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**RESULTS OF A FIELD TEST OF HEATING SYSTEM EFFICIENCY
AND THERMAL DISTRIBUTION SYSTEM EFFICIENCY IN A
MANUFACTURED HOME**

**J. W. ANDREWS, R. F. KRAJEWSKI, AND J. J. STRASSER
BROOKHAVEN NATIONAL LABORATORY**

**L. KINNEY AND G. LEWIS
SYNERTECH SYSTEMS CORPORATION, SYRACUSE, NY**

MAY 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

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

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Under Contract No. DE-AC02-76CH00016**

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

Abstract

A two-day test using electric coheating was performed on a manufactured home in Watertown, New York. The main objective of the test was to evaluate planned procedures for measuring thermal distribution system efficiency. (Thermal distribution systems are the ductwork or piping used to transport heat or cooling effect from the equipment that produces it to the building spaces in which it is used.) These procedures are under consideration for a standard method of test now being prepared by a special committee of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers. The ability of a coheating test to give a credible and repeatable value for the overall heating system efficiency was supported by the test data. Distribution efficiency is derived from system efficiency by correcting for energy losses from the equipment. Alternative means for achieving this were tested and assessed. The best value for system efficiency in the Watertown house was 0.53, while the best value for distribution efficiency was 0.72.

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INTRODUCTION

Brookhaven National Laboratory (BNL) performed a short-term coheating test on a manufactured home in Watertown, New York, in cooperation with Synertech Systems Corporation (Synertech) of Syracuse, New York. The main goal of the test was to evaluate procedures to be used in a standard test method for the efficiency of thermal distribution systems now under development by a committee of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). Thermal distribution systems are the ductwork or piping used to transport heat or cooling from the equipment that produces this thermal energy to the building spaces in which it is used. Details of the planned approach are given in Modera et al. 1994. A more basic description of the parameters used in the test method is given in Andrews 1994. Of primary interest were the feasibility of the test methods and the reasonableness and repeatability of the results.

Furthering the accomplishment of this goal, three specific objectives were pursued:

- o To assess the ability of coheating to obtain reasonable and repeatable values of system efficiency.
- o To evaluate alternative ways of measuring energy losses in the heating equipment, for the purpose of computing distribution efficiency from system efficiency.
- o To learn whether the two technical teams using different equipment would obtain similar results in certain ancillary tests of the building.

BNL has been active in the area of efficient thermal distribution systems for small buildings since the early 1980's. Although this has been a relatively small program area within the U.S. Department of Energy's Office of Building Technologies, it has been shown to have a large potential for energy savings of between one and two Quads (1 Quad = 10^{15} Btu) of primary energy annually. [Andrews and Modera 1991]

As part of its work plan for 1995, BNL constructed an electric coheating system to be used in evaluating the practicability and reliability of the proposed ASHRAE test method (in its current draft form). This coheating system comprises a central data acquisition and processing unit and a multiplicity (8 to 10) of controllable electric heaters, each with an associated temperature sensor consisting of three thermocouple junctions mounted on a laboratory ring stand at heights of 2, 4, and 6 feet above the floor. The coheating system is used to maintain, within the various rooms or zones of a house, a temperature profile that had been maintained by the as-found heating system in an earlier test.

The electric energy needed to maintain this condition is measured and compared with the amount of purchased energy (gas, oil, and/or electricity) needed by the as-found heating system during its test. Assuming that both the coheat and heating system tests are performed under identical conditions, the system efficiency η_{system} is given by the ratio of the coheating energy to the purchased energy needed by the heating system. The degree to which system efficiency falls short of 1.00 (100%) reflects inefficiencies in both the heating equipment and the thermal distribution system. The ASHRAE test method will provide the means to separate the two subsystems, with distribution efficiency as a bottom-line figure of merit for the thermal distribution system.

Synertech has also been active in thermal distribution research for over a decade. Currently, among other efforts, it is participating in a project whose aim is to improve the efficiency of thermal distribution systems in new manufactured homes. As part of this project, six unoccupied homes manufactured in the winter of 1994-95 were instrumented and submitted to a series of testing protocols aimed at assessing overall energy performance of the units and the thermal efficiency of their distribution systems.

Early in 1995, BNL contacted Synertech to explore the possibility of quickly obtaining access to an unoccupied home that: (1) had a duct system (as opposed to a hot-water or other nonducted thermal distribution system); (2) had already been characterized by Synertech; and (3) would be an appropriate test bed for the coheating protocol. Time was of the essence, as the 1995 winter was rapidly approaching its end. One of the homes studied by Synertech was selected, and with the gracious permission of the owner, Hefferon Manufactured Housing, Watertown, New York, arrangements were made for BNL to bring its apparatus to Watertown. The testing took place on March 27-29, 1995.

PHYSICAL DESCRIPTION OF THE TEST HOME

This building is an unoccupied, fully-furnished manufactured home called the Riverbirch Model 092 (with porch). It was manufactured on December 21, 1994 as a double-wide unit in Sangerfield, New York by Titan Homes, Inc. Specifications used in its manufacture comply with the U.S. Department of Housing and Urban Development (HUD) guidelines of October, 1994.

After manufacture was completed (less than a two-day process) the two parts of the building were towed to its permanent site, a development area in a new subdivision three miles south of Watertown, New York. It now rests on a six-inch uninsulated poured concrete pad. Intermediate support is provided by dry stacked concrete block piers. Concrete blocks are also dry stacked to form a skirt along the entire perimeter of the building, which is not

load bearing. Four vent openings (8" x 10") with operable louvers are installed on the long sides of the building while one of the short sides of the building has a small (18" x 36") access opening closed by a hatch-like door.

This floor plan is an open, contemporary style (Figure 1). Kitchen, living, and dining areas are located at one end of the house. These connect with one another through wide openings without doors. The other end of the floor plan contains three bedrooms and two bathrooms. These private areas are compartmentalized with doors for bedrooms, master bath, a walk-in closet, etc. The measured interior floor area is 1585 ft² and the heated volume is 12 690 ft³.

The roof is a 5/12 A-line gable. Walls are framed with 2" x 6" studs except at the interior partition which has doubled 2" x 3" wood framing. The exterior walls are insulated with 6" fiberglass batts and the ceiling has 6" to 8" of blown cellulose. Exterior walls are sheathed with 1/2" compressed wood chip board and are wrapped in building paper. The entire wall assembly is covered with vinyl siding. The roof is sheathed with 1/2" plywood, covered with roofing paper, and finished with 3-tab asphalt shingles.

