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HEAT BALANCE and EFFICIENCY MEASUREMENTS of CENTRAL, FORCED-AIR, RESIDENTIAL GAS FURNACES

E. C. Hise

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E. C. Hise, P.E.
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OCTOBER 1975

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ABSTRACT

Residential central gas furnaces of the atmospheric combustion and sealed combustion types were tested in the laboratory and in residences. Complete heat balances were taken at steady-state operation, cyclic operation, and off-design conditions, and the instantaneous efficiencies were determined. The seasonal fuel utilization efficiencies were synthesized for the two types of furnaces and for a range of adjustments in four climates. Recommendations are offered for a high-efficiency furnace design and installation standards which will reduce fuel consumption by one-third.

1. INTRODUCTION

The purpose of this investigation was to identify and quantify the inefficiencies of installed residential gas furnace systems rather than to state efficiencies of gas furnace systems. There are many manufacturers, capacities, and types of furnaces, as well as a variety of installations, residences, occupancy habits, degrees of maintenance, and climates. Thus, there can be many "efficiencies" for existing system installations. The systematic measurement of each possible source of inefficiency, however, outlines the probable range of efficiency of existing installations, defines the "attainable" efficiency of a good installation, and points out the modifications required to achieve the attainable efficiency.

We conducted the investigation by first measuring the base efficiency of the appliance (furnace) at steady state and full load and then measuring again at each of the possible operating conditions to quantify the resulting efficiency change. The results of these heat balance and other tests were used in two ways:

1. Modifications were made to the equipment or operating conditions and the equipment was then tested for both efficiency improvement and possible recommendation for general use.

2. The resulting data and the data from initial tests were incorporated with hourly weather data into a computer program of a model residence. Seasonal fuel utilization efficiencies were then calculated for a variety of operating conditions.

This report describes in sequence the results of the individual tests, the analytical syntheses of seasonal efficiencies, the recommendations for existing and new installations, and, finally, the detailed experimental and computational procedures.

2. SUMMARY

2.1 RESULTS AND CONCLUSIONS

The seasonal fuel utilization efficiency of a gas furnace system depends upon the design and manufacture of the *furnace*, the manner in which it is *installed*, the setting of its *adjustments and controls*, and its *age and maintenance*. Each of these was measured individually, and a range of values was determined for each element.

American Gas Association (AGA) certified-design *furnaces* measured in the laboratory and in the field had a steady-state, full-load efficiency of at least 75%. Furnaces could be designed to achieve a full-load efficiency of about 82% without suffering excessive problems from condensation of the water of combustion. Most of the installed and currently available furnaces use a pilot as the ignition device. The pilot gas consumption of about 1 cfh is not recoverable during standby. In most installations, this nonrecoverable pilot consumption amounts to 5 to 10% of the total seasonal fuel consumption. A furnace that draws its combustion and stack-dilution air from the heated space will require up to 8% more fuel in a season to heat the required makeup air.

The *installation* can contribute two major sources of inefficiency. A furnace that is oversized by a factor of 2, which is not uncommon, will require 8 to 10% more fuel than a furnace of correct size. An installation that has uninsulated ducts passing through unheated crawl or attic spaces will lose about 1.5 Btu/hr per square foot of duct per degree of temperature differential between duct air and outside air. This can amount to 40% of a furnace output under mild conditions and can easily be much greater. Ducts that leak air into or out of unheated spaces can lose almost any amount of heat. Undersized ducting will reduce the amount of circulating air and will affect the capacity of the furnace but will normally have little effect upon its efficiency.

The *adjustment* of the primary combustion air and of the firing rate has very little effect upon efficiency. The adjustment of the fan switch does affect the cyclic efficiency, and the conventional high temperature setting of this switch can reduce the seasonal efficiency by as much as 8%.

The effects of *aging* and *maintenance* (or lack of maintenance) were not significant in the three old furnaces tested. The steady-state, full-load efficiency in the "as-found" condition was comparable to that of the new furnaces in the laboratory installation. These conclusions are presented graphically in Fig. 2.1.

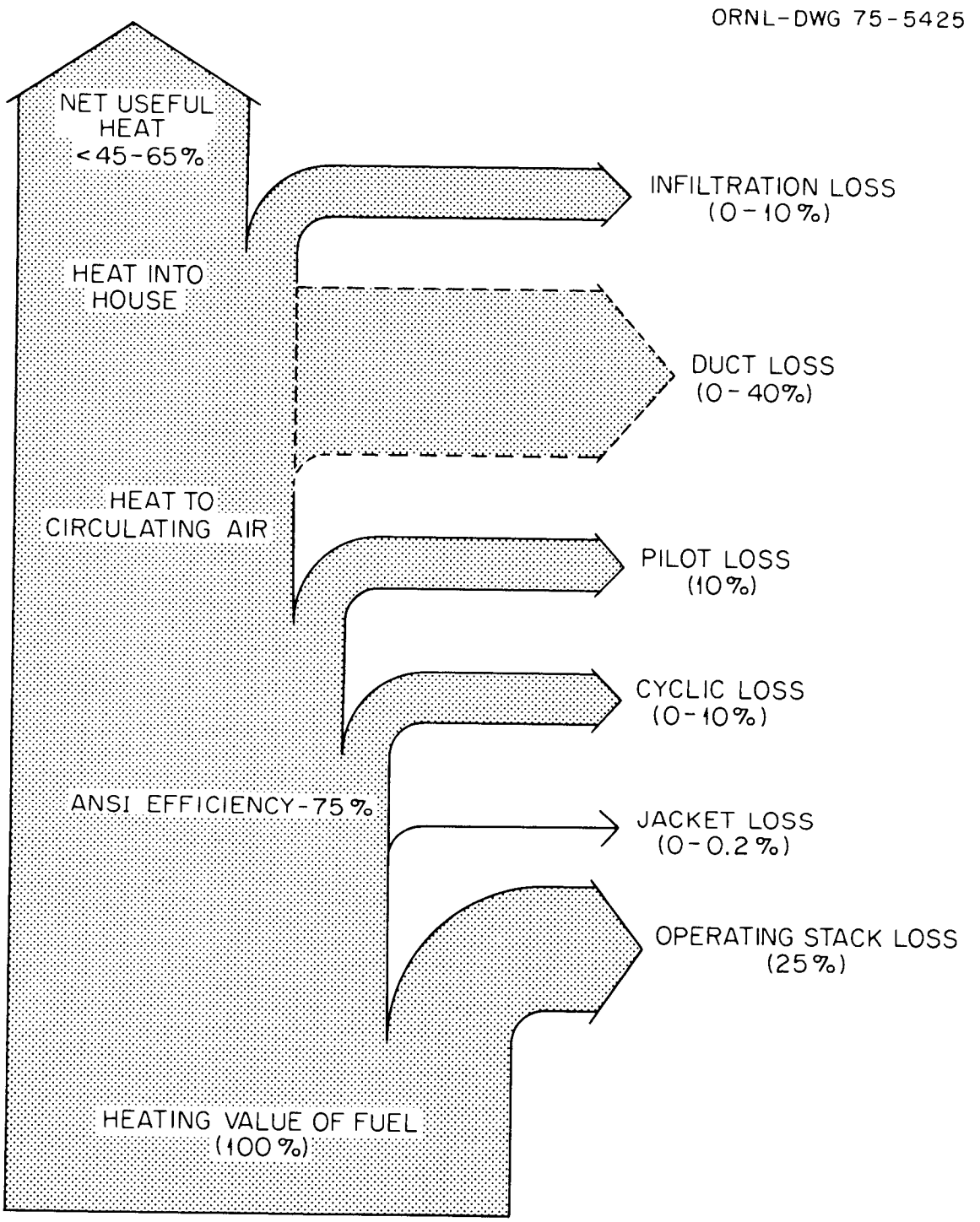

Fig. 2.1. Energy flow for a gas furnace system.

2.2 RECOMMENDATIONS

Recommendations for existing installations are as follows:

1. Perform the routine cleaning and adjustment primarily for reasons of safety and capacity.

2. Assure that all ducting (return and supply) in unheated spaces is well insulated and does not leak.

3. Set the fan switch to turn the fan off at 5 F° above the room thermostat setting and on as closely above that as the switch construction permits.

4. Turn the pilot off during nonheating seasons.

Recommendations for new furnaces and new installations are as follows:

1. All furnaces should be designed for sealed combustion and indoor installation.

2. The American National Standards Institute (ANSI) full-load, steady-state efficiency standard should be raised from 75% to 80%.

3. Electric ignition should be installed.

4. The fan switch should be factory set to turn off at 75 to 80°F.

5. There should be installation codes or standards to assure that the furnace and ducting are sized and installed to achieve the attainable efficiency.

A program should be initiated to encourage the manufacture and installation of high-efficiency furnaces that incorporate the recommended items and to establish heating system codes and standards. The high-efficiency furnace, in a proper installation, will consume two-thirds the fuel of a typical existing installation.

3. TEST RESULTS

3.1 THE FURNACE

3.1.1 Introduction

The design and fabrication standards for residential gas furnaces are defined in the American National Standards Institute (ANSI) Standard Z 21.47-1973. The preface to the twentieth edition (1973) follows:

> This publication represents a basic standard for safe operation, substantial and durable construction, and acceptable performance of gravity and forced air central furnaces. It is the result of years of experience in the manufacture, testing, installation, maintenance, inspection and research on central furnaces designed for the utilization of gas.
>
> Nothing in this standard is to be considered in any way as indicating a measure of quality beyond compliance with the provisions it contains. It is designed to allow compliance of gravity and forced air central furnaces, the construction and performance of which may exceed the various provisions specified herein. In its preparation full recognition has been given to possibilities of improvement through ingenuity of design. As progress takes place, revisions may become necessary. When they are believed desirable, recommendations should be forwarded to the Chairman of the American National Standards Committee Z21, 8501 East Pleasant Valley Road, Cleveland, Ohio 44131.
>
> Safe and satisfactory operation of a central furnace depends to a great extent upon its proper installation and it should be installed in accordance with American National Standard Installation of Gas Appliances and Gas Piping, Z21.30-1964, manufacturers' installation instructions, and local municipal building codes.

The safety record of the gas furnace attests to the efficacy of the standard.

The tests reported herein are concerned with the efficiency of furnaces and systems, and deviations in the experimental procedures from recommended safety practice were done for the sake of measurement and are not intended as recommended procedures. The final recommendations are divided into those that can be implemented with existing equipment without affecting its safety and those that may require the development and certification of new devices.

3.1.2 General

There are over 70 manufacturers and over 100 trade names of central furnaces listed in the American Gas Association (AGA) Directory. The furnaces range in size from 50,000 to 400,000 Btu/hr input capacity, and they are designed variously to be installed within the conditioned space, in the attic, in the crawl space, or outdoors. Most furnaces designed for installation within the conditioned space are atmospheric combustion units, which draw their combustion and draft hood air from the conditioned space. Figure 3.1 shows a typical atmospheric combustion furnace and illustrates that the combustion chamber and the draft hood are open to the room and permit the continuous flow of room air up the stack. This air must be replaced by outside air, which either infiltrates or enters through ducts. The air must then be heated to room temperature. A much smaller group of the furnaces designed for installation within the conditioned space are sealed combustion or direct-vent units, which draw combustion air directly from the outside. One such unit is shown in Fig. 3.2, which illustrates the isolation of the combustion zone and the stack from the room. The furnaces designed for installation outside the conditioned space — in the attic, crawl space, or outdoors — draw their combustion air from the outside. Extensive laboratory tests were made of a typical indoor, atmospheric combustion furnace and an indoor, sealed combustion furnace. Field tests were made of an indoor, atmospheric combustion furnace; an indoor, sealed combustion furnace; and an outdoor furnace.

3.1.3 Full-load, steady-state efficiency tests

The ANSI standard stipulates that a furnace must have a combustion and heat exchanger efficiency of 75% at full load and steady state. This efficiency is defined as 100% — (stack loss + jacket loss). The stack loss is determined from measurements of the CO_2 content and the temperature of the stack gas. The jacket loss is calculated from measurements of the temperature and area of the jacket.

A new furnace with a 110,000 Btu/hr input capacity was tested at full load and steady state. In addition to the stipulated measurements of CO_2 and temperature, measurements were made of the mass flow and enthalpy gain of the circulating air, the temperature and mass flow of combustion and stack dilution air, the gas consumption rate, and the electrical consumption of the blower motor. A complete heat and material balance was calculated to verify the experimental and computational procedure. In this and subsequent experiments, balance totals of 97 to 103% were considered acceptable.

