Here we were only able to do one complete test (2 nights) because of time limitations. The system efficiencies and their average were:

Test Dates	η_{system}	η_{system}	η_{system}
	Ducts First	Coheat First	Average
3/27-28	0.507	0.532	0.520

Agreement between the two nights is much better than that between Night 1 and Night 2 of either test in the Long Island house.

Discussion of System Efficiency Variation in the Long Island House.

There is considerable variation in the coefficients (especially the HSC's) and in the system efficiencies. Some of this is probably due to thermal mass effects. (As mentioned above, thermal mass was intentionally added to the house as storage for its passive solar features.) Thermal mass retards the response of a building to changes in the indoor-outdoor temperature difference. When the outdoor temperature drops, the indoor-outdoor ΔT rises immediately, but the heating load takes longer to respond. This results in a smaller value of the HLC or HSC. We found that the temperature tended to drop more rapidly before midnight than after, which meant that whatever system was tested first would be favored. In particular, the large HSC on March 21 was accompanied by a significant temperature rise in the hours immediately before and after midnight. This would result in a lower indoor-outdoor ΔT for the duct system than under more normal conditions, but not much change in the heating load, due to the thermal mass retarding the impact.

However, thermal mass cannot explain all of the variation. For example, for the pair of tests on 3/15-16 and 3/16-17, the system efficiency was higher when the ducts went first even though the outdoor temperature profile was relatively flat during those two nights. An alternative explanation [Modera 1995b] could be that the portion of the duct system in the attic benefitted from residual solar gain when the heating system ran first (before midnight) but that little or no solar-derived heat was left in the attic after midnight. (Note that this is natural solar gain; there is no active or passive solar system on the roof.) In that case we would expect the HSC's to be lower when the ducts were tested first, which was in fact observed. Also, we note that the two days preceding the nights in which the ducts were tested first were only moderately sunny, receiving 30% (March 15) and 60% (March 17) as much insolation on a horizontal surface as would be expected on a bright sunny day in March (i.e. 500 langleys). The co-heating system, on the other hand, should not be affected nearly as much by attic temperature (in view of the fact that the second-floor ceiling is well insulated), so as far as this effect is concerned, the HLC's should not be biased on the basis of which system was tested first. This is consistent with our data: the highest and the lowest of the four HLC values were obtained when the co-heating system was tested first. (Note that the two days preceding the tests in which coheaters went first were quite sunny, with insolation exceeding 90% of the 500-langley sunny-day figure.)

The flip-flop protocol attempts to compensate for this and other artifacts of the order of testing by reversing the order during the second test. However, there is a concern that, for ducts in an attic, the measured value of the HSC when the ducts are tested first might depend significantly on whether the day before the test was sunny or not, while the HLC for the co-heating tests would not depend very much on this variable. That would mean that the measured system efficiency is dependent on solar gain, despite the attempt to eliminate this variable by covering the windows with reflective foil. In other words, in the case of attic ductwork, variation in HSC with the time of the test may not be an artifact of the test method at all, but may

be a real effect. Attic ducts used for heating may be more efficient during and shortly after sunny periods. (The reverse would then be true for cooling.) In that case, this will have to be recognized in any test method, not just the one involving co-heating. In the end, it may be that, for attic ductwork, Standard 152 will need either to account for the dependence of system efficiency on solar gain or to minimize it by restricting the ambient conditions under which systems with ducts in an attic can be tested.

In the Watertown house, the results obtained on two successive nights were much closer. Significantly, this house is of light construction and its ducts are not in the attic.

Room Temperatures. Another concern was whether a co-heating system could maintain sufficiently stable room temperatures to permit an unbiased test. Related to this is the question whether the heating system should be operated under the control of its own thermostat (to give a real-world picture of the whole system) or under the control of the investigators' computer (to provide greater repeatability between tests of the same heating system and better comparability from one heating system to another). Our tests were run with the heating systems under the control of their own thermostats. The computer controlled only the coheaters, although it acquired data from both coheaters and heating system.

Figure 3 shows the hourly-average room temperatures in the Long Island house for each six-hour test. The co-heating system was able to maintain this average within a small fraction of a degree (F or K) of the setpoint. The heating system showed somewhat greater variation, but still within the narrow range of 0.2 F [0.1 K] within a test and 0.4 F [0.2 K] between tests.