Interior ceiling surfaces are 1/2" gypsum board, which follows the gable roof line. Wall surfaces are 3/8" finished gypsum or 1/8" simulated wood paneling. Floors in high-use and water-susceptible areas (e.g. kitchen, utility, bath) are typically sheet vinyl. The front-door entryway is floored with a simulated wood material. All remaining floors are carpeted.

Windows are double-glazed, aluminum reinforced vinyl with one tilt sash and one fixed sash. The roof is interrupted by skylights in the sunroom (two), kitchen (two), bath (one) and master bath (one). The skylights are double glazed in aluminum frames. The front and rear doors are metal with insulating foam cores and double-glazed panes. The sunroom has a double-glazed sliding door that opens onto a porch.

Heat is provided by a natural gas-fired, forced-air furnace rated at 96,000 Btu/h input and 75,000 Btu/h output, for a nominal steady-state efficiency of 80%. The Annual Fuel Utilization Efficiency (AFUE) for this model furnace is 75%. (An AFUE lower than the 78% minimum provided by the National Appliance Energy Conservation Act [NAECA] was possible because this is a mobile home.) The furnace is a downflow type with a counterflow heat exchanger. Combustion air from the crawlspace mixes with natural gas at the bottom of the furnace; combustion products flow through a heat exchanger and a double-wall flue pipe through the roof.

Return air enters a louvered door at the top of the furnace, where it is blown via a squirrel-cage fan through the heat

exchanger and then through the bottom of the furnace to the supply plenum. Heated air flows through an enclosed galvanized sheet-metal duct system, which is housed in the floor cavity. The duct system is composed of a trunk line that runs along the long axis of each half of the building with periodic branch lines to 14 registers in the individual rooms (one or two per room). Heat is provided to each room in the building by risers coming directly from the main duct, or sometimes by branch lines that go to the outside wall, typically below a window.

The two parallel trunk ducts are connected by a 12-inch diameter flex duct installed on site. This flex duct rests on grade, on a concrete slab that is the floor of the crawlspace below the building. Return air from the interior is unducted; however, there is a make-up air duct from the roof that feeds outside air into the furnace cabinet. The cross-section of this duct is approximately 1/10th the cross section of the portion of the return air orifice through which house air flows. Combustion air is drawn via an opening from the crawlspace directly below the furnace, and the furnace is constructed so that when the cabinet is closed, the combustion air stream and the conditioned air stream are isolated from each other.

The floor of each half of manufactured homes produced by Titan is composed of a frame of 2" x 6" joists on 16-inch centers glued and stapled to 3/4-inch plywood. During manufacture, this assembly is turned upside down to facilitate installing plumbing and ductwork. The ductwork is fabricated apart from the main assembly line by a single worker. When it is time to install the distribution system, two workers take the pre-assembled system to the upside-down floor, which has holes in place to accept risers from the main trunk. Risers are stapled in place, the trunk itself is strapped to the floor joists, and several *ad hoc* duct runs are completed as necessary. Then 3.5-inch fiberglass blanket insulation is rolled over the floor assembly, keeping the distribution system and the plumbing on the floor side of the insulation.

The insulation in turn is covered by 6 mil fiberglass-reinforced black PVC sheet stock, which is stapled around the edges of the unit. Then a hoist is used to install a pre-fabricated frame, consisting of two large I beams that run the length of the unit about three feet from its edges with lighter weight C-shaped stock as cross members. This assembly, which also includes axles and wheels, is bolted to the bottom of the structure, and the entire bottom is spayed with a sealant. At the next stage of assembly, the floor is turned over, and the unit traverses the remainder of the assembly line on its own wheels. The process from start to finish takes about 12 working hours.

The home at the Watertown site is also heated by a fireplace, which was inactive and sealed throughout the testing.

The electrical system is composed of two 20 A appliance circuits to the kitchen, one 30 A circuit for the electric hot water unit, one 30 amp circuit for the washing machine, a 15 A circuit for the furnace, and 15 A circuits for the remaining lights and outlets in the building. Rated service to the whole building is 100 A.

The building is intentionally ventilated by small manually-controlled fans in both baths, and a manually-controlled range hood fan. There is a small (6" x 6") louvered vent in the laundry area adjacent to the furnace. During the tests, the vent directly connected from the top of the furnace through a duct to the roof was sealed with tape.

GENERAL DESCRIPTION OF TEST

System efficiency tests using coheaters were performed in the Watertown house during the nights of March 27 and 28. During each night's testing, the heating system was operated for six hours and the coheating system for six hours. On the first night, the heating system was tested first; on the second night, the order of tests was reversed.

Eight coheaters were distributed throughout the house as follows: two in the kitchen area, two in the living area, and four in the bed-bath area. These heaters were operated under computer control to maintain a room temperature within 0.5 F of setpoint (1.0 F deadband) as measured for each heater by a stack of 3 thermocouples at heights of 2, 4, and 6 ft above the floor. The heaters were placed as near as possible to the supply register locations, but in most instances compromises had to be made because there were more supply registers than heaters. All but one of the heaters were placed within 2 ft of an exterior wall; the thermocouple stacks were placed in more central locations in their rooms, out of the direct path of air from the heaters and supply registers. Six of the heaters were ordinary 1300 W electric-resistance space heaters; the other two were 500 W halogen lights used in small rooms. These were used to avoid overloading circuits in the house.

For the first night's test, the heating system (furnace and ducts) was monitored from 6 to 12 p.m. Then at midnight the system was changed over to coheating, with the heaters instructed to maintain the same average temperatures in their respective zones as were provided by the duct system. The coheat test was run until 6 a.m. After that, the coheaters continued to run through the day in preparation for the second night's testing from 6 p.m. to midnight. Finally, the house was returned to the heating system, which received its second test during the early morning hours.

Figure 1. Floor Plan of Watertown Test Home

The ability of the computer control system to maintain a set temperature was confirmed; the average temperature during any hour never varied by more than 0.1 F. On the other hand, the heating system's own thermostat was less able to maintain such tight control. The average temperature of the conditioned space varied by as much as 1 F during the course of a six-hour test (Figure 2). This points up an issue that needs to be considered in refining the test method, namely whether to allow the heating system in the house to be controlled by its own thermostat or to require that it also be under computer control.

We did run into several unanticipated problems which we were fortunately able to overcome with good cooperation between the Synertech and BNL personnel. We had intended to measure gas usage of the furnace by reading the gas meter, but unfortunately the smallest increment on the meter (except for a fast-sweep hand discussed below) was 100 cubic feet, equivalent approximately to one therm or 100,000 Btu at 1000 Btu/ft³. By interpolation one might hope to get one more significant digit, but this is still 10,000 Btu, far too gross a scale to be useful in a six-hour test. The meter did have two sweep hands that went full circle at 1/2 and 2 ft³, which would have been more than accurate enough, but to use these would have required stationing someone outside the house with a flashlight for six hours to count revolutions.