The jacket loss was determined to be consistently about 0.2% and is subsequently ignored. Typical heat balance data and reductions are shown in Table 3.1. The furnaces tested consistently showed a stack loss of 23 to 25% or an ANSI efficiency of 75 to 77%. The measurement of full-load, steady-state furnace efficiency by stack CO_2 and temperature

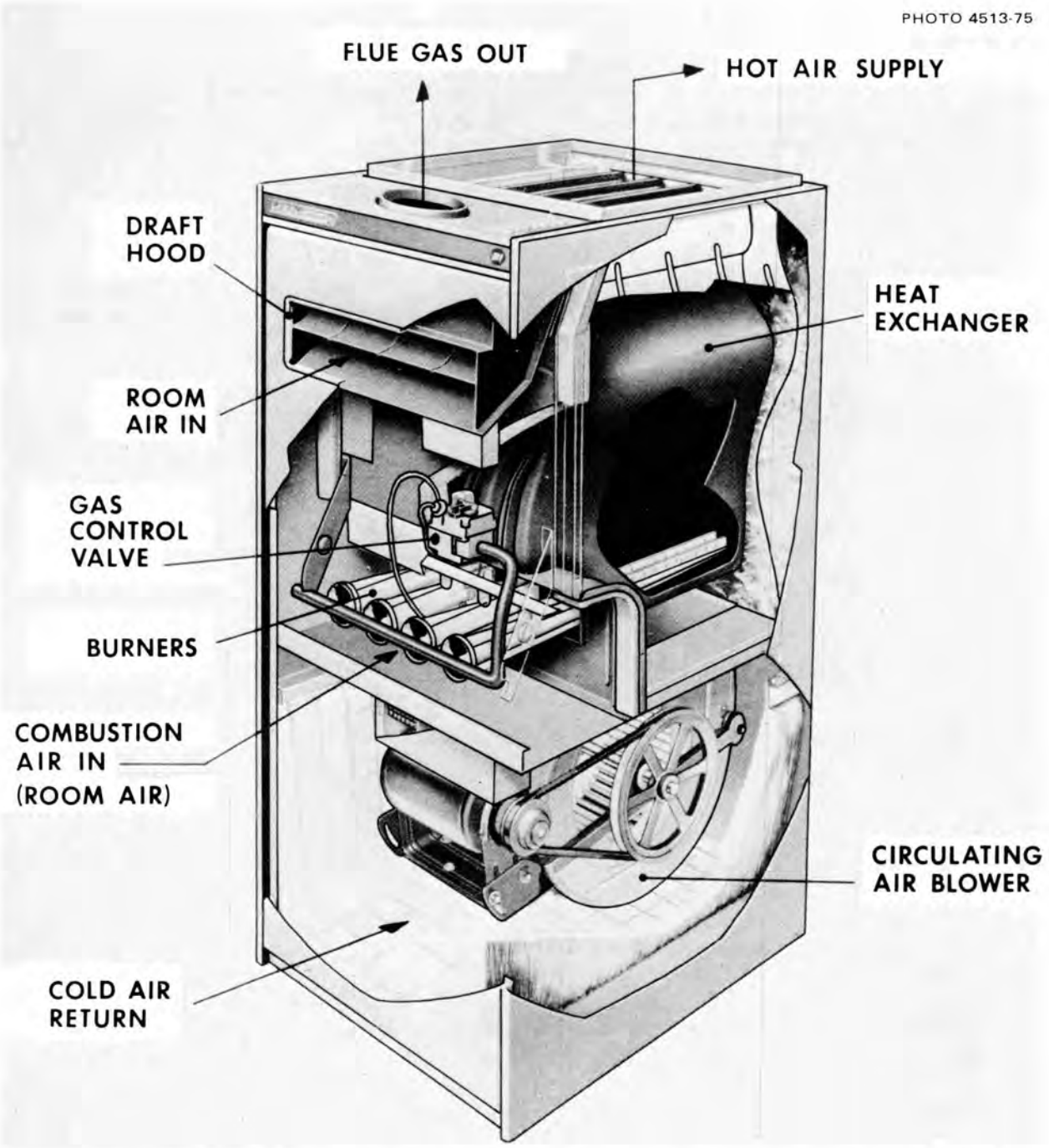

Fig. 3.1. Atmospheric combustion furnace — central, upflow, forced circulation.

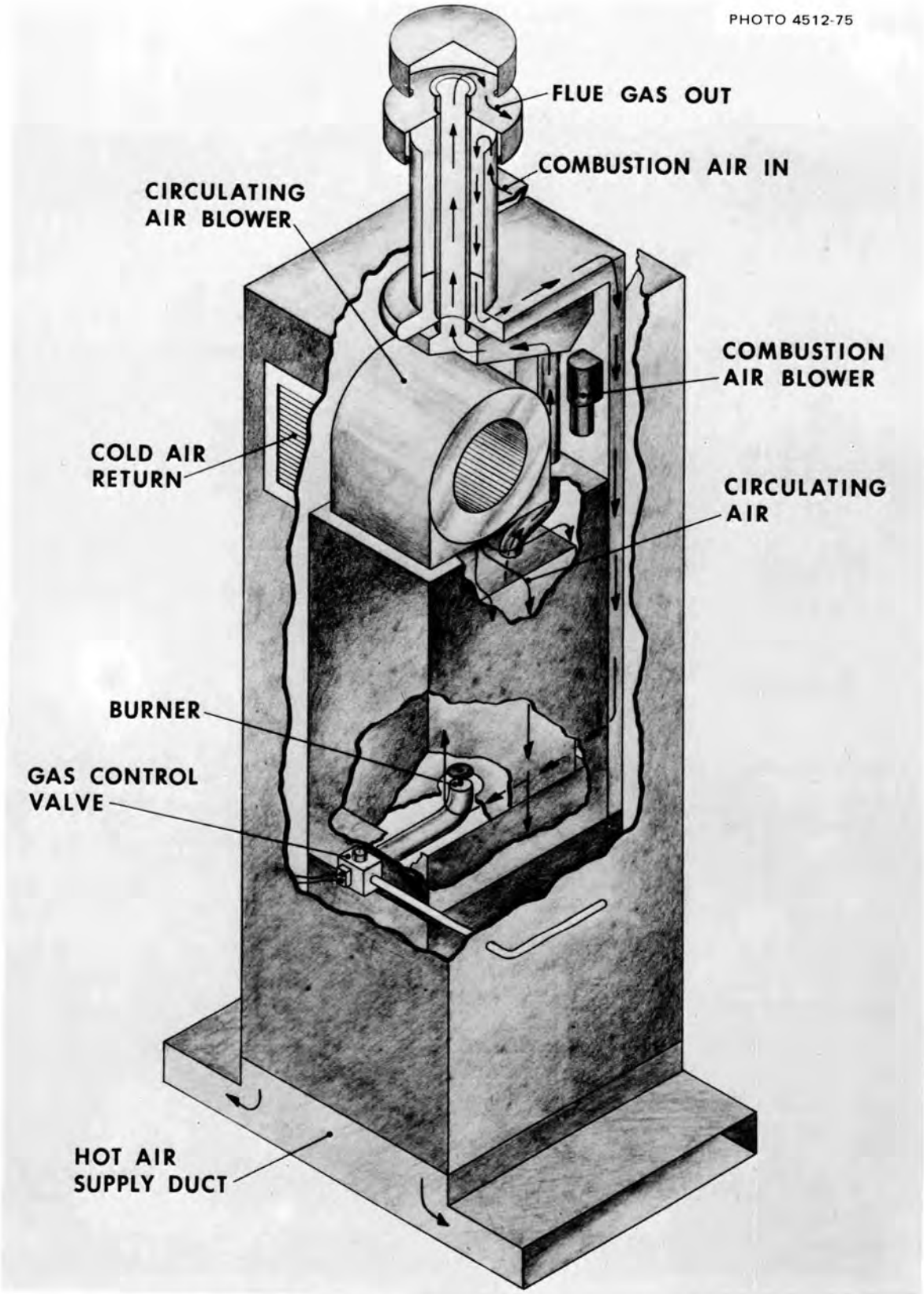

Fig. 3.2. Sealed combustion gas furnace — central, downflow, forced circulation.

Table 3.1. Gas furnace heat balance at full load, steady state

Log book page number	Q in fuel		Q in blower		Q out flue (Btu/hr)	Q out circulating air (Btu/hr)	Flue loss (%)	Jacket loss (%)	ANSI[b] efficiency (%)	Heat[a] balance efficiency (%)
	cfh	Btu/hr	Watts	Btu/hr[a]						
17	107	110,175	343	4037	25,380	85,290	23	0.2	76.8	74.7
25	106	109,200	343	4037	25,662	82,115	23.5		76.5	72.5
29	109	112,470	343	4037	28,117	83,480	25		75	71.7

[a]For seasonal fuel utilization efficiency the electrical consumption is charged at network heat rate of 11.76 Btu/Whr (29%).

[b]ANSI efficiency is 100% - (flue loss + jacket loss).

measurement was considered to be sufficiently accurate for later field determinations of full-load, steady-state efficiency.

The sum of the flue loss, the jacket loss, and the heat gain to circulating air balances well with the heat input from fuel and electrical input to the blower. The difference between the ANSI efficiency and the heat balance efficiency results from a difference in the definition of efficiency. The heat balance efficiency charges the furnace with the electrical consumption at network heat rate, whereas the ANSI efficiency does not charge blower power.

The practical limiting value for efficiency is controlled by the requirement for excess air to assure complete combustion, the requirement of a draft hood for safety in indoor atmospheric combustion installations, and the necessity to avoid condensation of the water of combustion in the heat exchanger or stack. Thus, if one assumes a draft hood installation, 150% excess air (2.5 times stoichiometric) in the flue past the draft hood, and a stack temperature of 270°F, the efficiency would be 80 to 81%. Assuming no draft hood (sealed combustion or outdoor installation), 50% excess air, and a stack temperature of 270°F, efficiency would be 84 to 85%.

3.1.4 The pilot

Gas furnaces almost universally incorporate a continuously burning pilot as a safe and convenient means of ignition. A pilot-heated thermoelectric device closes the gas valve when the pilot is not lit to prevent the release of gas when no ignition exists. Furnace pilots measured consume 1 to 1.3 cfh of gas. During operation, the pilot heat is added to the main burner and is recovered in the furnace efficiency. During standby, the pilot heat is not recovered since the circulating fan is not in operation and since the pilot combustion occurs with 50 to 100 times stoichiometric air (5,000 to 10,000% excess air).

Thus, during full-load combustion the energy in the gas is transferred to the circulating air at about 75% efficiency; however, during standby the energy of the pilot gas is completely lost. To evaluate the magnitude of this loss, it is necessary to determine the number of hours of standby operation during a heating season. As a part of the computer analysis discussed in Sect. 4, the operating time during a heating season was determined for a given set of conditions. Table 3.2 displays the results of a set of calculations based on eight years of hourly weather tapes for an 1800 ft^2 house insulated to current FHA standards, located in Philadelphia, and equipped with a 110,000-Btu/hr input furnace. The peak hourly heating requirement was calculated to be 40,000 Btu/hr, which would require a 54,000-Btu/hr input furnace operating at 75% efficiency. The 110,000-Btu/hr input furnace is oversized and on its "worst day" operates only half-time; however, the typical practice is to install at least this large a furnace under these conditions.

Table 3.2. Pilot consumption

Heating season only; 110,000 Btu/hr input furnace; 75% full-load efficiency; 1800 ft^2 model house; Philadelphia; average of eight years weather tapes; inside heated to 68°F

Month	Monthly heating load (cubic feet of gas)	Monthly furnace input at 75% efficiency to supply load (cubic feet of gas)	Hours of furnace operation	Monthly pilot standby consumption (cubic feet of gas)
September	80	107	1	719
October	1,113	1,484	13	731
November	4,464	5,952	51	669
December	9,419	12,558	108	636
January	10,466	13,954	120	624
February	7,713	10,284	89	583
March	5,888	7,850	68	676
April	2,009	2,678	23	697
May	326	434	4	740
Seasonal total		55,301		6,075
Plus standby pilot		6,075		
		61,376		

Pilot loss as percent of total gas consunption = 6,075/61,376 = 9.9%

An energy loss of about 10% is significant and raises the following questions: (1) Is there an available solution? (2) What are the economics? There are eight to ten electric ignition and safety units for gas-fired equipment in production and commercial use. They add about $30.00 to the retail cost as compared to the same appliance equipped with a gas pilot and safety unit. At current gas prices of $1.50 to $4.50/1000 ft^3, the cost of the 6075 ft^3/year consumption of the standby pilot shown in Table 3.2 is $9.00 to $27.00 per year.

3.1.5 Source of combustion and draft hood air

The general types of central furnaces were described in Sect. 3.1.2. The purpose of this test series was to determine the effect of the combustion air source on the system efficiency. Analyses of the measurements reflect the opinion that adequate ventilation of a residence is a valid heat load equally imposed on any heating system but that the infiltration load necessitated by the furnace design should be charged against furnace efficiency.

Atmospheric combustion units. There are two distinct rates of stack air flow, one during combustion and another during standby, both of which can be fairly well defined.

During combustion, the percentage CO_2 in the stack downstream of the draft hood depends somewhat on stack configuration and indoor-outdoor temperature differential but is normally from 3.5 to 4.5%. This means that the total air supplied to the stack is 3.5 to 2.5 times stoichiometric or 34 to 24 ft^3 of air per cubic foot of gas. Thus a 110,000 Btu/hr furnace pumps air out of the house at the rate of 2500 to 3500 cfh during combustion. Typical measurements and calculations are shown in the Appendix.

During standby the combustion chamber and draft hood remain open, and conditioned air continues to flow up the stack. Figure 3.3 shows typical curves for standby stack air flow in the laboratory test furnace, starting at the combustion rate and decaying to an asymptotic value. Figure 3.4 shows the decay curve for one residential installation. The asymptotic or final value of standby air flow is a function of the indoor-outdoor temperature differential and the stack configuration. The decay rate is a function of the heat exchanger cooldown rate which, in turn, depends upon the fan switch adjustment.

For any set of fixed conditions the efficiency loss due to infiltration can be calculated, and if these calculations are integrated over the entire heating season, the average seasonal efficiency loss can be determined. The infiltration loss calculation is based on the size of the furnace, the duty cycle, the fan switch adjustment, and the indoor-outdoor temperature differential. Table 3.3 shows the results of such calculations using different furnace capacities and hourly weather tapes for several locations.

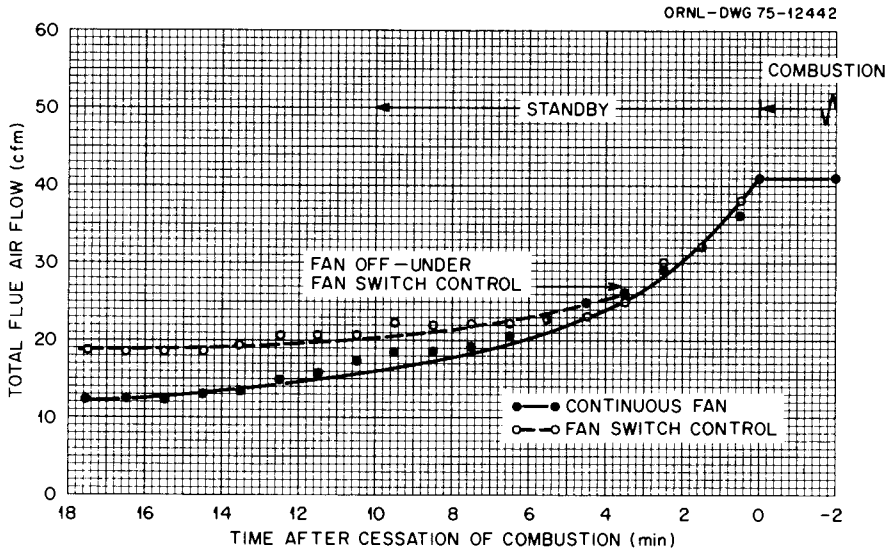

Fig. 3.3. Flue air flow in laboratory test. (Reproduced from chart graph.)