The Watertown house (Figure 4) showed a similar level of control for the co-heating system. The home's heating system, however, exhibited a much larger range of temperature variation (a maximum of 1.3 F [0.7 K] within a test and 2.2 F [1.2 K] between tests. On both nights there was a net downward drift in conditioned-space temperature between the two tests.

We make no judgment here concerning the question of how the heating system should be controlled, but offer the data as input to the discussion.

Interior Thermal Mass. As mentioned above, in both nights of testing in the Watertown house, there was a significant drop in conditioned-space temperature between the two tests. This amounted to 0.3 F (0.17 K) on the first night and 0.8 F (0.44 K) on the second night. Such a temperature relaxation provides a certain amount of "free heat" to the system that is tested second. (A temperature rise would exact a heat penalty on the second system.) We can estimate the impact of this effect on the measured system efficiency as follows.

For the case where the heating system is tested first and the co-heating system second, we can treat the heat released by the thermal mass inside the thermal barrier as another internal gain. If M is this thermal mass, in Btu/F (W-h/K), then we can modify Equation 1 to read

$$HLC' = \frac{H_{coheat} + H_{ig-coheat} + M (T_{c-system} - T_{c-coheat})}{(T_c - T_a)_{coheat} t_{coheat}} \quad (4)$$

or

$$\frac{HLC'}{HLC} = 1 + \frac{M (T_{c-system} - T_{c-coheat})}{H_{coheat} + H_{ig-coheat}} \quad (5)$$

This leads to a corrected value η'_{system} for system efficiency:

$$\eta'_{system} = \frac{HLC'}{HSC} = \eta_{system} \left[1 + \frac{M (T_{c-system} - T_{c-coheat})}{H_{coheat} + H_{ig-coheat}} \right] \quad (6)$$

Where the heating system is tested second, we can use the same reasoning embodied in Equation 6, which will now penalize the heating system and reduce the system efficiency if the indoor temperature declines.

In the Watertown house, the total energy input during the coheat period, including internal gains, was 88 370 Btu (93.2 MJ) on the first night and 74 100 Btu (78.2 MJ) on the second night. These quantities, together with the indoor temperature drops quoted above, enable us to calculate the value of M for which η'_{system} is the same on both nights. This value is 3350 Btu/F (6.4 MJ/K). We note the light construction typical of a home that must be trucked to the building site, and also consider that only the thermal mass inside the thermal barrier is effective here. Given the average HLC of 350 Btu/F-h (184 W/K), and assuming that the thermal mass inside the thermal barrier is half the total thermal mass in the house, this would correspond to a thermal time constant of 19 hours. Although we did not do a cooldown test of the house, this value does not appear unreasonable.

The value of η_{system} corresponding to the above thermal mass is 0.513. We will use this value in the calculation of distribution efficiency below.

The greatest average indoor temperature change between tests in the Long Island house was 0.1 F (0.06 K). We do not know the value of the thermal mass inside the thermal barrier for the Long Island house, but if we assume a value of 10,000 Btu/F (19.0 MJ/K) as a benchmark, the theory developed above would lead to a maximum 1.3% error (0.8 percentage points) in the system efficiencies shown in Table 1.

Basement Temperatures. One major concern about the flip-flop protocol relates to cases where ductwork is located in a basement or other partly conditioned space. Heat lost from ducts to such a space may raise its temperature and thus retard heat loss from the adjacent conditioned space. The duct losses then do some good and may be regarded as being partially regained. Because of this effect, it is possible that a co-heating test might unfairly penalize the duct system in the following way. When the coheaters are first turned on at midnight of the first test night, they will "see" a basement that is warmer than it would have been in the absence of a duct system (because of the heat that the ducts have been losing to the basement up to the moment they are shut down). The coheaters then might get something of a "free ride," since they would have the benefit of a warm basement retarding heat loss but not have to expend any energy to maintain it.

The crucial issue, then, is how long does it take the basement temperature to relax to its no-duct equilibrium value? Figure 5 shows data from the Long Island house that shed some light on this. During the "flip-flop" tests for system efficiency, temperatures were measured at three levels in the basement, approximately 2, 4, and 6 ft (0.6, 1.2, and 1.8 m) above the

basement floor. These sensors were located near the center of the basement, approximately 6 ft (2 m) to the side of the long axis of the main trunk duct. In each of the four cases, a clear exponential relaxation of basement temperature occurs, averaging 3 F (1.7 K) for the highest level, with the temperature falling when coheaters replace the duct system and rising in the reverse case. The time constant for this relaxation was approximately one hour.