Figure 2. Hourly Average Temperatures Maintained by Heating System and by Coheaters

Instead, we were able to tie into a relay that Synertech had in place already. This relay closed when the burner came on. BNL was able to modify its data acquisition program to acquire and integrate burner on-time. Then, by calibrating the burner input rate manually using the above-mentioned sweep hands on the gas meter, we were able to calculate gas usage over any one-hour period or combination of such periods. At the same time, we also were able to tie into a relay that allowed us to measure fan ontime. This, together with average furnace inlet-outlet temperature difference and measured air flow rate at the plenum gave us a measurement of furnace heat output in real time.

PRELIMINARY TESTS PERFORMED INDEPENDENTLY BY BNL AND BY SYNERTECH

One objective of the testing done at Watertown was to see to what extent certain tests performed by both organizations would agree. Four tests of this nature were performed:

- o Furnace input rate
- o Whole-house air leakage
- o Duct air leakage
- o Furnace air-flow rate

BNL's tests were performed during the March 27-29 time frame during which the coheat tests were performed. Synertech's tests were performed prior to this period.

Furnace Input. Furnace input was measured by recording the time required for the gas meter to record a certain number of revolutions of the indicator dial for which one revolution equaled 2 cubic feet. A heating value of 1000 Btu/ft³ was then used to obtain the input rate, according to the formula:

$$\text{Input Rate (Btu/h)} = \frac{\text{Gas Volume (ft}^3\text{)} \times 1000 \text{ Btu/ft}^3}{\text{Run Time (h)}} \quad (1)$$

BNL's input value, based on usage of 16 cubic feet in a time interval of 604.4 seconds (0.1679h), was therefore

$$16 \times 1000 / 0.1679 = 95\,295 \text{ Btu/h}$$

Synertech's measured input value was 95 746 Btu/h. The difference between the two measured rates was therefore 0.5%.

Subsequent to these measurements, the higher heating value for the gas was obtained from the local gas company. This value was 1019.4 Btu/ft³. On the basis of this information, a corrected value of 97 300 Btu/h was used in subsequent calculations.

Whole-House Leakage. Whole house leakage was measured by Synertech and by BNL using fan pressurization. BNL performed a depressurization test using a four-wall-average outside pressure. (The outside pressure tap was ganged to four plastic hoses of equal length, with the ends of these hoses distributed among the four walls of the house.) The test was done during the day when coheating was in progress, which meant that the registers were sealed at the time. Pressure differences ranged from 15 to 55 pascals. The results are shown on a log-log plot in Figure 3. The best straight line through the test points is given by the equation:

$$\log_{10} CFM = 2.30 + 0.64 \log_{10} \Delta P \quad (2)$$

The coefficient of 0.64 is close to the nominal 0.65 normally expected in such tests.

The whole-house air leakage, reported as a flow rate in cfm at 50 pascals, or CFM_{50} , is then 2440. To obtain an effective leakage area, the flow rate at 4 pascals, or CFM_4 , was computed to equal 485 cfm. This was then multiplied by the factor 0.2835 [ASTM 1992] to obtain an effective leakage area (ELA) of 137 in².

Synertech obtained a CFM_{50} value of 2999, which is 23% higher than the BNL value. However, the registers were not sealed during the Synertech tests.

Duct Leakage. Duct leakage was measured with a fan-pressurization device similar to a blower door but smaller and designed for use on ducts. It will be referred to here as a duct blower. BNL and Synertech each used its own duct blower, but both were from the same vendor (Minneapolis Duct BlasterTM). The test was performed in the same manner by Synertech and BNL.

All registers were sealed and the duct blower was attached to the return opening to the furnace in such a way that no air flow was possible into the sealed duct system except through the duct-blower fan. Two tests were then performed. In the first test, an outside door to the house was opened, so that the pressure in the house would equal that outside. The ducts were then pressurized to 50 pascals. This yielded a total duct air leakage value.

In the second test, the outside door was closed and the house was pressurized to 50 pascals relative to the outside with a blower door at the same time that the sealed duct system was pressurized to 50 pascals using the duct blower. Since the house and the ducts were at the same pressure, any leakage between ducts and house would be neutralized, and only duct leakage to the outside would be measured.

Neither team was present when the other performed its measurements. Calculations were also done independently.

The table below gives the values of CFM₅₀ measured by BNL and Synertech for total duct leakage and duct leakage to the outside:

Duct Leakage Condition	Total Duct Leakage	Duct Leakage to Outside
Organization Performing Test		
BNL	539	246
Synertech	506	233

BNL's measurements averaged 6% higher than Synertech's, which is reasonable agreement for this type of measurement.

Furnace Air Flow Rate. Furnace air flow rate was measured with the duct blower in the following fashion. With the registers unsealed and an outside door to the house open, the air handler was activated and the pressure difference was measured between two fixed points, one in the supply duct and one in the living space. Following this, the air handler was shut down and the duct blower was attached to the return side of the furnace so that no air could enter the air handler except through the duct blower. The air handler was then turned back on and the duct blower fan started up. Its speed was adjusted until the pressure difference between the inside and the outside of the ducts was the same as in the first test without the duct blower. When this condition was reached, the assumption is made that the air flow through the ducts is the same as it was originally. This air flow can be read off from the duct blower, since all the air flowing through the supply ducts has come through the duct blower.

BNL and Synertech each performed this test, on different days and without the other team present. It was found that the pressure in the duct varied significantly depending on whether the louvered outer furnace door was open or closed. The results with the door closed (which is the normal operating condition) as obtained by both organizations, were:

Organization	Furnace Air Flow Rate (cfm)
BNL	933
Synertech	991

The difference between these measurements is 6%. Synertech also performed measurements of air flow through the registers, using the duct blower as a powered flow hood. The sum of these air flows was 817 cfm. Because of time constraints, BNL did not make this measurement.

Figure 3. Whole-House Air Leakage Rate vs. Indoor-Outdoor Pressure Difference

SYSTEM EFFICIENCY

Probably the most important of the three objectives of this test was to evaluate coheating as a method of determining system efficiency. The methodology employs an average of two nights' testing with reversal of the order of coheating and heating system operation. This should give a better result than one night's testing alone, especially in buildings with high thermal mass. Even in a house such as the one in Watertown, which has low thermal mass and thus perhaps little bias from that cause, increasing the number of tests from one to two reduces the random variation expected in the result by approximately 30% $[(1-1/\sqrt{2}) \times 100]$.