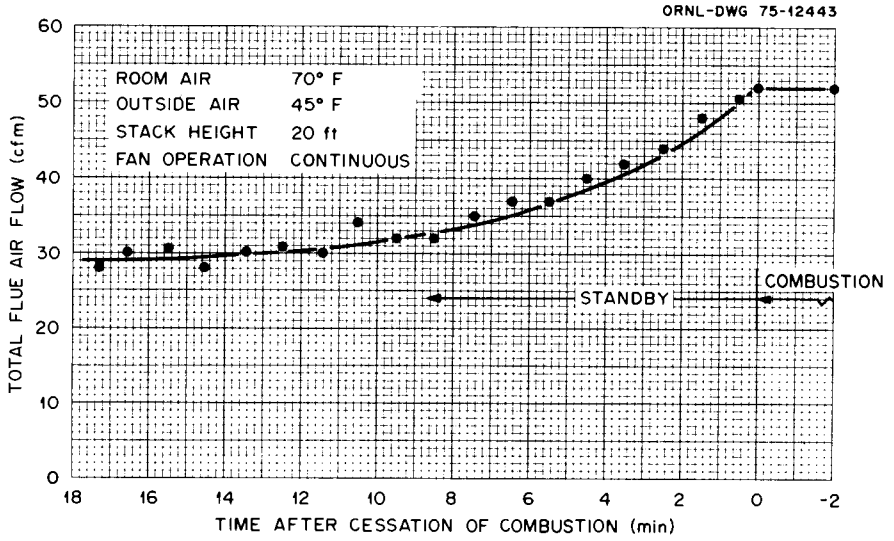

Fig. 3.4. Flue air flow in residential test. (Reproduced from chart graph.)

Table 3.3. Effect of furnace-induced infiltration on the seasonal efficiency of an atmospheric combustion furnace

Location	Furnace size[a]	Seasonal fuel utilization efficiency (%)		Infiltration loss (%)
		Source of combustion air[b]		
		Outside	Inside	
San Diego	C	52.8	51.4	1.4
	2×	48	46.3	1.7
Atlanta	C	61.3	58.3	3
	2×	55.2	50.3	4.9
Philadelphia	C	63.8	60.3	3.5
	2×	57.5	51.7	5.8
Minneapolis	C	66.4	61.4	5
	2×	59.6	51.6	8

[a]C = correct; 2× = twice oversize.

[b]"Inside" assumes an atmospheric combustion furnace that draws its air from the heated space. "Outside" assumes the same furnace that draws its air from outside the heated space by means not defined.

The infiltration loss varies with climate and with furnace size. However, it will be shown in the following section on sealed combustion units that the magnitude of the loss is greater than that shown in Table 3.3 because of an effect that cannot be distinguished in analyzing the conventional furnace.

Sealed combustion units. The sealed combustion unit shown in Fig. 3.2 was designed for mobile home installation where the roof is in an invariable relation to the furnace. The furnaces of this type, up to about 70,000 Btu/hr input, are naturally aspirated; that is, natural convection and gas jet pumping drive the combustion air and flue gasses through the relatively short stack. The 100,000-Btu/hr input model has a combustion air booster blower and is, therefore, not restricted to mobile home geometry. The booster blower has a centrifugal switch to permit opening the gas valve only when combustion air is assured. A new 100,000-Btu/hr sealed combustion furnace was tested in the laboratory.

The full-load, steady-state heat balances again gave an efficiency of about 75%. The flue gas CO_2 concentration was 7.5 to 8.0%, and the temperature was 560 to 570°F. The higher concentration and temperature are attributable to the lack of draft hood dilution.

The furnace has a standing pilot, thus incurring the approximate 10% seasonal loss resulting from the consumption of fuel without delivery of heat to the residence.

However, the sealed combustion unit has two salutary effects, one obvious and one somewhat surprising. Obviously, sealed combustion eliminates the infiltration loss since no air from the conditioned space is drawn into the flue. Surprisingly, the cyclic efficiency can be higher than steady state. The effect of cyclic operation on the efficiency of the sealed combustion furnace and of the atmospheric combustion furnace is discussed in detail in Sect. 3.3.4. The seasonal efficiency for a sealed combustion furnace, calculated for the same conditions as for the atmospheric combustion furnace, is shown in Table 3.4.

Outdoor furnaces. One outdoor furnace installed at a residence was field tested for full-load, steady-state efficiency and was determined to be 75% efficient. The tests for pilot, infiltration, and cyclic losses were not performed. The pilot loss would, of course, be the same as for any furnace with a standing pilot, and there is no infiltration loss since the stack in no way communicates with the conditioned space. The cyclic loss (Sect. 3.3.4) would be expected to be similar to, but greater than, that of an indoor, atmospheric combustion furnace since the combustion chamber can be cooled below room temperature by the convection flow of outdoor air. Thus the seasonal efficiency is about the same as that of the indoor, atmospheric combustion furnace.

Table 3.4. Seasonal efficiency of a sealed-combustion furnace

Location	Furnace size[a]	Seasonal fuel utilization efficiency (%)
San Diego	C	61.1
	2×	58.4
Atlanta	C	71.2
	2×	66.5
Philadelphia	C	73.4
	2×	69.6
Minneapolis	C	75.5
	2×	72

[a]C = correct; 2× = twice oversize.

3.2 THE INSTALLATION

Three factors in an installation can reduce the seasonal operating efficiency below the full-load, steady-state efficiency of the furnace: (1) the installed capacity, (2) circulating air flow, and (3) duct losses.

3.2.1 Installed capacity

The size of a residential furnace has been considered critical only to assure adequate capacity for the coldest day and recovery from setback. The cost of a furnace installation has been small, and the cost differential between several capacities has been even smaller. Thus, it has not been customary to perform a careful heating load calculation. However, the size of the furnace can affect the system in two quantifiable ways: the frequency and duration of cycling and the stack-induced infiltration air flow rate. The nature and magnitude of the cyclic losses are discussed in the section on adjustments and controls (Sect. 3.3). The nature and magnitude of the stack air losses are discussed in the following paragraph on the source of combustion and stack dilution air.

Both of these losses are incorporated into the analytical synthesis and are more fully discussed in Sect. 4. Table 3.5 is a resume of four cases showing several correct and oversize installations.

Table 3.5. Effect of furnace size on efficiency

Location	Furnace size[a]	Seasonal fuel utilization efficiency (%)[b]
San Diego	C	51.4
	2×	46.3
Atlanta	C	58.3
	2×	50.3
Philadelphia	C	60.3
	2×	51.7
Minneapolis	C	61.4
	2×	51.6

[a]C = correct; 2× = twice oversize.

[b]Atmospheric combustion with air supplied from the conditioned space.

3.2.2 Circulating air flow

An inadequate duct system can reduce the circulating air flow below that specified by the manufacturer, thus raising stack temperature and lowering the efficiency. A series of tests was performed to measure the response of furnace efficiency to air flow variations from 104 to 66% of the manufacturer's rating. It was necessary at lowered flow rates to disable the limit switch to obtain continuous operation. The test conditions and resulting efficiencies are shown in Table 3.6. The efficiency loss is small, even at 66% rated flow. However, the limit switch would normally shut off the fire and prevent the temperature rise found in the lowered flow tests. Thus the normal effect of reduced circulating air flow is not a significant direct reduction of efficiency but rather a reduction of capacity through a cycling of the fire.

3.2.3 Duct losses

The forced air central furnace distributes its heat to the residence through a supply and return duct system. This duct system can easily be the major source of efficiency loss through both heat and mass transfer by at least three mechanisms. Uninsulated supply and return ducts passing through an unheated crawl or attic space will conduct and radiate heat directly to that space. Leaks at joints will lose heated air out of the supply duct and will draw unheated air into the return duct. Vertical duct chases can cause ducts to be placed adjacent to the sheathing of exterior walls and can act as chimneys to carry heated air directly to the attic space.

Table 3.6. Efficiency at different circulating air flows

Log book page number	Circulating air flow			Flue gas		ANSI efficiency (%)	Heat balance efficiency (%)
	Percent rated	Cubic feet per minute	Temperature rise (°F)	Percent CO_2	Temperature rise (°F)		
32	104	1061	81	4.25	276	76.5	73.4
33	100	922	84	4.5	282	76.5	73.5
34	96	886	89	4.5	286	76.5	75
35	88	809	96	4.5	291	76	73.8
36	66	611	128	4.5	306	75.5	74.6

The duct leakage and chase losses can have a wide range of magnitude and would have to be calculated on an individual basis; however, an illustrative example will indicate the magnitude of the convection and radiation loss from an uninsulated duct. The overall heat-transfer coefficient of a bare sheet metal duct was experimentally measured to be about 1.5 Btu/hr-ft^2-°F, with the duct air at 160°F and the ambient air at 50°F. Figures 3.5 and 3.6, extracted from ANSI Z21.47-1973, show values for h_c and h_{R1}. Galvanized sheet has an emissivity of 0.23 to 0.28. Figure 3.5 shows an average convection coefficient of about 1 Btu/hr-ft^2-°F, and Fig. 3.6 shows a radiation coefficient of 0.3 Btu/hr-ft^2-°F for an average overall coefficient of 1.3 Btu/hr-ft^2-°F for the temperature conditions stated. Using an average overall coefficient of 1.4 Btu/hr-ft^2-°F and the assumed duct dimensions and temperatures shown, one can calculate the duct loss as follows:

Assume

supply air temperature, 160°F;
crawl space temperature, 50°F;
main duct, one 20 ft long x 12 in. x 18 in.;
branch ducts, six 10 ft long x 8 in. diam;

then

loss per square foot of duct surface = 1.4 Btu/hr-ft^2-°F x 110°F = 154 Btu/hr-ft^2;
duct area = 100 ft^2 (main) + 125 ft^2 (branch) = 225 ft^2;
total loss = 154 Btu/hr-ft^2 x 225 ft^2 = 34,650 Btu/hr;
output of 110,000-Btu/hr furnace at 75% efficiency = 82,500 Btu/hr; percent loss to uninsulated duct = 42%.

The above calculation is highly simplified for the sake of illustration. The more rigorous calculation using mean duct temperature yields the correct solution — 29,550 Btu/hr loss (36%).

The overall heat-transfer coefficient of the same duct insulated with 1 in. of fiberglass was experimentally measured to be 0.5 Btu/hr-ft^2-°F. At this coefficient the loss from the above assumed duct system would be 11,660 Btu/hr or 14% of the furnace output, which is still too great a loss. It is thus quite evident that an inadequate duct installation can constitute the major heat loss in a heating system.

Mobile home ducting is a special case. The supply duct is usually placed in the frame prior to laying the floor. After the floor is laid, the floor registers are installed through the floor and through cutouts in the duct. The light-gage sheet metal and the assembly method make it difficult to obtain tight joints. The two extreme ends of the duct are frequently closed by folding the end over in a manner similar to closing a paper bag.

The American National Standards Institute standard relating to supply system ducting in mobile homes is reproduced in Fig. 3.7.

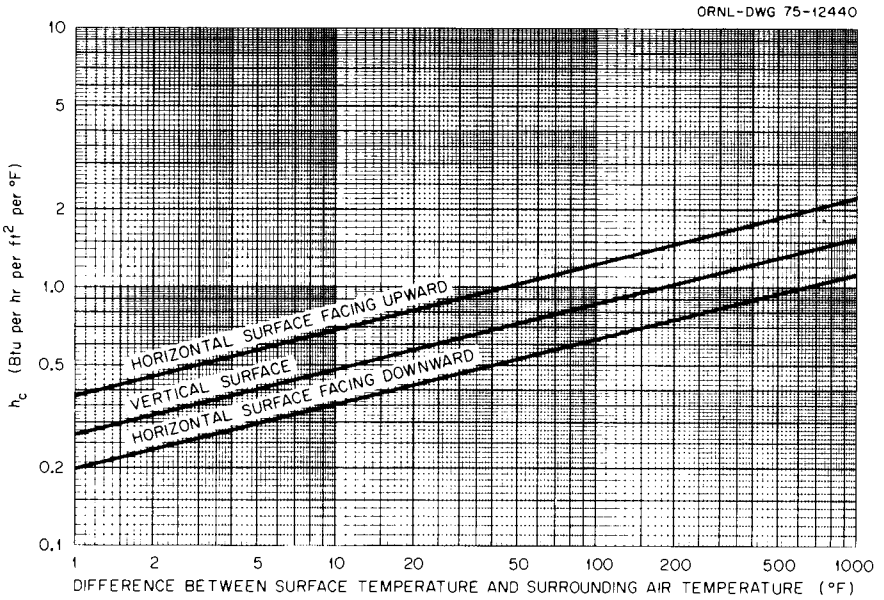

Fig. 3.5. Convection coefficients for vertical and horizontal surfaces. Source: American National Standards Institute, Inc., *Gas-Fired, Gravity, and Forced Air Central Furnaces*, ANSI Z21.47-1973, 1973, Exhibit 1, p. 87.