The temperatures at the lower levels show similar time variation but with smaller amplitudes. The one anomaly, for which we have no explanation, is the closeness of the two lowest temperature profiles on March 20-21.

A theoretical discussion of the impact of duct systems on basement temperature [Andrews 1993] identified three mechanisms by which heat lost from ducts to a basement is eventually transferred from the basement to the outside. Each of these mechanisms was judged to have its own relaxation time constant. The first mechanism is exfiltration of basement air warmed by the ducts. The second is conductive heat loss to the ambient, primarily through the upper portion of the foundation walls. The third mechanism is conductive heat loss to the ground, primarily through the lower portion of the foundation walls and the basement floor. Typical time constants for these mechanisms were estimated to be several hours, several days, and a month or more, respectively. In particular, the relaxation time for the air exchange mechanism should equal the inverse of the air-exchange rate. Perfluorocarbon tracer gas measurements on the house showed a basement air-exchange rate of 1.0 h^{-1} during operation of the duct system and 0.6 h^{-1} during co-heating. The temperature relaxation observed is therefore attributed to the air-exchange mechanism.

The question remained whether there was any additional longer-term relaxation of the basement temperature. Two longer periods of co-heating were examined to determine whether any such effect could be seen. Analysis of the data was complicated by the fact that the basement temperatures rose and fell in response to changes in the outdoor ambient temperature, making it difficult to separate out small, slow changes from other causes. We can say, however, that if any such changes occurred over a 1- to 2-day time frame, they were probably less than 1 F [0.6 K].

To the extent that rapid relaxation via the air-exchange mechanism is the major factor governing the change in basement air temperatures when switching between the duct system and the coheaters, it should be possible to reduce bias by delaying the start of the second half of each night's test. Table 3 shows the percentage of the original bias that remains, averaged over a 6-hour test period, both with and without delays, for three values of the relaxation time constant. For our observed 1-hour time constants with no delay between tests, we conclude that our results retain approximately 16% of the bias caused by the finite relaxation time of the basement temperature. The next section considers how we might quantify this bias, at least approximately.

Effect of Basement Ducts on System Efficiency. The above discussion leads naturally to a consideration of how significant the change in basement temperature might be. One way to approach this would be to ask what the effect on the calculated system efficiency would be if the basement temperature took a very long time to relax. Having determined that, we can then multiply this effect by the appropriate percentage in Table 3 to obtain the impact of the basement temperature profile in the actual case.

Taking the Long Island house as an example, let us first estimate the reduction in heat loss if the basement temperature is 3 F (1.7 K) higher with a duct system than without one. The area of the basement ceiling is 1000 ft^2

(93 m²). Its thermal resistance (R-value) is estimated as shown in Table 4 (ASHRAE 1993).

Because the first floor is covered approximately 50% with carpet and 50% with vinyl tile, we will use the average of the above R-values, or 3.4 ft²-F-h/Btu (0.6 m²-K/W). Using this value together with the abovementioned floor area and incremental temperature difference, one obtains an incremental (reduced) heat loss of 880 Btu/h (260 W).

Air exchange from the basement to the first floor was measured during heating system operation using the perfluorocarbon tracer gas technique; the average value over 17 hours was 100 cfm (170 m³/h). The incremental (reduced) heat loss from this air exchange due to the 3 F (1.7 K) higher basement temperature is then 320 Btu/h (90 W).

Table 3. Percent of Bias Remaining Due to Basement Thermal Regain, for Various Relaxation Time Constants and Delay Times.

Time Constant for Basement Temperature Relaxation	Percent of Bias Remaining		
	No Delay	One-Hour Delay	Two-Hour Delay
1 hour	16%	6%	2%
2 hours	31%	19%	12%
3 hours	43%	31%	22%

Table 4. Estimation of Thermal Resistance of Basement Ceiling in the Long Island House.

Layer	Thermal ft ² -F-h/Btu	Resistance m ² -K/W
Still Air, Upper Surface [1]	0.92	0.16
Floor Coverings [2]		
a) Vinyl Tile	0.05	0.01
b) Carpet and Rubber Pad	1.23	0.22
Plywood, 3/4 in. (19 mm) [2]	0.93	0.16
Still Air, Lower Surface [1]	0.92	0.16
Totals		
a) With Tile Flooring	2.82	0.49
b) With Carpet and Pad	4.00	0.70

Sources in ASHRAE 1993: [1] Table 1; [2] Table 4.