For the first night's test, the heating system was monitored from 6 to 12 p.m. The average room temperatures maintained at the locations of the eight thermocouple stacks ranged from 68.5 to 72.8 F. Then at midnight the system was changed over to coheating, with the heaters instructed to maintain these same average temperatures in their respective zones. The registers were sealed for the coheating tests, a process that was completed by 12:50 a.m. The coheat test was run until 6 a.m. After that, the coheaters continued to run through the day in preparation for the second night's testing.

The windows were covered with aluminum foil during the first night, those on the east side being done after the registers were sealed and the remainder early the next morning. We had some concern that this had not been done before the first heating system test, but time constraints did not permit it to be done any earlier.

Computation of system efficiency used the following steps, intended to be applicable regardless of whether the heating equipment is fueled by gas or oil or served by electric power. It is assumed that (1) no gas or oil is used other than for the heating system during the test; and (2) all electric power is used either to operate the heating equipment or is dissipated internally in the house as lighting or appliance power. In particular, it is assumed that the water heater is turned off (whether gas or electric), that no other gas appliances are operating during the test, and that no outside electric lights or appliances are used during the test.

The following procedure for calculating system efficiency was prepared by one of us (Andrews) as part of a draft protocol for review by ASHRAE SPC152P in June, 1995.

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:

$$HLC = \frac{H_{coheat} + H_{ig-coheat}}{(T_c - T_a)_{coheat} t_{coheat}} \quad (3)$$

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

2. If it has not already been done, compute $H_{ig-system}$, 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 power or energy per unit time) as the ratio of $H_{ig-system}$ 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} :

$$\Delta T_{io-adj} = (T_c - T_a)_{system} - \Delta T_{bal} \quad (4)$$

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

$$HSC = \frac{G_{system} + E_{system}}{\Delta T_{io-adj} t_{system}} \quad (5)$$

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.

The detailed calculations are shown in the charts on the next two pages. The following is a summary of the results:

System Efficiency η_{system} , First Night = 0.506

System Efficiency η_{system} , Second Night = 0.528

Average Value of η_{system} = 0.517

Error Analysis. We can use the difference between the two measured values of system efficiency as an indicator of the random error involved in the measurements. One major difficulty with this is that, having only two measured values, the error bars tend to be large. Also, systematic error is not addressed by this procedure. Nevertheless, it is instructive to obtain 80% and 90% confidence intervals using the t-distribution.

The sample standard deviation s ($n=2$) of the data set consisting of the values 0.506 and 0.528 is 0.016. A confidence interval (plus or minus) can be obtained from this using the critical value of t for the desired confidence level, according to the equation

$$t = \frac{\bar{x} - \mu}{s/\sqrt{n}} \quad (6)$$

where the number of degrees of freedom is $n-1$.

The critical values of t for one degree of freedom are 3.078 (80% confidence) and 6.314 (90% confidence). The corresponding error bars (plus or minus) are given by $\bar{x} - \mu$ in the above equation, or equivalently by $ts/\sqrt{n}$:

80% confidence: $3.078 \times 0.016 / 1.414 = 0.035$

90% confidence: $6.314 \times 0.016 / 1.414 = 0.071$

A "bottom up" error analysis based on the measured quantities going into the calculation of system efficiency needs to be done. The theoretical analysis is underway, but is beyond the scope of this report. So, provisionally, we adopt the 80% confidence interval above, and quote:

$$\eta_{\text{system}} = 0.517 \pm 0.035 \quad (80\% \text{ confidence})$$

Watertown, New York
First Night Test, March 27-28

Step 1 -- Heating Load Coefficient (HLC) Using Coheat Data

Average indoor temperature: 71.6 F
Average outdoor temperature: 29.0 F $(T_c - T_a)_{\text{coheat}} = 42.6 \text{ F}$
Test interval duration t_{coheat} : 6.22 h
Electric energy to house = 25.7 kWh Body heat = 600 Btu
Coheat plus internal $H_{\text{coheat}} + H_{\text{ig-coheat}} = 25.7 \times 3415 + 600$

$$HLC = \frac{25.7 \times 3415 + 600}{(71.6 - 29.0)(6.22)} = 333 \text{ Btu/h-F}$$

Step 2 -- Internal Gains During Duct System Test

Electric Energy to House = 5.9 kWh Body Heat = 0
Furnace Fan Energy = 0.2 kW X 1.62 h = 0.32 kWh
Net Internal Gain $H_{\text{ig-system}} = (5.9 - 0.32) \times 3415 + 0 = 19\ 060 \text{ Btu}$
Test interval duration t_{system} = 5.97 h
IGR = 19 060 Btu/ 5.97 h = 3190 Btu/h

Step 3 -- Adjusted Indoor-Outdoor ΔT --Duct System Test

$$\Delta T_{\text{bal}} = \text{IGR}/\text{HLC} = 3190/333 = 9.6 \text{ F}$$

Average indoor temperature: 71.9 F
Average outdoor temperature: 38.3 F

$$\Delta T_{\text{io-adj}} = 71.9 - 38.3 - 9.6 = 24.0 \text{ F}$$

Step 4--Heating System Coefficient (HSC) for Duct System Test

Energy to Furnace:
Gas burner ontime 0.9619 h X 97 300 Btu/h input $G_{\text{system}} = 93\ 600 \text{ Btu}$
Furnace Fan Energy 0.32 kWh X 3415 Btu/kWh $E_{\text{system}} = 1\ 100 \text{ Btu}$

Interval over which gas input was measured = 6.00 h

$$HSC = \frac{93\ 600 + 1100}{24.0 \times 6.00} = 658 \text{ Btu/F-h}$$

Step 5--System Efficiency

$$\eta_{\text{system}} = \text{HLC}/\text{HSC} = 333/658 = 0.506$$

Watertown, New York
Second Night Test, March 28-29

Step 1 -- Heating Load Coefficient (HLC) Using Coheat Data

Average indoor temperature: 71.6 F
Average outdoor temperature: 37.8 F $(T_c - T_a)_{\text{coheat}} = 33.8 \text{ F}$
Test interval duration t_{coheat} : 6.03 h
Electric energy to house = 21.7 kWh Body heat = 0
Coheat plus internal $H_{\text{coheat}} + H_{\text{ig-coheat}} = 21.7 \times 3415 + 0$

$$HLC = \frac{21.7 \times 3415 + 0}{(71.6 - 37.8)(6.03)} = 364 \text{ Btu/h-F}$$

Step 2 -- Internal Gains During Duct System Test

Electric Energy to House = 5.6 kWh Body Heat = 400 Btu
Furnace Fan Energy = 0.2 kW X 1.94 h = 0.39 kWh
Net Internal Gain $H_{\text{ig-system}} = (5.6 - 0.39) \times 3415 + 400 = 16 \text{ 510 Btu}$
Test interval duration $t_{\text{system}} = 5.50 \text{ h}$
IGR = 16510/5.50 h = 3000 Btu/h