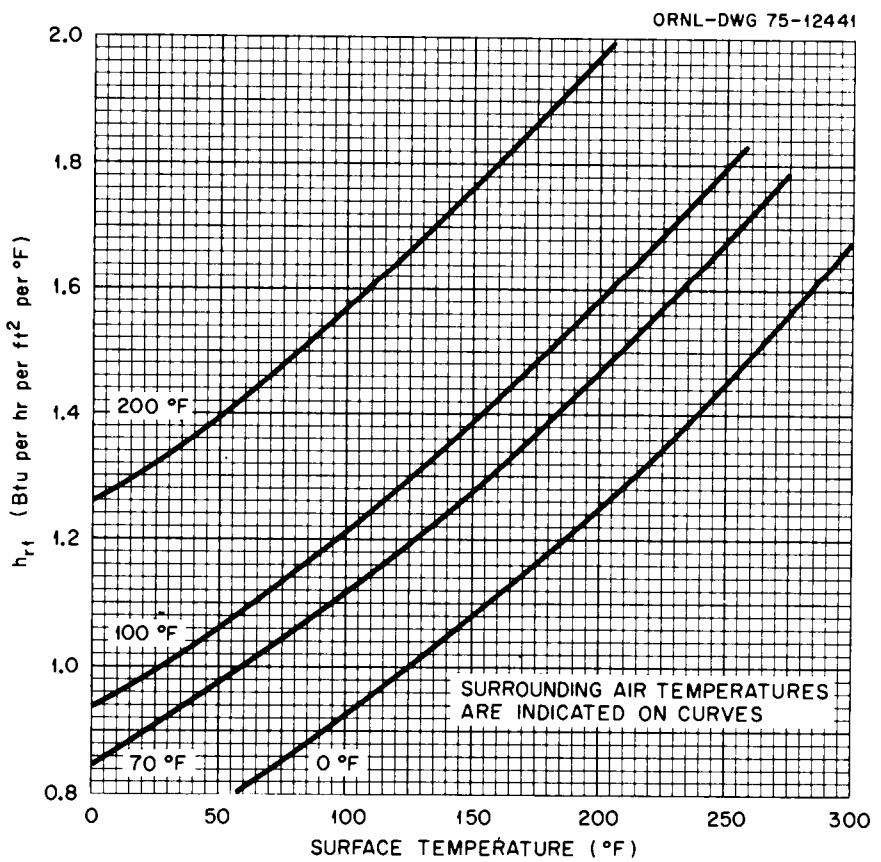

Fig. 3.6. Coefficient of heat transfer (by radiation) for emissivity of 1.0. Source: American National Standards Institute, Inc., *Gas-Fired, Gravity and Forced Air Central Sources*, ANSI Z21.47-1973, 1973, Exhibit 2, p. 87.

6.10 Circulating Air System

6.10.1 Supply System

6.10.1.1 Duct Material. Supply ducts shall be made from galvanized stell, tin-plated steel, or aluminum, or shall be listed Class 0, Class 1 or Class 2 air ducts. Class 2 air ducts shall be located at least 3 feet from the furnace bonnet or plenum. A duct system integral with the structure shall be of durable construction that can be demonstrated to be equally resistant to fire and deterioration. Ducts constructed from sheet metal shall be in accordance with Table D-3.

TABLE D-3

Minimum Metal Thickness for Ducts*

Duct Type	Diameter 14 inches or less	or	Width over 14 inches
Round	0.013 in.		0.016 in.
Enclosed Rectangular	0.013 in.		0.016 in.
Exposed Rectangular	0.016 in.		0.019 in.

*When "nominal" thicknesses are specified, 0.003 inch shall be added to these "minimum" metal thicknesses.

TABLE D-4

Maximum Allowable Static Pressures in Supply Duct Systems

Input to Forced-Air Furnace, Btu/hr	External Static Pressure Inches Water Column Measured at the Furnace Outlet — Temperature of Outlet Air Determined by Function of Limit Control	
	Above 165°F	165°F or Less
55,000 and under	0.10	0.20
Over 55,000 to 80,000	0.12	0.24
Over 80,000 to 100,000	0.15	0.30

6.10.1.2 Sizing of Ducts. Ducts shall be designed so that when a labeled forced-air furnace is installed and operated continually at its normal input rating in the mobile home, with all registers in full open position, the static pressure measured in the duct plenum shall not exceed that shown in Table D-4, or exceed that shown on the label of the appliance. When an air-cooler coil is installed between the furnace and the duct plenum, the total static pressure between the furnace and the coil shall not exceed that shown on the label of the furnace. The minimum dimension of any branch duct shall be at least 1½ inches, and of any main duct, 2½ inches.

6.10.1.3 Airtightness of Supply Duct Systems. A supply duct system shall be considered substantially airtight when the static pressure in the duct system, with all registers sealed and with the furnace air circulator at high speed, is at least 80 percent of the static pressure measured in the furnace casing, with its outlets sealed and the furnace air circulator operating at high speed. For the purpose of this paragraph and 6.10.2, pressures shall be measured with a water manometer or equivalent device calibrated to read in increments not greater than 1/10 inch water column.

6.10.2 Return Air Systems

6.10.2.1 Return Air Openings. Provisions shall be made to permit the return of circulating air from all rooms and living spaces, except toilet room(s), to the circulating air supply inlet of the furnace.

6.10.2.2 Duct Material. Return ducts, if provided, shall be in accordance with the following:

(a). Portions of return ducts directly above the heating surfaces, or closer than 2 feet from the outer jacket or casing of the furnace shall be constructed of metal in accordance with Table D-3 or shall be listed Class 0 or Class 1 air ducts.

(b) Return ducts, except as required by (a) above, shall be constructed of one-inch (nominal) wood boards (flame spread classification of not more than 200), other suitable material no more flammable than one-inch board or in accordance with 6.10.1.1.

(c) The interior of combustible ducts shall be lined with noncombustible material at points where there might be danger from incandescent particles dropped through the register or furnace such as directly under floor registers and the bottom of vertical ducts or directly under furnaces having a bottom return.

Fig. 3.7. ANSI standard for supply system ducting. Source: American National Standards Institute, Inc., *Standard for Mobile Homes*, A119.1, Sect. 6.10.1, January 17, 1974.

Actual flow vs static head measurements were made on the furnace and duct system of one mobile home. The 70,000-Btu/hr input furnace is labeled to have a maximum allowable static pressure of 0.6 in. H_2O. The duct system is designed in accordance with the appliance label provision of paragraph 6.10.1.2 shown in Fig. 3.7, rather than with Table D.4 of Fig. 3.7. The flow-head characteristics of the blower shown in Fig. 3.8 were obtained by measuring the air flow and the static pressure with the supply register progressively sealed and, finally, with the base of the furnace sealed.

The static pressure in the duct system with all the registers sealed (1.17 in. H_2O) is 78% of the static pressure at shutoff (1.5 in. H_2O) which approaches the minimum airtightness specifications of the quoted ANSI standard. The leakage flow (Fig. 3.8) at the allowable 80% pressure (1.2 in. H_2O) is about 330 cfm, which at the operating pressure of 0.6 in. H_2O would be a leakage flow of about 233 cfm. This allowable leakage flow amounts to 25% of the measured furnace output of 930 cfm. Some of this leakage flow may appear in the mobile home; however, it is an uncontrolled and undesirable situation.

From Table 3.4 it can be estimated that a correctly sized, sealed combustion furnace would have a seasonal efficiency of about 73% delivered to the hot air plenum. However, the duct leakage calculated above is a direct loss of 25% of that output or about 15% of the fuel input. Thus the system efficiency is reduced to about 55%. Any heat loss due to inadequate duct insulation would, of course, further reduce the seasonal efficiency.

3.3 ADJUSTMENTS AND CONTROLS

There are normally four adjustments or controls, in addition to the safety devices, on residential gas furnaces — the room thermostat, the gas manifold pressure, the primary combustion air, and the fan switch.

3.3.1 The room thermostat

The absolute setting of the room thermostat is under the control of the resident and affects fuel consumption but not system efficiency. The differential — the temperature difference between turning the furnace on and turning it off — may or may not be adjustable. A narrow differential will maintain a more even temperature and will give the sensation of comfort at a lower average temperature. It will also cause more frequent cycles of short duration.

The more subtle aspects of the relationship of heating system performance to the sensation of comfort and the subsequent effect of the comfort sensation on system efficiency are beyond the scope of this investigation. However, some measurements were made to illustrate several types of system response. Figure 3.9 a and b illustrates temperature recordings made in two significantly different residences having forced-air gas

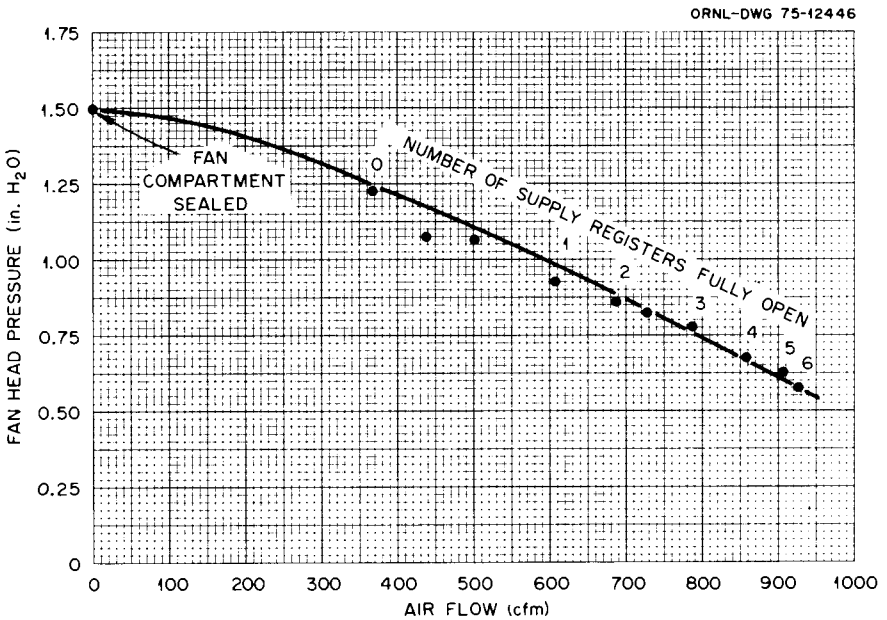

Fig. 3.8. Head-capacity curve for a mobile home furnace fan.

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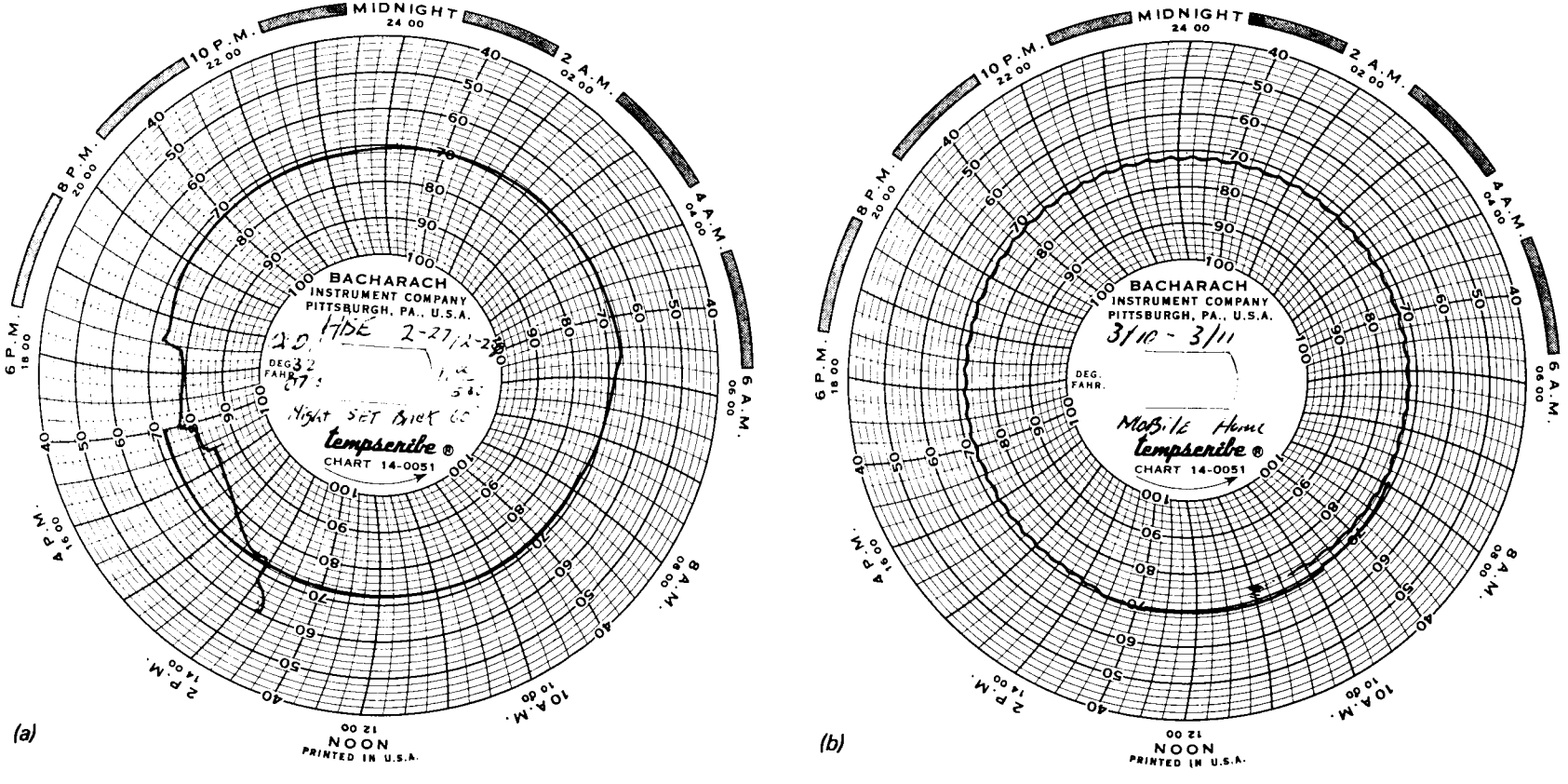

Fig. 3.9. Twenty-four-hour indoor temperature recordings for (*a*) a conventional residence and (*b*) a 50-ft mobile home. (Reproduced from chart graph.)

furnaces. In each case the instrument was placed at waist height out of any obvious drafts. The first recording (Fig. 3.9a) was made in a conventional, 1400 ft^2 residence with a heated basement. The fan switch was set to turn the fan off at a bonnet temperature of 70°F. The valid trace is from 7 PM, February 27, to 5 PM, February 28, 1975. The minimum outdoor temperature was about 32°F. The thermostat incorporates an automatic setback to 60°F at 11 PM and setup to 72°F at 5:30 AM. The room temperature shows virtually no temperature cycling.