The sum of the conductive and infiltration incremental heat losses is 1200 Btu/h (350 W). This represents a best estimate of the numerical value of the effective regain of heat lost from the ducts to the basement, based on the basement temperature difference with and without ducts.

The average heat load of the house can be estimated as the average energy use by the resistance coils (17,000 Btu/h [5.0 kW]) multiplied by the average system efficiency (0.61) to obtain 10,400 Btu/h (3.0 kW). The bias in the system efficiency value that would ensue if the basement temperature did not relax when the co-heating system was turned on may then be estimated as 1200/10400, or 11.5%. Estimating from Table 3 that 16% of this bias remained in our experiment, we conclude that our system efficiencies were biased

downward by approximately 2%. A revised best estimate of the system efficiency (equal to the distribution efficiency because electric resistance was used as the source of heat) is therefore 1.02×0.613 , or 0.63.

As mentioned above, in addition to this short-term basement-temperature relaxation, there may be longer-term effects. Although we were not able to confirm these, we also could not rule out longer-term changes on the order of 1 F (0.6 K). A rise in basement temperature due to duct losses of this magnitude, which persisted through the entire co-heating test, would increase the true system and distribution efficiency in the Long Island house by one-third of the 11.5% calculated above for a 3 F (1.8 K) rise. If present, such an effect would raise the true system and distribution efficiency to 0.65.

Distribution Efficiency

Of all the factors defined in the proposed ASHRAE Standard Method of Test, perhaps the most important will be the distribution efficiency. Distribution efficiency η_{dist} is defined (assuming identical exterior environmental conditions and indoor temperature profile) as the ratio of the purchased energy the tested system would require if it had a perfect distribution system to the purchased energy required given the as-found distribution system. [Modera et al. 1992] A perfect distribution system is a theoretical construct. By definition it has no energy losses and no impacts on the equipment efficiency or the heating load. Distribution efficiency is related [Jakob et al. 1986, Andrews 1994] to the system efficiency via the following equation:

$$\eta_{dist} = \frac{\eta_{system} F_{equip}}{\eta_{equip}} \quad (7)$$

where the equipment efficiency η_{equip} is the ratio of equipment heat output to purchased energy input to the equipment in the system as found, and the equipment factor F_{equip} is the ratio of the equipment efficiency with the as-found distribution system to the efficiency the equipment would have if the distribution system were made perfect.

For a single-capacity gas or oil furnace, F_{equip} is expected to be slightly greater than unity (within a few percent) because with the existing distribution system it will have a larger fractional on-time than if distribution losses were eliminated. The consequent reduction in cycling losses usually improves equipment efficiency somewhat.

Equipment Efficiency. Generally, the most important factor relating system efficiency as measured above to distribution efficiency will be equipment efficiency. Equipment efficiency for fuel-fired furnaces can be evaluated in at least two ways:

- o Stack temperature and CO₂-concentration measurements yielding steady-state efficiency, with correction factors for cycling and jacket losses; or
- o Direct measurement of energy inputs to and outputs from the equipment during the time interval that the system is tested.

One of the objectives of the testing in Watertown was to investigate the practicality of these two methods of test. In particular, an issue within the ASHRAE SPC152P committee has been whether to base results for fuel-fired furnaces on stack measurements as performed in ASHRAE Standard 103 or to depend on input-output tests as performed for heat pumps in ASHRAE Standard 116.

During the early morning hours of March 29, we performed an input-output test of the furnace efficiency. The test was not definitive, in that only a single temperature sensor was used at the supply plenum and immediately in front of the furnace's return grille. (As mentioned above, the house has no return ductwork.) The difference between these two temperatures, averaged over those times when the air-handler fan was on, was combined with fan on-time and fan-on air-flow rate to yield the furnace's thermal output. Fuel use was determined from burner on-time and fuel input rate. The equipment efficiency value for the test period obtained in this manner was 81.5%.

Two steady-state efficiency measurements using stack temperature and flue-gas CO_2 content were also made. The temperature difference between the flue gas and the combustion air intake ranged from 492 to 497 F [256 to 258 K] and the flue-gas CO_2 content from 8.4% to 9.0%, yielding a steady-state efficiency of 77%.