Step 3 -- Adjusted Indoor-Outdoor ΔT --Duct System Test

$$\Delta T_{\text{bal}} = \text{IGR}/HLC = 3000/364 = 8.2 \text{ F}$$

Average indoor temperature: 70.8 F
Average outdoor temperature: 30.9 F

$$\Delta T_{\text{io-adj}} = 70.8 - 30.9 - 8.2 = 31.7 \text{ F}$$

Step 4--Heating System Coefficient (HSC) for Duct System Test

Energy to Furnace:
Gas burner ontime 1.2219 h X 97 300 Btu/h input $G_{\text{system}} = 118 \text{ 900 Btu}$
Furnace Fan Energy 0.39 kWh X 3415 Btu/kWh $E_{\text{system}} = 1 \text{ 300 Btu}$
Interval over which gas input was measured = 5.50 h

$$HSC = \frac{118 \text{ 900} + 1300}{31.7 \times 5.50} = 689 \text{ Btu/F-h}$$

Step 5--System Efficiency

$$\eta_{\text{system}} = HLC/HSC = 364/689 = 0.528$$

DISTRIBUTION EFFICIENCY

Of all the factors defined in the 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. 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 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 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 in the heating system being tested 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 because with the existing distribution system it will have longer run times than if distribution losses were eliminated. With the existing distribution system it will therefore have larger fractional on-times, with consequent reduction in cycling losses and improvement in efficiency. It is important to note that the deviation of F_{equip} from 1.00 is expected to be small.

Equipment Efficiency. For most types of equipment, 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 Via stack temperature and CO_2 -concentration measurements of steady-state efficiency, with correction factors for cycling and jacket losses; and
- o Via 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 on stack measurements or to depend entirely on input-output tests.

Our experience in the Watertown house suggests that stack measurements should at least be used as a check on the input-output data. Whether greater reliance than this should be placed on such measurements must be based on a judgment of how accurate the input-output measurements can be, if carefully done, relative to the accuracy possible in steady-state measurements coupled with corrections for cycling. The following discussion of results of measurements made by Brookhaven and Synertech on this house may shed some light on this question.

Basically, the input-output approach to measuring equipment efficiency involves measuring the temperature difference across the plenum and integrating this over time, weighted by air flow rate. (If it can be assumed that the air flow rate is either zero or a single fan-on value, then the temperature difference itself can be integrated over those times when the fan is on and the result multiplied by the fan-on flow rate. This flow rate may be measured separately, e.g. by using a duct blower. This integral, multiplied by the specific heat of air per unit volume, gives the thermal output in Btu or joules for the test period.

The fossil fuel input to the furnace is found by measuring the weight of fuel used by the furnace and multiplying that by the higher heating value of the fuel. As discussed above, this was achieved in the present case by measuring the gas input rate and multiplying that by the burner ontime. The heating value of the electrical input to the furnace fan was estimated by multiplying the fan ontime by the thermal equivalent of the motor horsepower:

$$(1/3 \text{ hp} \times .746 \text{ kW} \times 3415 \text{ Btu/kWh})$$

and dividing by 0.85 to allow a reasonable estimate of motor efficiency.

Return- and supply-side temperature measurements were performed as follows. Because the furnace had no return ductwork, we used a single thermocouple located just outside the return grillwork of the furnace as our indicator of return temperature. On the supply side, a thermocouple was inserted into the air stream leaving the furnace. The thermocouple wire was embedded in a small block (approx. 2 inches on a side) of rigid foam insulation just behind the junction, with the junction protruding about 1 inch beyond it. This was to keep the junction from making physical contact with the duct wall. The thermocouple wire was then attached to a flexible steel tape measure and inserted into the duct. This allowed it to be placed at a distance from the register corresponding to the position of the furnace outlet.

We identified the following potential sources of experimental error associated with this approach:

1. The air leaving the furnace may not be at a uniform temperature over the cross section of the supply duct where the measurements are made. Also, simply measuring the temperature at several points and averaging these may not help, since the air velocity is likely to be nonuniform as well.

With respect to this issue, we were not able to perform any kind of cross-sectional averaging of flow rates and temperatures, due to time limitations. Only a single thermocouple was used to measure the temperature of warm air leaving the furnace. The SPC152P protocol will require a more extensive procedure for measuring this temperature.

2. The air flow rate as measured by the duct blower (or other method) may be inaccurate.

This flow rate was measured by both BNL and Synertech, with a 6% difference between the two groups' results (as already discussed).

3. Even if the average temperatures in the air stream into and out of the plenum are measured precisely, and the fan-on air flow rate is also measured, the resulting value for heat out of the equipment may not capture useful heat delivered during the fan-off period.

With the commonly used upflow furnace, heat may flow upward from the furnace into the supply ducts during the fan-off period by natural convection. Should the furnace get credit for this? In Watertown, where the furnace was a downflow model located within the conditioned space, we sensed what seemed like a significant flow of heat into the utility room when the fan was off, but we were not able to quantify this effect.

Despite these caveats, we obtained measured values, for eleven half-hour periods during the early morning hours of March 29, of burner fractional ontime, air-handler fractional ontime, average supply-plenum temperature, average temperature at the furnace return grille, and the difference between the two temperatures. These are given in Table 1.

The fuel input energy was computed from the average burner fractional ontime and the measured fuel input rate:

$E_{in} = \text{Input Rate} \times \text{Burner Fractional Ontime} \times \text{Test Duration}$

$= 97\,300 \text{ Btu/h} \times 0.2220 \times 5.5 \text{ h} = 118\,800 \text{ Btu}$

Table 1. Furnace Input-Output Quantities--Measured Values

Time at End of Half Hour	f_{burn}	f_{fan}	$T_{\text{supply-avg}}$ (F)	$T_{\text{return-avg}}$ (F)	$\Delta T_{\text{supply-return}}$ (F)
0:37	0.2564	0.3333	122.91	72.48	50.43
1:07	0.2827	0.4450	124.90	72.96	51.94
1:37	0.1979	0.3542	121.75	73.61	48.14
2:07	0.2062	0.3247	124.17	73.75	50.42
2:37	0.2539	0.3523	122.53	73.66	48.87
3:07	0.2448	0.3906	125.77	73.79	51.98
3:37	0.1979	0.3854	121.11	74.01	47.10
4:07	0.1969	0.3212	122.75	73.85	48.90
4:37	0.2021	0.3212	122.85	73.85	49.00
5:07	0.1959	0.3247	122.14	73.84	48.30
5:37	0.2073	0.3212	124.01	74.05	49.96
Averages 5.5 hr	0.2220	0.3522			49.55