The second recording (Fig. 3.9b) was made March 10 and 11 in a 50-ft mobile home. The thermostat was set at 72°F, and the fan was set to operate continuously. Here the recording shows a cycling of about 2° in amplitude during a 20- to 25-min period. The period is, no doubt, related to the thermal inertia of the structure; however, the amplitude should be almost entirely related to thermostat differential.

3.3.2 Burner manifold pressure

The jets and burners are designed to operate at a specific gas manifold pressure which is maintained by the regulator in the gas control valve, if the gas supply system remains within a specified pressure range. The specified manifold pressure should be maintained both for obtaining rated capacity and for safety reasons, such as reliable ignition and freedom from flashback.

However, for the purpose of testing the furnace efficiency at reduced firing rates, as might result from a maladjusted regulator or a drop in supply pressure, the manifold pressure was reduced by partly closing the gas shutoff valve. The measured efficiency as a function of firing rate is shown in Table 3.7. There appears to be a slight loss of efficiency at a 50% firing rate, but there is certainly no appreciable change of efficiency over any normally occurring reductions in the firing rate.

One might anticipate that reducing the firing rate while maintaining the correct fuel-to-air ratio would yield some improvement in efficiency through reduced loading of the heat exchanger. A brief test of this thesis was made by replacing the gas jets to obtain a 50% firing rate and then reducing the primary and secondary combustion air until the CO_2 content of the gases exiting from the combustion chamber was approximately normal. The draft hood dilution rate was so high that the stack CO_2 measurements were not reliable; however, the efficiency calculations based on the heat gain in the circulating air and on the analyses of the gases exiting from the combustion chamber are in agreement. As shown in Table 3.8, the furnace efficiency was lowered by simply rejetting, but it rose to normal at a normal air-to-fuel ratio. This relative simple modification resulted in no improvement in efficiency.

Table 3.7. Effect on efficiency of reducing firing rate by reducing gas manifold pressure

Percent of rated input	Percent efficiency calculated from circulating air heat balance	Percent efficiency calculated from stack	Log book page number
100[a]	74.7	77	17
81[a]	74.7	75	16
53[a]	74.7	76	18
50[a]	71.7	76	95
50[b]		72	149
100[b]		74	149

[a]Laboratory test of new furnace.

[b]Field test of furnace installed in residence.

Table 3.8. Effect on efficiency of reducing firing to one-half the manufacturer's rating (using smaller jets)

Combustion air condition	Stack		Combustion chamber exit		Percent efficiency by circulating air	Percent efficiency by stack	Percent efficiency combustion chamber	Log book page number
	Percent CO_2	Temperature	Percent CO_2	Temperature				
Normal	1.5	239	Not measured		68	69		65
Reduced to 0.307 ft^2 entry	2.25	244	4.25	495	70.5	74.5	69	66
Almost blocked	1.25	185	6	440	75.6	71.5	76	73

3.3.3 Primary air

Most burners have a provision for adjusting the amount of primary combustion air (air that is induced by the gas jet into the burner head). The adjustment is normally made to obtain the correct flame appearance, which should, in turn, result in complete combustion. To determine the sensitivity of efficiency to primary air adjustment, the shutters on the experimental furnace were adjusted from fully open to closed. Table 3.9 shows the results of the test. The primary air should be correctly adjusted for safe operation; however, it is obvious that the furnace efficiency is not very sensitive to a small maladjustment.

3.3.4 Limit and fan control

Furnaces have a bimetal thermostat element that senses bonnet air temperature and serves both to limit the maximum bonnet temperature and to control the circulating air fan or blower. The fan control has an adjustable set temperature for turning the fan on and another for turning it off. The practice varies but seems to be set "fan on" at 125 to 140°F and "fan off" at about 100°F.

The normal sequence of operation of the furnace control system is:

1. The room temperature drops below a set point on the thermostat and the gas is fired.

2. The bonnet air temperature rises to the "fan-on" setting, and the circulating air blower is activated.

3. The room temperature rises to a set point on the thermostat, and the fire is turned off.

4. The bonnet air temperature drops to the "fan-off" setting, which turns off the blower.

The heat stored in the heat exchanger after the blower is off is not distributed to the heated space but instead increases the driving force to send room air up the stack in the atmospheric combustion unit or to circulate outside air through the combustion chamber in sealed combustion or outdoor units. Thus, during each cycle the stored heat below the "fan-off" setting is not only lost, but in the indoor, atmospheric combustion furnace, it increases the infiltration loss. Therefore, the cyclic efficiency is different from the steady-state efficiency, and, further, the cyclic efficiency is affected by the setting of the fan switch and the cycle duration.

Tests were conducted to determine the efficiency at different loads or duty cycles with different fan switch settings. A 30-min cycle was assumed in all cases, and tests were performed at 10, 20, 40, 60, and 80% loads. The fan switch was set to turn on at 140°F and off at 100°F for one series and was set to run continuously for one series. Complete

heat and material balances were taken in each case. A pen recorder plotted the temperature curves for both circulating air and stack air, and the areas under the curves were integrated to determine quantitatively the division of heat into the circulating air and the stack.

Since their response is quite different, the atmospheric combustion and the sealed combustion furnaces were tested separately.

Atmospheric combustion furnace. Figures 3.10 and 3.11 are for a 20% duty cycle (6 min on and 24 min off); they illustrate the difference between the high (on at 140°F and off at 100°F) and low (on and off at 5 F° above room temperature) fan switch settings. At the high fan switch setting (Fig. 3.10), the fire was on for about 45 sec before the circulating blower turned on at point "C". The flue temperature rose more steeply than in Fig. 3.11, and there is, therefore, more area (greater loss) under the beginning of the curve in Fig. 3.10. Also, in Fig. 3.10, after the fire went off (point B), the blower pulled the bonnet temperature down to about 100°F, turned off (point D), and remained off for about 75 sec until the bonnet temperature rose from the stored heat and turned the blower on (point E). During this period the flue temperature curve of Fig. 3.10 departs sharply from that of Fig. 3.11. The blower then ran for 15 sec (from E to F), turned off, and remained off. The flue temperature in Fig. 3.10 and stack air flow rate of Fig. 3.3(B) remained much higher than in Figs. 3.11 and 3.3(A). The efficiencies cited in Fig. 3.11 were determined by both graphically and analytically integrating the temperature and flow curves for the different cyclic operations. Using the efficiencies shown in Fig. 3.12, the seasonal efficiency of an indoor, atmospheric combustion furnace was calculated for the "high" and "low" fan switch settings for various climates and conditions. The results are shown in Table 3.10.

Sealed combustion furnace. The cyclic efficiency response of the sealed combustion furnace is quite different from the atmospheric combustion furnace. Figures 3.13 and 3.14 illustrate the reason. Figure 3.13 is a trace of the flue gas, return air, and supply air temperatures for a 20% duty cycle with the switch set on "high," and Fig. 3.14 is a 20% duty cycle with the fan switch set on "low." For the sake of comparison, the flue and supply air temperatures for steady-state, full-load operation are superimposed in dashed lines in both figures. It is apparent, on inspection, that the average flue-gas temperature, in short-cycle operation, is below that of steady-state operation and that the stack loss is thus reduced. It is also apparent that the useful heat delivered — the area under the supply air temperature curve — is greater for the "low" fan switch setting (Fig. 3.14) than for the "high" fan switch setting (Fig. 3.13). Quantitative answers were obtained by integrating the areas in the same fashion as for the atmospheric combustion furnace. Compared with the atmospheric combustion furnace, however, the efficiency of the sealed combustion furnace is higher for short cycles than for steady-state operation because, during the fire-off period, there is no loss of room air and a low rate of loss up the stack of the heat stored in the heat exchanger. The cyclic efficiency at both the "high" and "low"

Table 3.9. Effect on efficiency of variations in primary air adjustment

Log book page number	Primary air shutter setting	Percent CO_2 in stack	Percent efficiency calculated from heat balance	Percent efficiency calculated from stack loss
141	Closed	4	72	74.5
96	Correct	4	72	75
97	Open	4	75.6	75

Table 3.10. Seasonal fuel efficiency of an atmospheric combustion furnace

		Seasonal fuel utilization efficiency (%)	
		Fan switch setting[b]	
Location	Furnace size[a]	High	Low
San Diego	C	51.4	56.0
	2×	46.3	53.2
Atlanta	C	58.3	65.8
	2×	50.3	60.2
Philadelphia	C	60.3	67.2
	2×	51.7	61.4
Minneapolis	C	61.4	68.0
	2×	51.6	61.0

[a]C = correct; 2× = twice oversize.

[b]Fan switch setting: high — blower on at 140°F and off at 100°F; low — blower on and off at 5 F° above room temperature.

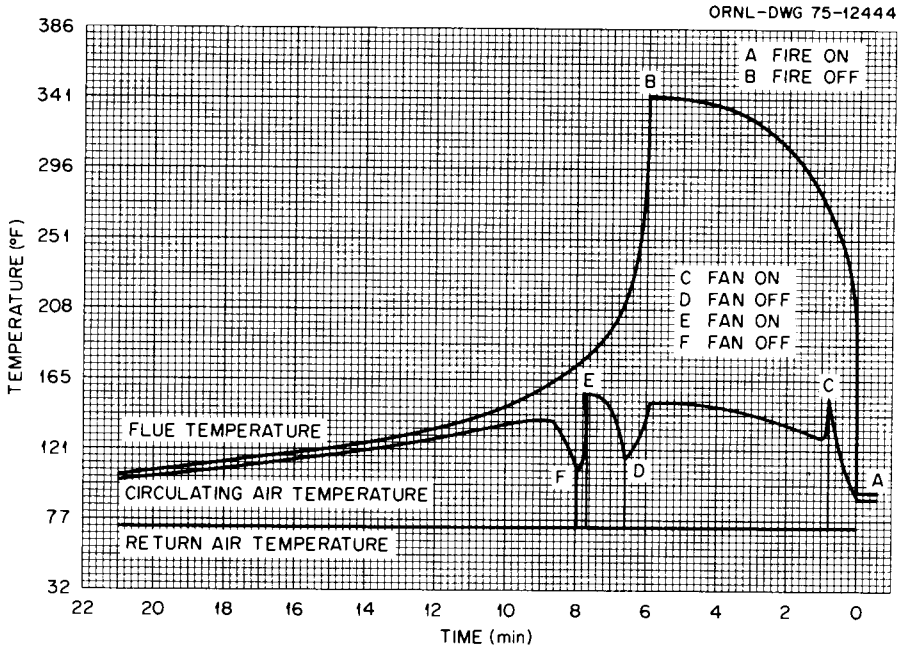

Fig. 3.10. Flue gas and circulating air temperatures for an atmospheric combustion furnace (high fan switch setting and 20% duty cycle). (Reproduced from chart graph.)

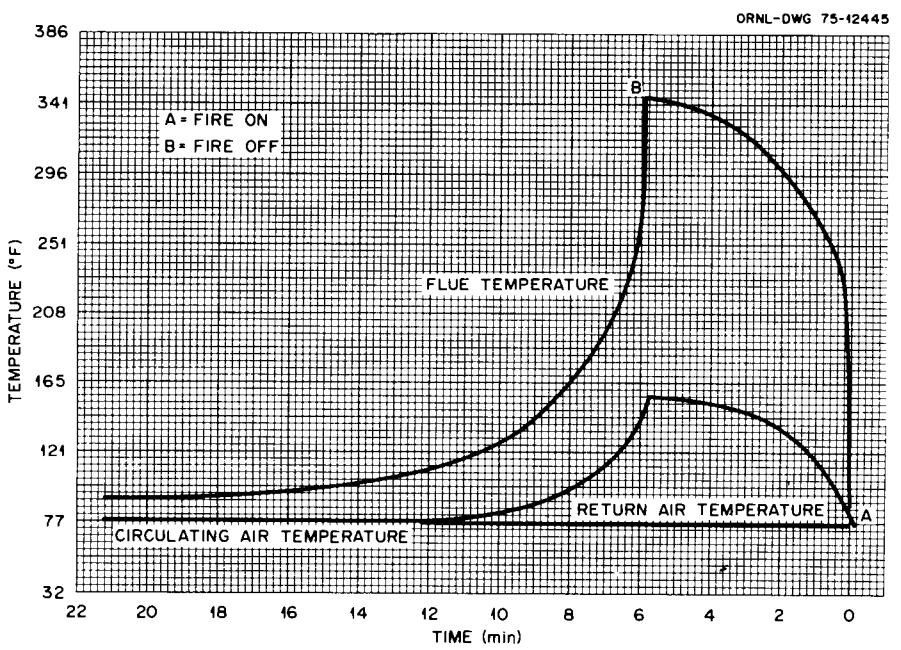

Fig. 3.11. Flue gas and circulating air temperatures for an atmospheric combustion furnace (low fan switch setting and 20% duty cycle). (Reproduced from chart graph.)

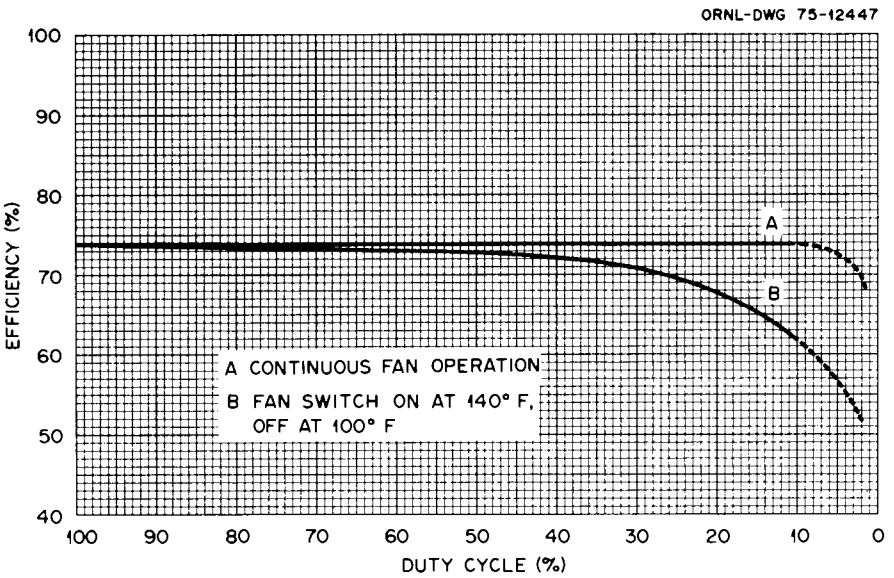

Fig. 3.12. Cyclic efficiency of a gas furnace (heat delivered to supply duct/heat content of gas and fan). All cycles are of 30-minute duration (20% duty — 6 min on and 24 min off). (Reproduced from chart graph.)