Given an AFUE of 75% and a measured steady-state efficiency of 77%, it appears to us unlikely that the equipment efficiency under cycling conditions would be as high as 81.5%. (The average burner fractional on-time was 0.22.) However, given the tentative nature of our input-output measurement, it might be considered something of a victory that we obtained an equipment efficiency as reasonable as we did. Although our experience in Watertown will not, by itself, enable SPC152P to resolve the question of how to measure equipment efficiency, it did strengthen our belief that even if the input-output approach is adopted, a stack efficiency measurement will provide a very valuable check on the results in the case of combustion equipment.

For the purpose of obtaining a useful equipment efficiency as input to the distribution efficiency calculation, we have chosen to take the measured steady-state efficiency (77%) as our benchmark and subtract a nominal 2 percentage points for cycling losses, yielding a best estimate of 75% for η_{equip} , which is in agreement with the AFUE for this appliance. (No subtraction for jacket losses was made since the furnace was in the heated space.)

Equipment Factor. The one additional item needed to calculate distribution efficiency is the equipment factor. An equation was developed for proposed use in SPC152 for furnaces. This equation is:

$$F_{\text{equip}} = 1 + 0.18 (1 - \eta_{\text{equip}}) (1/\eta_{\text{system}} - 1/\eta_{\text{equip}}) \quad (8)$$

The derivation of this equation from ASHRAE data (ASHRAE 1992, Jakob et al. 1986) is given in Andrews et al. 1995. The data set used a specific furnace with a seasonal efficiency that varied slightly from run to run, but which generally approximated 0.75. Variations in equipment efficiency were factored into Equation 8 by assuming that $F_{\text{equip}} - 1$ is proportional to $1 - \eta_{\text{equip}}$. This assumption was based on the fact that a 100%-efficient furnace would necessarily have F_{equip} equal to 1.

Using our value of 0.75 for η_{equip} and our best value of 0.513 for η_{system} , we obtain an estimated equipment factor of 1.028.

Calculation of Distribution Efficiency. A best value of distribution efficiency was calculated from the above numbers according to the equation

Conclusions

We have drawn the following conclusions from this work:

$$\begin{aligned}
 \eta_{dist} &= \frac{\eta_{system} F_{equip}}{\eta_{equip}} \\
 &= \frac{0.513 \times 1.028}{0.75} \\
 &= 0.70
 \end{aligned}$$

1. The "flip-flop protocol" performed well in the two homes tested, with no apparent difficulties arising from the rapid changeover from one system to another. Some caution may be in order with respect to homes with attic ductwork and/or very large thermal mass.
2. In the Long Island house, the basement temperature relaxed very quickly when the control was shifted between the heating system and co-heating. This indicates that the flip-flop protocol may be usable even in houses with basement ducts, especially if a transition time is provided between tests performed on the same night.
3. The co-heating system was able to control room temperatures very precisely. Thermostat control of the heating system gave mixed results, but appeared to be acceptable in both the houses tested.
4. With respect to equipment efficiency, additional investigation of detailed measurement techniques is recommended. Even if the input-output approach is adopted in Standard 152, stack-efficiency measurements for fuel-fired furnaces are recommended as a check on the results.
5. Distribution efficiencies as measured in the two houses (0.63 and 0.70) are within a typical range established by previous experimental work. Specific support is given to co-heating as a test method.

Acknowledgments

This research was supported by the Building Equipment Division, Office of Building Technologies, U.S. Department of Energy, for which we are grateful. We thank Larry Kinney and Glenn Lewis of Synertech Systems Corporation, Syracuse, New York, for obtaining the use of the Watertown house, for providing a detailed description of the house, and for assistance in setting up the input-output test. A joint report on tests performed by both organizations is in preparation (item 2 in the references). We thank Mark Modera of Lawrence Berkeley Laboratory for several useful suggestions, including the flip-flop protocol. Finally, we thank Esher Kweller of the U.S. Department of Energy for his tireless efforts on behalf of thermal distribution research and for useful discussions on the technical aspects of this work.