The thermal output from the furnace during the same period was calculated from the air-handler fractional ontime, the fan-on air flow rate, and the average supply-return temperature difference:

$$E_{\text{out}} = 1.08 \times \text{Fan-On Air Flow (cfm)} \times \text{Fan Fractional Ontime} \times \text{Test Duration (h)} \times \text{Average Supply-Return Temperature Difference (F)}$$

where 1.08 is the product of the specific heat of air per unit volume and a factor of 60 to convert cfm to ft³/h. Using the air flow rates measured by BNL and Synertech, we obtain, respectively:

$$E_{\text{out-1}} = 1.08 \times 933 \times 0.3522 \times 5.5 \times 49.55 = 96\,700 \text{ Btu}$$

$$E_{\text{out-2}} = 1.08 \times 991 \times 0.3522 \times 5.5 \times 49.55 = 102\,700 \text{ Btu}$$

Equipment efficiency η_{equip} for the test interval is equal to the ratio of E_{out} to E_{in} . It equals 0.814 if the BNL-measured air-flow rate is assumed, and 0.864 if the Synertech-measured air-flow rate is used.

As a benchmark, we wanted to compare these values with the steady-state efficiency of the furnace. Normally, we would expect the furnace efficiency under cycling conditions to be lower than the steady-state value. As measured here, the difference would

include the jacket losses (which are losses in the input-output method but not in the steady-state efficiency determination) as well as an allowance for off-cycle stack losses. This difference would normally be in the range 1 - 6 percentage points for burner fractional ontimes close to the 22.5% used in determining Annual Fuel Utilization Efficiency (AFUE).

What is the steady-state efficiency of the furnace in this test house? We have two indicators: the ratio of the nominal output and input values, and the result of a stack efficiency measurement.

The nominal output and input rates are 76 000 Btu/h and 95 000 Btu/h, respectively, which corresponds to a steady-state efficiency of 80%.

Steady-state efficiency was measured using stack temperature and CO₂ concentration values obtained via repeated measurements with a Bacharach test kit. Two separate runs are summarized as follows:

Furnace Door Condition	Furnace Room Temperature (F)	Stack Temperature (F)	Percent CO ₂	Steady-State Eff'y.
Closed	85	582	9.0	77.3%
Open	86	578	8.4	77.0%

Given the range of measured and nominal values for steady-state efficiency in the high 70's, we consider it unlikely that the cycling efficiency of the equipment, net of jacket losses, should be in the 80's. Nevertheless, in view of the "first-cut" nature of our input-output measurement, it might be considered something of a victory that we got an equipment efficiency, based on raw data, this close to what we would expect from the stack efficiency. Nevertheless, our experience convinces us that a steady-state efficiency test will be an extremely valuable check on the input-output measurements, even if the latter are done very carefully.

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} . We note that this is consistent with the quoted AFUE for this furnace.

Equipment Factor. The one additional item needed to calculate distribution efficiency is the equipment factor. An equation was developed in the fall of 1994 by one of us (Andrews) for proposed

use in SPC152 for furnaces. This equation, based on SP-43 data, was:

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

(The derivation of this equation from ASHRAE data is given in Appendix 1.) Using our best value of 0.75 for η_{equip} and our average measured value of 0.517 for η_{system} , we obtain an estimated equipment factor of 1.027.

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

$$\begin{aligned} \eta_{\text{dist}} &= \frac{\eta_{\text{system}} F_{\text{equip}}}{\eta_{\text{equip}}} \\ &= \frac{0.517 \times 1.027}{0.75} \\ &= 0.707 \end{aligned}$$

Error Analysis of Stack-Efficiency Approach. The upshot of these considerations is that directly measuring equipment efficiency using input-output methods may be usefully supplemented by a value of distribution efficiency that can be obtained from:

1. A careful measurement of system efficiency using coheating.
2. A careful measurement of steady-state efficiency using stack temperature and CO_2 .
3. A reasonable estimate of cycling losses and (if the furnace is in an unconditioned space) jacket losses.
4. Equipment factor calculated according to Equation 4.

The argument for this is based on the relative magnitudes of the errors involved in these quantities. The following will be used as baseline assumptions:

1. We believe that coheating will be able to deliver an expected error in system efficiency, to a reasonable confidence level (80% to 95%), on the order of 5% of the measured value. Reducing the error much below this value is likely to involve greatly increasing difficulty. In the benchmark calculation below, we will assume that system efficiency has been measured as 0.50 ± 0.025 (a 5% error bar).

2. Steady-state efficiency can reliably be measured within a percentage point using stack temperature and CO₂ concentration. We will assume that η_{ss} has been measured as 0.80 ± 0.01 .

3. For most mid-efficiency furnaces (meeting NAECA standards), cycling losses may be estimated to fall in a range between 0.5 and 2.5 percentage points, with a best estimate of 1.5 percentage points (67% error bar).

4. For the same furnaces as in (3), jacket losses may be estimated to fall in a range between 1 and 5 percentage points, with a best estimate of 3 percentage points (67% error bar).

Because the furnace in this test was in the conditioned space, jacket losses should be very small or zero. However, this is not the usual case, and because we would like this error analysis to be representative of a worst-case scenario for the use of stack measurements, we will perform an error analysis that is consistent with the above estimates for a furnace that is outside the conditioned space.

We analyze the errors using the following equations:

$$\eta_{dist} = \eta_{system} \frac{F_{equip}}{\eta_{equip}} \quad (9)$$

$$\eta_{equip} = \eta_{ss} - \Delta\eta_{cyc} - \Delta\eta_{jac} \quad (10)$$

where Equation 5 is the basic relation between these quantities and Equation 6 defines the corrections to steady-state efficiency due to cycling and jacket losses, respectively. (Note that for equipment located in the conditioned space, $\Delta\eta_{jac} = 0$.)