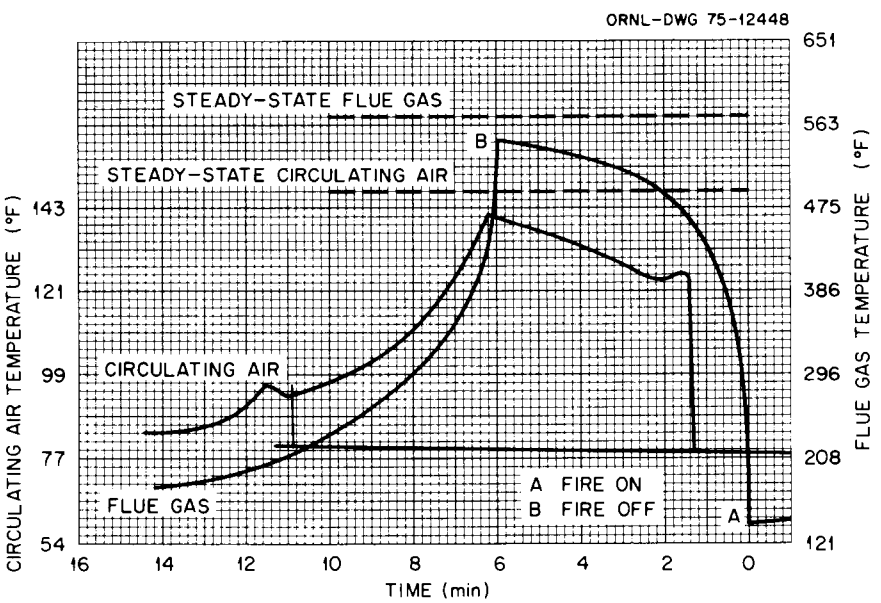

Fig. 3.13. Flue gas and circulation air temperatures for a sealed combustion furnace (high fan switch setting and 20% duty cycle). (Reproduced from chart graph.)

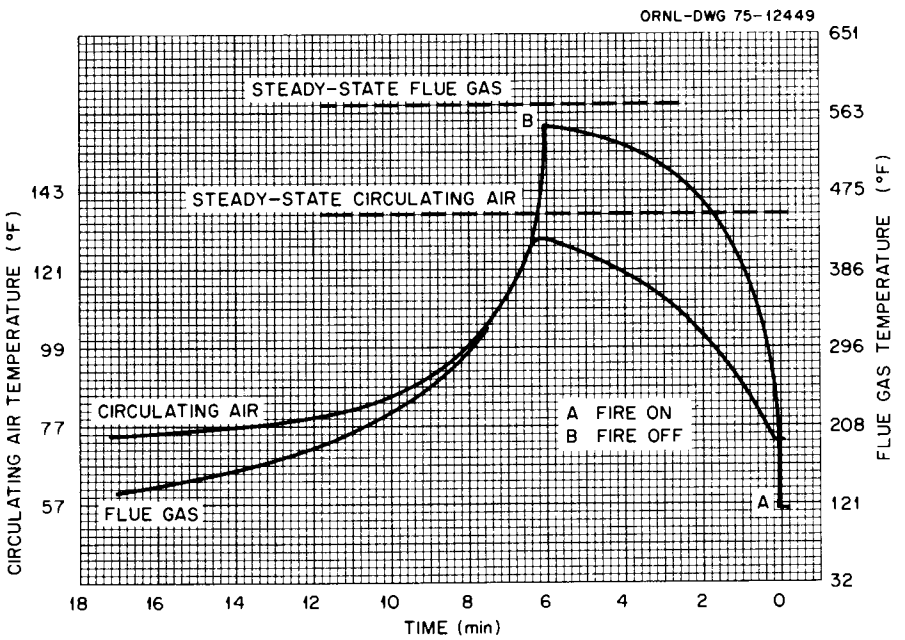

Fig. 3.14. Flue gas and circulating air temperatures for a sealed combustion furnace (low fan switch setting and 20% duty cycle). (Reproduced from chart graph.)

fan switch settings was plotted, and the seasonal efficiency of a sealed combustion furnace for various climates and conditions was calculated. The results are shown in Table 3.11.

For either type of furnace, there is a bonnet temperature below which blower operation results in a net loss of efficiency. The heat recovered from the exchanger is no longer worth the energy expended by the blower. The analyses of this balance for both energy and cost are presented in Appendix C. The economic breaking point varies with utility rates, but a good general recommendation is to set the fan switch to turn the fan off at 5 F° above room thermostat setting and to turn it on at as low a temperature above the "off" setting as fan switch construction permits.

Table 3.11. Calculated seasonal efficiency of a sealed combustion furnace

		Seasonal fuel utilization efficiency (%)	
		Fan switch setting	
Location	Furnace size[a]	High	Low
San Diego	C	61.1	62.1
	2×	58.4	61.6
Atlanta	C	71.2	73
	2×	66.5	71.9
Philadelphia	C	73.4	74.5
	2×	69.6	73.8
Minneapolis	C	75.5	76.5
	2×	72	76

[a]C = "correct"; 2× = twice oversize.

3.4 AGE AND MAINTENANCE

The ANSI full-load, steady-state efficiency test was performed on three residential furnaces in their "as-found" condition:

1. Coleman, 110,000 Btu/hr rated input, 13 years old

2. Coleman, 85,000 Btu/hr rated input, 5 years old

3. Fedders outdoor unit, 120,000 Btu/hr rated input, 9 years old.

The heat exchanger of furnace 1 was very rusty. The flue of furnace 2 had a long horizontal run from the furnace to the outside wall, with a water heater flue entering the horizontal run and an outside-mounted vertical stack extending just above the eave. There were a number of rust holes through the lower portion of the vertical flue. This flue functioned badly, and the draft hood frequently spilled flue gas into the basement. Furnace 3 appeared to be in good condition. It is a powered-combustion model (one equipped with a combustion air blower) and requires no flue extension beyond the housing.

All three of these furnaces gave a full-load, steady-state efficiency (based on stack CO_2 concentration and temperature) ranging from 73 to 77%. In these three cases, at least, no significant loss of efficiency could be attributed to aging.

4. COMPUTER ANALYSIS

Two operational losses that detract from the full-load, steady-state efficiency of the gas furnace were the cyclic losses and the infiltration losses, both of which vary with the duty cycle or load. Thus for a given installation, that is, a specific size furnace in a specific residence, the "instantaneous" efficiency can be calculated for the indoor and outdoor conditions of the moment. If these "instantaneous" efficiencies are properly totaled for the heating season, the seasonal fuel-utilization efficiency can be determined as a function of such variables as installed capacity, fan switch setting, and source of combustion air.

An existing computer program was used along with appropriate weather tapes to calculate the heating and cooling loads for an 1800 ft^2 model home, insulated to the appropriate FHA standard for each climate region. A subroutine was added which permits the installation of a furnace of any capacity, fan switch operation, and source of combustion air and which incorporates the experimentally determined values for cyclic efficiency and combustion air. The computer calculates the heating requirement each hour, determines the percent duty cycle for the furnace installed, and, referring to its program, determines the combustion and draft hood air associated with that duty cycle and that indoor-outdoor temperature differential. The heating load associated with that infiltration is added to the residence heating requirement, and the computational loop is iterated several times to arrive at the total heating load and duty cycle. The computer then determines the efficiency associated with that duty cycle and fan switch setting and computes the input of gas and electrical energy required to satisfy the load. The residence heating requirement, the infiltration load, and the energy input are computed for the season; and the seasonal fuel utilization efficiency is calculated.

The model house data were calculated for a location in each of the four climate zones with the insulation and glazing specified for that zone (HUD Minimum Property Standards 4900.1, 1973 Edition, July 1974 Revision). In each case the parameters studied were:

Furnace size

a. equal to the maximum demand

b. twice maximum demand — considered to be an average installation

Fan switch setting

a. on at 140°F, off at 100°F (High)

b. on at 10 F° above room, off at 5 F° above room (Low)

Type of furnace

a. indoor, atmospheric combustion

b. indoor, sealed combustion.

The results of this parametric study are shown in Table 4.1.

Table 4.1. Calculated furnace efficiencies as a function of type, size, adjustment, and climate

Location	Installed furnace size (Btu/hr input)	Fan switch setting	Seasonal fuel utilization efficiency (%)[a]	
Degree-day range			Atmospheric combustion	Sealed combustion
San Diego 0-2500	35,000	High	51.4	61.1
		Low	56.0	62.1
	70,000	High	46.3	58.4
		Low	53.2	61.6
Atlanta 2500-4500	80,000	High	58.3	71.2
		Low	65.8	73.0
	160,000	High	50.3	66.5
		Low	60.2	71.9
Philadelphia 4500-8000	60,000	High	60.3	73.4
		Low	67.2	74.5
	120,000	High	51.7	69.6
		Low	61.4	73.8
Minneapolis >8000	80,000	High	61.4	75.5
		Low	68.0	76.5
	160,000	High	51.6	72
		Low	61.0	76

[a]Based on the measurements made on commercially available furnaces with standing pilots.

The anomalous appearance of the Atlanta requirement of a furnace size which exceeds Philadelphia's and equals the Minneapolis requirement results from the increasingly higher insulation and glazing standards for colder climates and from identical "worst-day" temperatures for Atlanta and Philadelphia.

In climates above 2500 degree-days, the seasonal efficiency is about 50% if we assume that a typical, existing indoor, atmospheric combustion furnace is oversized to twice the maximum heating demand; draws its combustion air from within the conditioned space; and has a fan switch set to turn on at 140°F and off at 100°F. If the switch setting is changed to on-at-80°F, off-at-75°F, the calculated seasonal efficiency is increased to about 58%. If the furnace is replaced with a correctly sized furnace with the low fan switch setting, the calculated seasonal efficiency rises to 63 or 64%; and if the furnace is replaced with a sealed combustion furnace of the appropriate size, the calculated efficiency is 73 to 76%.

As can be seen from Table 4.1, the above statements do not hold true for a very mild climate, such as San Diego, because the nature of the climate is such that even the smallest furnace that will satisfy "worst-day" conditions is grossly oversized and is, therefore, on very low duty cycles (or standby) most of the time. In that climate the pilot consumption for the assumed nine-month heating season amounts to one-fourth, or more, of the total gas consumption. Thus the efficiency of the assumed typical installation would improve from 45% to about 61%, with an annual savings of about 6000 ft^3 of gas, if the standing pilot were replaced with electric ignition. The nonrecoverable pilot consumption becomes a smaller percentage of the total gas consumption as the climate becomes more severe and as the duty cycle of the furnace increases. As shown in Table 3.2, the nonrecoverable pilot consumption in Philadelphia for a 120,000-Btu/hr-input, atmospheric combustion furnace at the high fan switch setting is 9 to 10% of the total.

5. RECOMMENDATIONS

The recommendations for improvement in the fuel-utilization efficiency of gas furnaces are divided into those applicable to existing installations and those applicable to new furnaces and new installations.

5.1 EXISTING INSTALLATIONS

5.1.1 Duct systems

The greatest controllable loss from a forced, hot-air heating system is from uninsulated or leaking ducting. The use of uninsulated ducts in crawl spaces and attics has been quite common in some areas. The rationale offered with respect to crawl spaces has been that the heat loss into the crawl space would warm the floors and improve comfort. If this rationale is followed, then during the heating season the crawl space should be as well insulated and sealed as the house.

It is recommended that:

1. all ducts be examined and all leaks repaired.

2. the ducts in unheated spaces be at least as well insulated as the house.

3. vertical ducts in walls or chases be sealed between the duct and the building structure to prevent upward convection, or "chimney," flow of air beside the duct.

5.1.2 Fan switch setting

The second-greatest, controllable loss results from the cyclic operation of the furnace. This loss can be nearly eliminated by setting the fan switch to extract all the useful heat from the heat exchanger. The rationale for the high fan switch setting has been that moving air feels cooler than still air at the same temperature and that, thus, a direct register discharge below about 100°F may not give the desired sensation of heating. Directing or baffling the registers away from likely locations of room occupants can overcome this objection. The efficiency of the system will be appreciably improved by setting the fan switch to turn off at about 75°F and on as close above that as fan switch construction permits. An unquantified increase in system efficiency will also result because the longer blower operation reduces stratification.

5.1.3 Pilot

Pilot operation cannot, of course, be discontinued on existing installations during the heating season. However, turning the pilot off during the three- to six-month nonheating season will save 750 to 1000 ft^3 of gas per month for that period.

5.1.4 Routine cleaning and maintenance

Our tests showed no significant loss of efficiency resulting from poor burner adjustment, dirty filters, etc.; however, our laboratory and field tests were not exhaustive. In the interest of safety, as well as possible improvement in efficiency and performance, routine cleaning and adjustment should be performed.

5.2 NEW FURNACES AND INSTALLATIONS

5.2.1 Sealed combustion furnace

Under comparable conditions the sealed combustion furnace had a seasonal efficiency 6 to 20% higher than the atmospheric combustion furnace. Sealed combustion furnaces are currently being manufactured and are installed almost exclusively in mobile homes. There appears to be no technological or economic bar to their use. It is recommended that all new furnaces be designed for sealed combustion and indoor installation.

5.2.2 Efficiency standard

It is recommended that the present ANSI standard for the full-load, steady-state efficiency be raised from 75% to 80%.

5.2.3 Ignition

It is recommended that all new central furnaces be equipped with electric ignition instead of a standing pilot.

5.2.4 Fan switch setting

It is recommended that the fan switch be factory set to turn off at 75-80°F and on at about 90-95°F. The differential between off and on should be only great enough to prevent fan recycle.