References

1. Andrews, J.W. 1993. Test Procedures and Protocols: Their Relevance to the Figure of Merit for Thermal Distribution Systems. BNL-60317, Vol. 2, pp. 20-21.
2. Andrews, J., R. Krajewski, J. Strasser, L. Kinney, and G. Lewis. 1995. Field Test of Heating System Efficiency and Thermal Distribution System Efficiency in a Manufactured Home. Brookhaven National Laboratory Report in Press.
3. Andrews, J.W. 1994. Conceptual Framework for the Thermal Distribution Method of Test. BNL-61462.
4. ASHRAE. 1992. 1992 HVAC Systems and Equipment Handbook, p. 29.16. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
5. ASHRAE. 1993. 1993 Handbook of Fundamentals, Chapter 22. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
6. Caffey, G. 1979. Residential Air Infiltration. ASHRAE Transactions, Vol. 85, Part 1.
7. Cummings, J.B. and Tooley, J.J., Jr. 1989. Infiltration and Pressure Differences Induced by Forced Air Systems in Florida Residences. ASHRAE Transactions, Vol. 95, Part 2.
8. Gammage, R.B.; Hawthorne, A.R.; and White, D.A. 1984. Parameters Affecting Air Infiltration and Air Tightness in Thirty-One East Tennessee Homes. Proc. ASTM Symposium on Measured Air Leakage Performance of Buildings, Philadelphia, PA.
9. Jakob, F.E., D.W. Locklin, R.D. Fischer, L.J. Flanigan, and R.A. Cudnik. 1986. SP-43 Evaluation of System Options for Residential Forced-Air Heating. ASHRAE Transactions, Vol. 92, Part 2B.
10. Jones, R., W. Loss, G. Dennehy, and H. Ghaffari. 1984. Case Study of the Brookhaven House. BNL 51860.
11. Lambert, L.A. and Robison, D.H. 1989. Effects of Ducted Forced-Air Heating Systems on Residential Air Leakage and Heating Energy Use. ASHRAE Transactions, Vol. 95, Part 2.
12. Matthews, T.G.; Thompson, C.V.; Monar, K.P.; and Dudley, C.S. 1990. Impact of Heating and Air Conditioning System Operation and Leakage on Ventilation and Intercompartmental Transport: Studies in Unoccupied and Occupied Tennessee Valley Homes. Journal Air Waste Management Assoc. 40:194-198.
13. Modera, M.P. 1989. Residential Duct System Leakage: Magnitude, Impacts, and Potential for Reduction. 1989. ASHRAE Transactions, Vol. 95, Part 2.
14. Modera, M.P., J. Andrews, and E. Kweiler. 1992. A Comprehensive Yardstick for Residential Thermal Distribution Efficiency. Proc. ACEEE 1992 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Washington, DC.
15. Modera, M.P. 1995. Lawrence Berkeley Laboratory. Personal communications: a, March; b, July.

16. Olson, J.R.; Palmiter, L.; Davis, B.; Geffon, M.; and Bond, T. 1993. Field Measurements of the Heating Efficiency of Electric Forced-Air Systems in 24 Homes. Bonneville Power Administration RCDP Cycle 3 Report.
17. Palmiter, L. and Bond, T. 1992. Impact of Mechanical Systems on Ventilation and Infiltration in Homes. Proc. ACEEE 1992 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Washington, DC.
18. Palmiter, L. and Francisco, P.W. 1994. Measured Efficiency of Forced-Air Distribution Systems in 24 Homes. Proc. ACEEE 1994 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Washington, DC.
19. Parker, D.S. 1989. Evidence of Increased Levels of Space Heat Consumption and Air Leakage Associated with Forced Air Heating Systems in Houses in the Pacific Northwest. *ASHRAE Transactions*, Vol. 95, Part 2.
20. Proctor, J.P. and Pernick, R.K. 1992. Getting It Right the Second Time: Measured Savings and Peak Reduction from Duct and Appliance Repairs. Proc. ACEEE 1992 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Washington, DC.
21. Robison, D.H. and Lambert, L.A. 1989. Field Investigation of Residential Infiltration and Heating Duct Leakage. *ASHRAE Transactions*, Vol. 95, Part 2.

Figure 1. Plan of the Long Island House

Figure 2. Plan of the Watertown House

Figure 3. Hourly Average Room Temperatures, Long Island House.
Top: First Two-Night "Flip-Flop" Test. Bottom: Second Test.

Figure 4. Hourly Average Room Temperatures, Watertown House

Long Island House--March 15-16, 1995

Long Island House--March 16-17, 1995

Figure 5. Basement Temperatures in the Long Island House During Four Nights of Testing

Long Island House--March 17-18, 1995

Long Island House--March 20-21, 1995