Combining these equations yields:

$$\begin{aligned} \eta_{dist} &= \frac{\eta_{system} F_{equip}}{\eta_{ss} - \Delta\eta_{cyc} - \Delta\eta_{jac}} \quad (11) \\ &= \frac{\eta_{system} F_{equip}}{\eta_{ss} \left(1 - \frac{\Delta\eta_{cyc}}{\eta_{ss}} - \frac{\Delta\eta_{jac}}{\eta_{ss}}\right)} \\ &\approx \frac{\eta_{system} F_{equip}}{\eta_{ss}} \left(1 + \frac{\Delta\eta_{cyc}}{\eta_{ss}} + \frac{\Delta\eta_{jac}}{\eta_{ss}}\right) \end{aligned}$$

Let us now calculate the error in η_{dist} . Because the percent error on η_{ss} is much less than the percent errors on cycling and jacket efficiencies, it can be ignored in taking the percent errors in the ratios

$$\frac{\Delta\eta_{cyc}}{\eta_{ss}} = \frac{0.015 \pm 0.01}{0.80} = 0.019 \pm 0.013$$

$$\frac{\Delta\eta_{jac}}{\eta_{ss}} = \frac{0.03 \pm 0.02}{0.80} = 0.038 \pm 0.025$$

The error in the quantity $(1 + \Delta\eta_{cyc}/\eta_{ss} + \Delta\eta_{jac}/\eta_{ss})$ is the square root of the sum of the squares of the errors of the terms:

$$(0^2 + 0.013^2 + 0.025^2)^{1/2} = 0.028$$

(Note that the term 1 always has error equal to zero.) The percentage error on this quantity is then $0.028/(1+0.019+0.038) = 0.027$ or 2.7%.

The assumptions in our baseline calculation are now summarized in Table 2. Note that these are intended to be representative of a typical duct system and a minimum-efficiency NAECA-compatible furnace not in the conditioned space, which is not the same as the situation in the Watertown house.

Table 2. Best Values and Error Bars For Input Quantities to Distribution Efficiency Calculation

Quantity	Best Value	Error Bar	Percent Error
η_{system}	0.50	0.025	5.0%
F_{equip}	1.027	0.018	1.8% (67% of deviation from unity)
η_{ss}	0.80	0.01	1.25%
$\Delta\eta_{cyc}$	0.015	0.01	67%
$\Delta\eta_{jac}$	0.03	0.02	67%

The percent error on η_{dist} is now the square root of the sum of the squares of the percent errors on the four quantities on the last line of Equation 11 (treating the parentheses as a single quantity):

$$(5.0^2 + 1.8^2 + 2.5^2 + 1.25^2)^{1/2} = 6.0$$

The same calculations can be carried out for other assumed errors in the system efficiency. These are displayed in Table 3.

Table 3. Percentage Errors in Distribution Efficiency Resulting from Various Assumptions Concerning the Percentage Error in System Efficiency.

Percentage Error in η_{system}	Percentage Error in η_{dist}
5.0	6.1
4.0	5.3
3.0	4.6
2.0	4.0
1.0	3.6
0.0	3.5

If the percentage error in system efficiency is relatively high, the use of stack-efficiency measurements will not add appreciably to it. If, on the other hand, the percentage error in system efficiency is low, then the stack-efficiency errors will not produce an unacceptably high baseline error (3.5%), either. (Note that these are percentages, not percentage-points.)

If we apply this analysis to the actual data from the Watertown house, where

- o The best value of η_{system} was found to be 0.517 ± 0.035 (80% confidence).
- o The calculated equipment factor was 1.027. We will assume an error of 67% of the deviation of this factor from unity, or 0.018.
- o The measured steady-state efficiency was 0.77. We will assume a one-percentage-point accuracy, or 1.3% of the measured value.
- o We have assumed 2% cycling losses, partly in recognition of the 75% AFUE for the furnace. We will assume an error bar of 67% of this value.
- o Finally, jacket losses are taken to equal zero because this furnace was in the heated space.

The value of $(1 + \Delta\eta_{cyc}/\eta_{ss} + \Delta\eta_{jac}/\eta_{ss})$ is then equal to 1.026 ± 0.017 . Together with the values of system efficiency, stack efficiency, and equipment factor as given above, these yield a distribution efficiency equal to

$$\eta_{dist} = \frac{0.517 \times 1.027 \times 1.026}{0.77} = 0.707$$

with error equal to

$$\frac{\Delta\eta_{dist}}{\eta_{dist}} = [(\frac{0.035}{0.517})^2 + (\frac{0.018}{1.027})^2 + (\frac{0.017}{1.026})^2 + (\frac{0.01}{0.77})^2]^{1/2}$$

or $\eta_{dist} = 0.707 \pm 0.052$.

To summarize, the arithmetic of quadrature ensures that as long as the other percentage errors in the multiplication/division problem are significantly less than the percentage error in system efficiency, it doesn't matter much what they are. In other words:

- o Beyond a certain point, there isn't much to gain in measuring one thing much more accurately than another.

Also:

- o Even a very large percentage error (such as 67%) may be insignificant if the term it applies to is added to a much larger one.

CONCLUSIONS

The principal insights gained from this test were:

- o The data and experience support the proposition that coheating can be a reliable means of obtaining the system efficiency.
- o For fuel-fired furnaces, equipment efficiency should be measured by a combination of input-output methods supplemented by an independent measurement of steady-state efficiency using flue-gas temperature and CO₂ content.

REFERENCES

Andrews, J.W. Conceptual Framework for the Thermal Distribution Method of Test. BNL-61462, November 1994.

ASTM 1992. Standard Test Method for Determining Air Leakage Rate by Fan Pressurization. E779-87 (Reapproved 1992). American Society for Testing and Materials, Philadelphia, PA. The factor 0.2835 is obtained from the equations for flow rate and leakage area in Sections 9.4.1 and 9.4.2 of this Standard, and applying the appropriate units conversion.

Jakob, F.E., D.W. Locklin, R.D. Fischer, L.J. Flanigan, and Cudnik, R.A. SP43 Evaluation of System Options for Residential Forced-Air Heating. ASHRAE Trans., V. 92, Pt. 2, 1986.

Modera, M.P., J. Andrews and E. Kweiler. A comprehensive yardstick for residential thermal distribution efficiency. Proc. ACEEE 1994 Summer Study on Energy Efficiency in Buildings. Asilomar, CA, August 1994.

APPENDIX 1. COMPUTED EQUIPMENT FACTOR FOR FUEL-FIRED FURNACES

The ASHRAE SP-43 Project (Jakob et al. 1986) obtained the following data from their validated simulation model for a house in Pittsburgh, with various levels of duct insulation and leakage (notation per SP-43):

Duct Effic'y. [E _D]	Load Mod. Factor [F _{LM}]	Inverse Product [E _D F _{LM}] ₋₁	Furnace Effic'y. [E _F]	Regression Value of E _F (See Text)
0.796	0.84	1.495	0.750	0.7496
0.774	0.85	1.520	0.750	0.7506
0.704	0.91	1.560	0.752	0.7521
0.668	0.94	1.592	0.754	0.7532
0.632	0.97	1.631	0.754	0.7546
0.609	1.00	1.642	0.755	0.7549 [SP-43 Base Case]
0.548	1.07	1.706	0.757	0.7569

The furnace was sized to 1.4 times the design heating load, in accordance with ASHRAE recommendations at the time of the experiment.