5.2.5 Installation codes and standards

As mentioned earlier, one of the major losses in a heating system can result from an improper installation. Both the FHA minimum property standards and the building code recognize an obligation to energy conservation by specifying the performance of the thermal envelope. This obligation should be extended by specifying the performance of the heating (and cooling) system. Items that should be encoded are the capacity, integrity, and insulation of the duct system.

Similarly, the ANSI standard for mobile home ducting should be upgraded.

These recommendations can be summarized. There should be an immediate program to discontinue the manufacture and installation of the conventional atmospheric combustion furnace and to develop and put into production a high efficiency furnace having sealed combustion; electric ignition; a full-load, steady-state efficiency of 80%; and a low fan switch setting. The FHA minimum property standards and the national, state, and local building codes should be revised to include adequate performance standards for heating systems. Incentives should be offered to encourage the replacement of existing furnaces with high-efficiency units and to encourage upgrading existing ductwork to the new standards. Whereas existing systems probably have a seasonal efficiency of 50%, new systems meeting the recommended standards should have a seasonal efficiency of 75%, resulting in a fuel savings of one-third.

A similar program to improve the standards for the furnace and installation in mobile homes will result in a similar saving.

Appendix A

STACK HEAT RECOVERY DEVICES

The function of the heat exchanger in a furnace is to extract the heat from the combustion gases and deliver it to the room. The higher the percentage of heat it extracts, the more efficient it is; conversely, the higher the percentage of heat that goes up the stack, the less efficient the exchanger. The magnitude of the stack heat loss as a function of stack temperature is indicated in Fig. A.1, which is valid only for the specific CO_2 concentration of 8.0%. This might be typical of a sealed combustion gas furnace or of an oil furnace but does not apply to an atmospheric combustion gas furnace since the CO_2 in its stack is 3.5 to 4.5%.

In existing furnaces that have a high stack temperature, it is possible to recover some of the lost heat by installing additional heat transfer surface in the stack.

A commercially available stack heat recovery unit was tested to determine its effectiveness. The unit is shown in Fig. A.2. The five heat pipes extending to the left are to be installed extending into the stack, where they recover heat and transport it to the right, where the fan blows it into the room. The stack side is sealed from the room side so that no stack gas enters the room. The manufacturer states that if the temperature is below 450°F, the unit will not provide sufficient heat to justify its installation and that a stack temperature above 800°F may damage the unit. It is intended primarily for use on residential oil furnaces whose stack temperatures normally fall within that range. The unit was tested on the sealed combustion gas furnace with a normal stack temperature measured to be about 570°F and with a CO_2 concentration of 7.0-7.5% for a stack loss of 24 to 26%. After the unit was installed, the stack gas entering the heat pipes was measured to be 690°F and 8.5% CO_2, for a loss of about 26%; and the stack gas leaving the heat pipe was measured to be 475°F and 8.5% CO_2, for a loss of 19-20%. A complete heat balance around the furnace and the heat recovery device was also made as a cross-check.

The increase in flue temperature and CO_2 resulted from the increased flue resistance, which caused a decrease in combustion air flow.

Both the stack gas measurements and the heat balances showed that the device extracted from the stack and delivered to the room about 6000 Btu/hr. The input to the furnace was 100,000 Btu/hr; thus the device recovered about 6% of the total input, or raised the overall efficiency from 75 to 81%. The amount of heat that the unit can recover is a function of stack temperature and would run from about 4% at 450°F to 8% at 800°F.

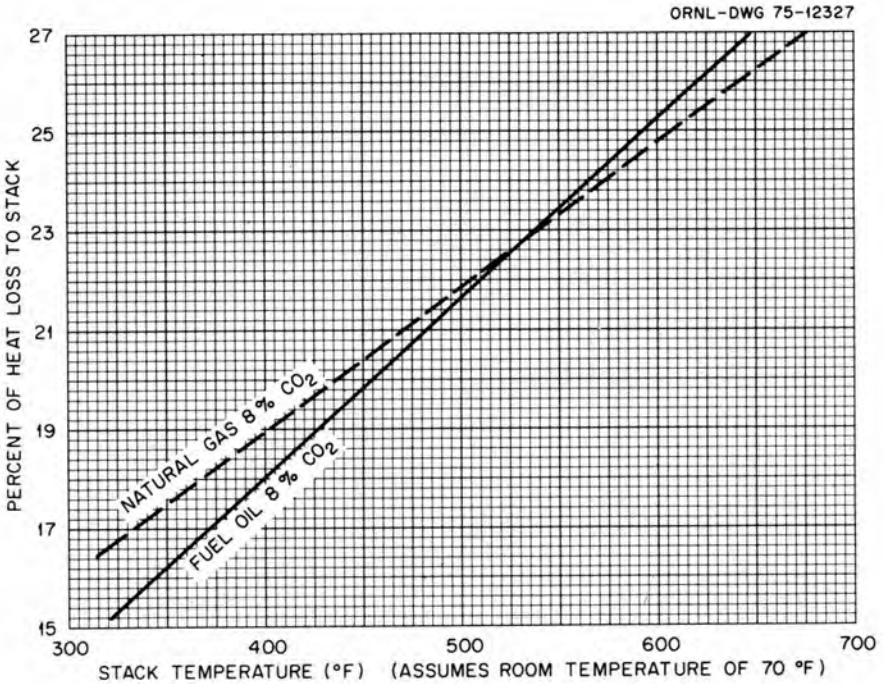

Fig. A.1. Stack heat loss of a sealed combustion gas furnace and an oil furnace.

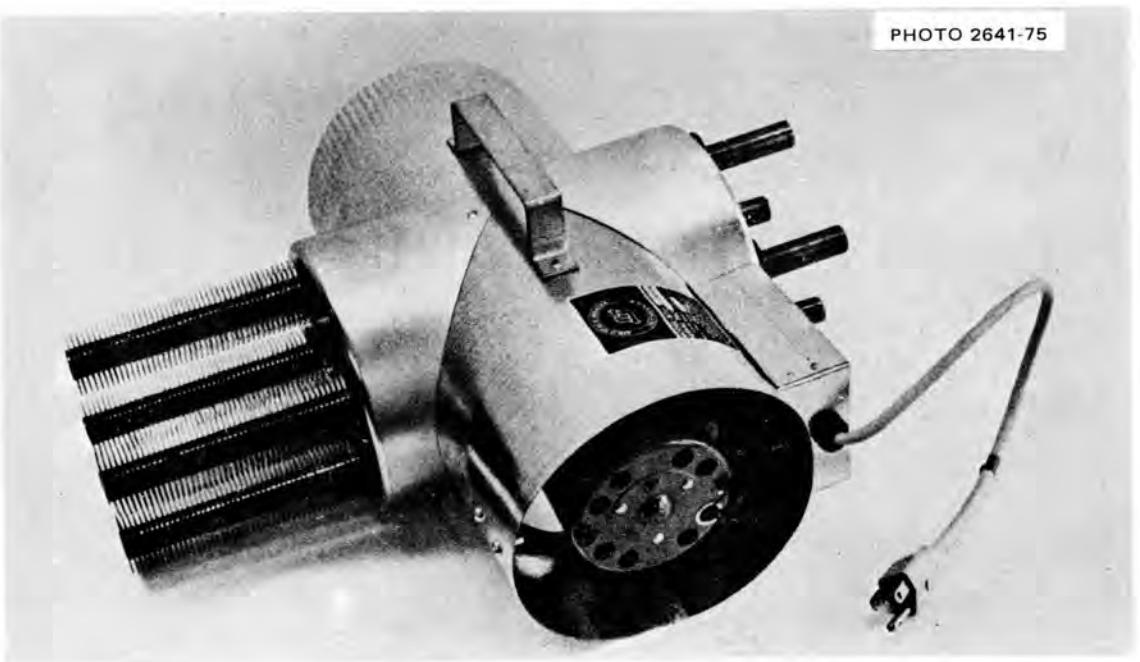

Fig. A.2. Stack heat recovery unit. Source: Installation Instructions for Air-O-Space Heater, Isothermics, Inc., August 20, 1974.

If we take the example of Table 3.2 (Sect. 3.1), the total gas consumption for the heating season was calculated to be 61,000 ft^3. An efficiency increase from 75 to 81% would save about 5000 ft^3/year. At present gas prices of \$1.50 to \$4.50 per 1000 ft^3, this amounts to \$7.50 to \$22.50 per year. The individual can determine the desirability of retrofitting a stack heat recovery unit to an existing furnace by first measuring the stack temperature and then estimating the percentage recovery and the annual saving on the total fuel bill.

Appendix B

EXPERIMENT DESCRIPTION

This appendix deals with the experimental setup, data collected, method of data collection, and equations used in performing the various tests conducted for this report. The topics discussed include steady state, cyclic, reduced firing, residential units, infiltration, duct losses, and seasonal efficiency.

The furnaces tested in the laboratory were an atmospheric combustion, forced-air-type central furnace rated at 110,000-Btu/hr input and a sealed combustion, forced-air-type central furnace rated at 100,000-Btu/hr input.

STEADY STATE

The steady-state test was performed to determine capability of the furnace to meet the ANSI standard of 75% efficiency.

The experimental setup used to perform tha ANSI steady-state efficiency test is shown in Fig. B.1. The instrumentation employed in performing the steady-state test included: a rotating-vane anemometer for determining circulating air flow; a hot-wire anemometer for determining combustion air flow; thermocouples for determining inlet wet- and dry-bulb temperatures, circulating air dry-bulb temperature, stack air dry-bulb temperature, and gas temperature; pressure gages for determining gas pressure and pressure drop across the blower; a Fyrite CO_2 meter for determining the percent CO_2 in the stack air; a barometer for determining barometric pressure; a watt-hour meter for determining electrical input to the blower; and a gas meter for determining the gas flow rate. This instrumentation allows collection of all the data necessary for analyzing the steady-state furnace efficiency.

The equations used for determining the furnace efficiency are:

$$C_p = 0.24 + 0.45 \times W \ , \tag{1}$$

$$h = 0.24 \times (t) + W \times [1061 - 0.444 \times (t)] \ , \tag{2}$$

$$\dot{m}_{as} = (\dot{V}_{as} \times A/SV) \times 60 \ , \tag{3}$$

$$\dot{m}_{ac} = (CFM_c/SV) \times 60 \ , \tag{4}$$

$$\dot{V}_f = \dot{V}_{fm} \ (P/407.52) \times [520/(T + 460)] \ , \tag{5}$$

$$Q_{oc} = \dot{m}_a \ (h_{out} - h_{in}) \ , \tag{6}$$

$$Q_{in} = (\dot{V}_f \times 1030) + (FW \times 3.412) \ , \tag{7}$$

Fig. B.1. Experimental setup for the steady-state efficiency tests.

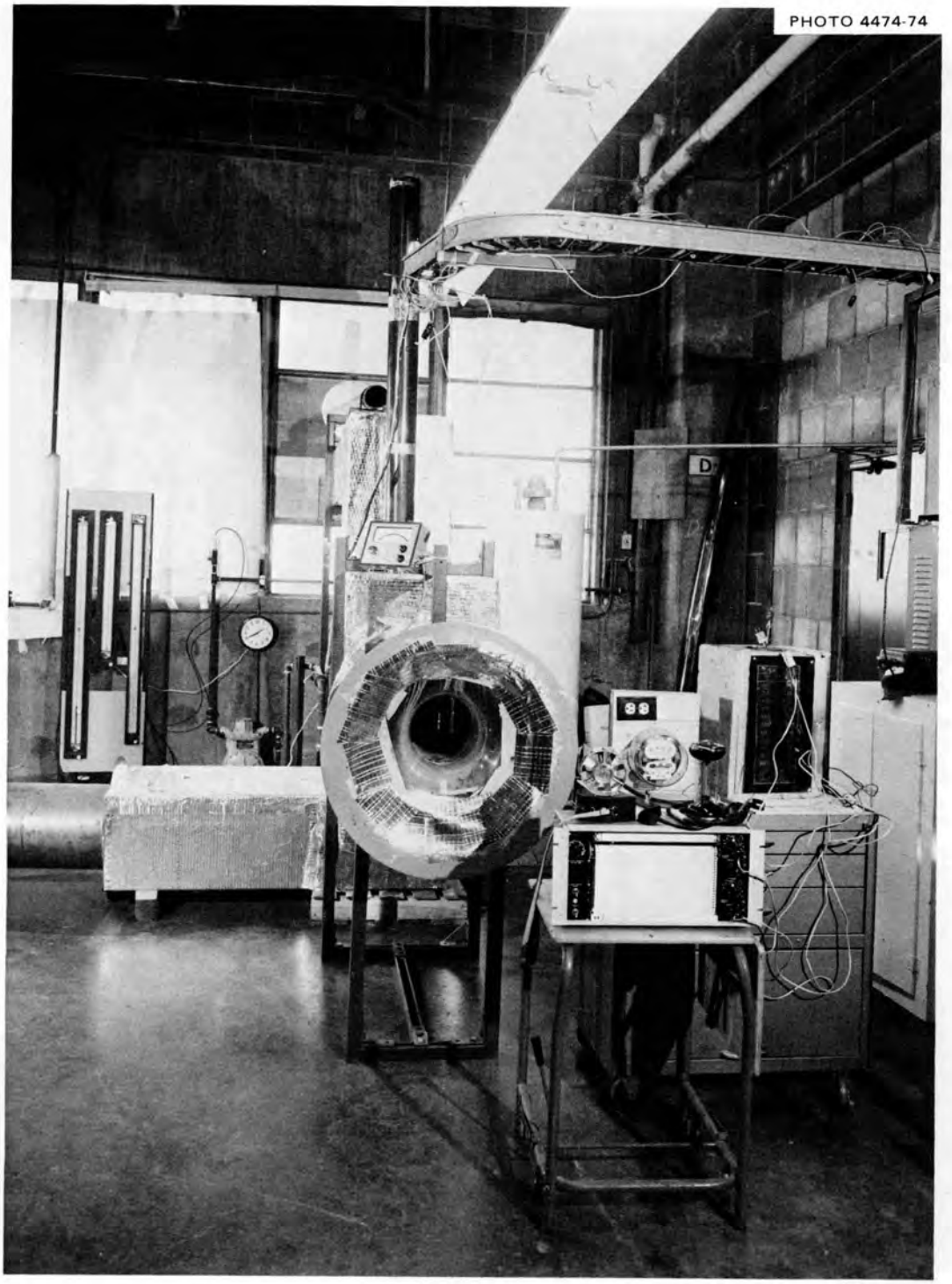

Fig. B.1 (continued)