In general, furnace efficiency increases when the fractional on-time of the furnace increases. This means that as ducts become less efficient, there is some compensatory increase in the efficiency of the furnace.

This variation in furnace efficiency is expected to be a function of the heat demand on the furnace. This demand for heat varies as the inverse of the product [E_DF_{LM}F_{MG}]. The table presented by Jakob gives the first and second of these quantities, but not the third. A back-calculation enabled us to determine that variation in F_{MG} was much smaller than those of the other two. (As E_D varied 45% and F_{LM} varied 27%, F_{MG} varied by only 1%.) We therefore use the inverse product of F_{LM} and E_D as the independent variable in a regression analysis, with E_F as the dependent variable.

Regression Analysis of SP-43 Case. The regression line through the data of the table is

$$E_F = 0.6978 + 0.034741 [E_DF_{LM}]^{-1} \quad (A1-1)$$

The predicted values of E_F are shown as the last column of the table.

It is now necessary to account for the fact that in the SP-43 project, their base case was not the same as the one used in SPC152P. SPC152P considers the base case to be one with as-found equipment but with a perfect distribution system, which we denote

as NELI for a "no energy loss or impact" duct system. SP-43's base case, on the other hand, was simply one of the cases having a real distribution system, selected as typical.

A word of caution is in order at this point. The SP-43 Project used the symbols E_F , E_D , F_{LM} , and F_{MG} for furnace efficiency, duct efficiency, load modification factor, and miscellaneous gain factor, respectively. For the purposes of the Standard Method of Test, we are using a slightly different system in which F_{MG} is not used but instead direct heat inputs to the load from the equipment or the distribution system, whether or not these occur through "approved" flow paths, are included in the load factor.

The other major difference, as stated above, is the different choice of base-case system. This has consequences for the load factor. F_{load} is the same as SP-43's F_{LM} except that here F_{load} is defined to equal 1.00 for the "NELI" base case with a perfect distribution system, whereas for SP-43, F_{LM} was defined to equal 1.00 for a different base case, which had a real duct system. This difference must be handled with care.

In order to get our base case from the SP-43 data, we need to extrapolate to 100% duct efficiency. If we create a new regression line with E_D as the independent variable and $[E_D F_{LM}]^{-1}$ as the dependent variable, then

$$[E_D F_{LM}]^{-1} = 2.14887 - 0.82352 E_D \quad (A1-2)$$

The predicted value of $[E_D F_{LM}]^{-1}$ for $E_D = 1.00$ is then 1.32535, or $F_{LM} = 0.7545$. Substituting this value into Equation A1-1, we obtain $E_F = 0.7438$.

We now need to "renormalize" Equation A1-2 to account for the different base case. That is, $F_{load} = 1.00$ for the same case for which $F_{LM} = 0.7545$, or in other words, F_{load} is always equal to 1.32535 times F_{LM} . The resulting equation, using now the symbols for our current standard method of test, is

$$\eta_{equip} = 0.7438 + (0.034741)(1.32535)([\eta_{del} F_{load}]^{-1} - 1) \quad (A1-3)$$

which, rounding to three decimal places, is

$$\eta_{equip} = 0.744 + 0.046 ([\eta_{del} F_{load}]^{-1} - 1) \quad (A1-4)$$

This agrees with the SP-43 data for our "perfect distribution" base case, for which, with the help of Equation A1-2 we obtained $E_F = 0.7438$, since the second term on the right hand side goes to zero. Checking SP-43's base case, the renormalized F_{load} will equal 1.32535 X 1.00. Using $\eta_{del} = E_D = 0.609$ (i.e. ignoring F_{MG} as we have been doing), we obtain $\eta_{equip} = 0.7548$, very close to the SP-43 value for E_F .

Generalization to Other Furnace Types. If the foregoing is accepted provisionally, we can note that the variation of furnace efficiency with the product $\eta_{del} F_{load}$ was computed for a specific furnace that was approximately 75% efficient. We can go on to argue that this variation will decrease for higher-efficiency furnaces. For a 100% efficient furnace, for example, the variation should clearly be zero, as it is for an electric furnace. If we pro-rate the amount of this variation on the basis of how far the as-found furnace efficiency deviates from 100%, then the decrease in furnace efficiency in a "perfect" duct system will be given by

$$\begin{aligned} & 0.046 ([\eta_{del} F_{load}]^{-1} - 1) (1 - \eta_{equip}) / (1 - 0.75) & (A1-5) \\ & = 0.18 ([\eta_{del} F_{load}]^{-1} - 1) (1 - \eta_{equip}) \end{aligned}$$

where the factor $0.046 ([\eta_{del} F_{load}]^{-1} - 1)$ is the second (variable) term on the right-hand side of Equation A1-4, and $(1 - \eta_{equip}) / (1 - 0.75)$ is a simple pro-rating factor comparing the inefficiency of the furnace that is part of the system being tested under this Standard with the inefficiency of the furnace that was used in the SP-43 tests.

Consequently, the predicted furnace efficiency in NELI is:

$$\eta_{equip-NELI} = \eta_{equip} - 0.18 ([\eta_{del} F_{load}]^{-1} - 1) (1 - \eta_{equip}). \quad (A1-6)$$

This is very nearly equivalent to

$$\eta_{equip} / \eta_{equip-NELI} = 1 + 0.18 ([\eta_{del} F_{load}]^{-1} - 1) (1 - \eta_{equip}) / \eta_{equip}. \quad (A1-7)$$

The ratio $\eta_{equip} / \eta_{equip-NELI}$ is precisely the definition of F_{equip} .

We can now make use of the fact that $\eta_{system} = \eta_{equip} \eta_{del} F_{load}$ and rewrite the equation

$$\begin{aligned} F_{equip} &= 1 + 0.18 (\eta_{equip} / \eta_{system} - 1) (1 - \eta_{equip}) / \eta_{equip}. & (A1-8) \\ &= 1 + 0.18 (1 - \eta_{equip}) (1 / \eta_{system} - 1 / \eta_{equip}). \end{aligned}$$

The table below displays values of F_{equip} that result from various values of system and equipment efficiency. The deviation from 1.00 is not large.

Table A1-1. Values of F_{equip} That Result from Various Choices of System Efficiency, Equipment Efficiency, and Furnace Location.

η_{system}	η_{equip}	F_{equip}
0.50	0.55	1.015
	0.65	1.029
	0.75	1.030
	0.85	1.022
	0.95	1.008
0.60	0.65	1.008
	0.75	1.015
	0.85	1.013
	0.95	1.006
0.70	0.75	1.004
	0.85	1.007
	0.95	1.003
0.80	0.85	1.002
	0.95	1.002