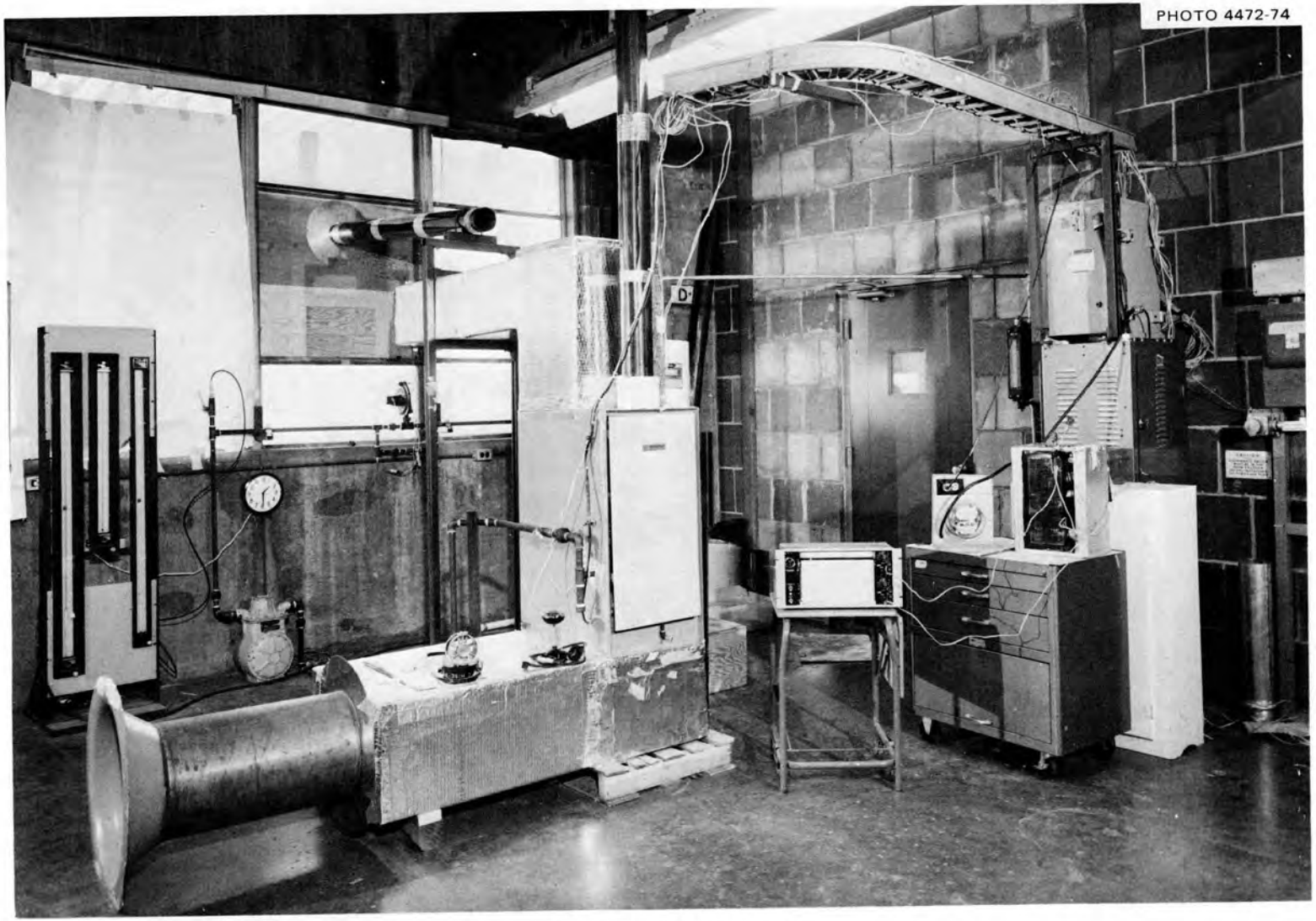

Fig. B.1 (continued)

$$Q_{os} = (\dot{m}_{as} \times C_p \times \Delta T) + \dot{V}_f \times 2.25 \times (1/SVG) \times 1185 , \quad (8)$$

Q_{os} = READ FROM ASHRAE NOMOGRAPH, *ASHRAE Handbook of Fundamentals, 1972*, p. 243, Fig. 2;

where

C_p = specific heat of inlet air, Btu/lb-°F,

W = humidity ratio of inlet air, lb_m water/lb_m dry air (where subscript m refers to mass),

h = enthalpy of air, Btu/lb_m,

t = dry-bulb temperature, °F,

$\dot{m}_{as}$ = mass flow rate of stack air, lb_m/hr,

$\dot{V}_{as}$ = velocity of stack air, ft/min,

A = area of inlet, ft^2,

SV = specific volume of inlet air, ft^3/lb_m,

$\dot{m}_{ac}$ = mass flow rate of circulating air, lb_m/hr,

CFM_c = volumetric flow rate of circulating air, ft^3/min,

$\dot{V}_f$ = flow rate of fuel, ft^3/hr,

$\dot{V}_{fm}$ = flow rate from gas meter, ft^3/hr,

P = absolute gas pressure, in. of H_2O,

T = gas temperature, °F,

Q_{oc} = heat output to circulating air, Btu/hr,

Q_{in} = heat input to furnace, Btu/hr,

FW = fan watt, watts,

Q_{os} = heat output stack gas, Btu/hr,

ΔT = $T_{stack} - T_{inlet}$, °F,

SVG = specific volume of gas at ambient conditions, ft^3/lb_m.

The humidity ratio was determined from the psychrometric chart using the inlet wet- and dry-bulb temperatures that were collected by thermocouples and recorded on a Brown multipoint recorder. The velocity of the stack air was measured with the hot-wire anemometer and read at 1-min intervals. The area of the combustion inlet was measured to be 0.301 ft^2. The specific volume of the air was taken from the psychrometric chart. The gas pressure was determined by adding the gage gas pressure to the barometric pressure. The fan watts were determined with a watt-hour meter. The stack and circulation air temperatures were measured by a nine-thermocouple grid, and both temperatures were recorded on the Brown recorder. The circulation air flow was measured with a vane-type anemometer, and the specific volume of natural gas was obtained from the supplier.

The sample calculations for these tests are contained in Appendix C.

CYCLIC

The purpose of the cyclic test was to determine if any efficiency loss occurrs during cyclic operation of the furnace. In order to perform this test in the laboratory, it was first necessary to choose some arbitrary cycle. This was chosen to be 30 min, with the fire burning for 3, 6, 12, 18, and 24 min. These firing durations produced duty cycles of 10, 20, 40, 60, and 80% respectively.

The experimental setup used during the cyclic tests was the same as for the steady-state test, as shown in Fig. B.1. The equipment used to obtain the raw data was the same as that used to perform the steady-state test with the addition of a two-pen strip chart recorder and planimeter for measuring the areas under the curves produced by the strip chart recorder.

The equations used for determining the furnace efficiency were Eqs. (1), (4), (5), and (7) from the steady-state test, in conjunction with the following equations.

$$Q_{oc} = (\dot{m}_{ac} \times 45 \times A_c \times C_p)/60 \ , \tag{10}$$

$$Q_{osl} = \dot{V}_f \times 2.25 \times 1185/SVG \ , \tag{11}$$

$$Q_{oss} = C_p \times 45 \times 0.301 \left(\sum_{i=1} V_{as} \times A_s \right) , \tag{12}$$

$$Q_{os} = Q_{osl} + Q_{oss} \ , \tag{13}$$

where all symbols are the same as in the steady-state equations with the exception that A = area under curve (min-mV), c = circulating air curve, and s = stack air curve. All Q's are in Btu's, with Q_{osl} being the latent heat of the stack air and Q_{oss} the sensible heat of the stack air. SVG is the specific volume of the fuel (ft^3/lb_m).

The pertinent data were collected in the same manner as described in the steady-state section, using a planimeter to measure the area under the curves produced by the two-pen recorder that recorded stack temperature and circulating air temperature. Typical recordings are shown in Figs. 3.10, 3.11, 3.13, and 3.14 of this report.

The sample calculations for this section are contained in Appendix C.

REDUCED FIRING

The purpose of this test was to determine whether any improvement in efficiency could be obtained by reducing the furnace firing rate. In order to perform this test, the furnace was set up in the same manner as for the steady-state test, as shown in Fig. B.1. In addition to this setup, it was necessary to restrict the flow of secondary air to the furnace by blocking off some portions of the secondary air openings with metal plates.

Both steady-state and cyclic tests were run in this mode using the same equations as in the full-firing, steady-state, and cyclic tests.

The sample calculations for these tests are in Appendix C.

RESIDENTIAL UNITS

In order to determine the effects of installation and age on gas-fired furnaces, it was necessary to move the experiment to some installed furnaces. The furnaces tested were: (1) Coleman central unit, 110,000-Btu/hr rated input, 13 years old; (2) Coleman, central unit, 85,000-Btu/hr rated input, 5 years old; and (3) Fedders outdoor unit, 120,000-Btu/hr rated input, powered combustion, 9 years old. All three of these furnaces were tested for their full-load, steady-state efficiency, but only furnace No. 1 was tested under cyclic operations.

The equations used to calculate furnace efficiency were the same as those previously described; however, the instrumentation was slightly modified. Temperatures were measured by thermocouples from a potentiometer instead of the Brown multipoint recorder. Air and gas flows were determined by the same instrumentation, and the stack and circulating air temperatures were recorded by the two-pen recorder.

The sample calculations for these tests are in Appendix C.

INFILTRATION

In order to determine the amount of energy lost by using room air for combustion and dilution air, it was necessary to determine the air flow induced by the stack during operation. For the purpose of this experiment, 100% of the stack air was considered as infiltration.

Equations (2) and (3) were used for determining infiltration losses, along with

$$Q_{inf} = \dot{m}_{as} (h_{out} - h_{in}) , \tag{14}$$

where

Q_{inf} = infiltration loss, Btu/hr,

h_{out} = enthalpy of air leaving stack,

h_{in} = enthalpy of air entering stack.

The experimental setup was the same as for the steady-state test and is shown in Fig. B.1. No additional instrumentation was needed to perform this experiment, and the data were collected in conjunction with the steady-state and cyclic tests.

The sample calculations for these tests appear in Appendix C.

DUCT LOSSES

In order to determine the magnitude of the losses incurred by running uninsulated ducts through unconditioned spaces, it was necessary to determine a "U" factor for both an insulated and an uninsulated section of duct.

The experimental setup consisted of a 10-ft section of 24 gage duct, with thermocouple grids at each end, attached to the furnace supply exit.

The equations used for determining the duct loss were:

$$Q_{lost} = \dot{m}_{ac} C_p \, (T_{in} - T_{out}) \, , \tag{15}$$

$$\text{"U"} = \frac{Q_{lost}}{A_s (T_m - T_\infty)} \, , \tag{16}$$

where

T_m = mean duct air temperature, °F,

T_∞ = ambient air temperature, °F,

A_s = surface area of duct, ft^2.

The sample calculations for this test are in Appendix C.

SEASONAL EFFICIENCY

In order to calculate seasonal furnace efficiency, it was necessary to simulate furnace operation over an entire season. This simulation was carried out by adapting the steady-state and cyclic data to a computer format and writing a subroutine that could be used in conjunction with an existing computer program that predicts space-heating loads for

residences, based on weather data. By simulating the furnace in this manner, it was possible to determine the seasonal efficiency of the furnace over the entire heating season.

Appendix C

SAMPLE CALCULATIONS

This appendix contains examples of the various calculations performed during this investigation. A complete list of the equations used is located in Appendix B.

STEADY STATE

In order to determine steady-state furnace efficiency, it is necessary to calculate the stack loss, the heat gain of the circulating air, and the heat input to the furnace.

The raw data for this sample case are:

inlet dry-bulb temperature	$t_{db} = 79°F$
inlet wet-bulb temperature	$t_{wb} = 64°F$
gas temperature	$T = 542°R$
gas pressure	$P = 7$ in. $H_2O + P_B$
barometric pressure	$P_B = 406.5$ in. H_2O
stack temperature	$T_S = 370°F$
circulating air temperature	$T_C = 164°F$
circulating air flow	$CFM_C = 944\ ft^3/min$
stack air flow	$\dot{V}_{as} = 142\ ft^3/min$
fan watts	$FW = 343\ W$
percent CO_2	$CO_2 = 5\%$
measured fuel flow rate	$\dot{V}_{fm} = 109\ ft^3/hr$

In order to determine the furnace efficiency, it is necessary to calculate the fuel flow rate at standard temperature and pressure and the total input from both fuel and electricity.

$$\dot{V}_f = \dot{V}_{fm}\ (P/407.52)(520/T) \tag{5}$$

$$= 109\ [(406.5 + 7)/407.52](520/542) = 106.1\ ft^3/hr\ ,$$

$$Q_{in} = \dot{V}_f \times 1030 + FW \times 3.412 \tag{7}$$

$$= 106.1 \times 1030 + 343 \times 3.412 = 110{,}455\ Btu/hr\ .$$

After determining the heat input, it is necessary to determine the heat output through the circulating air.

$$h = 0.24(t_{db}) + W[1061 + 0.444(t_{db})] \quad (2)$$

$$h_{in} = 0.24(79) + 0.0093[1061 + 0.444(79)] = 29.15 \text{ Btu/lb}_m ,$$

$$h_{out} = 0.24(164) + 0.0093[1061 + 0.444(164)] = 49.9 \text{ Btu/lb}_m ,$$

$$Q_{oc} = \dot{m}_a(h_{out} - h_{in}) \quad (6)$$

$$= 944(1/13.77)(60)(49.9 - 29.15) = 85{,}350 \text{ Btu/hr} .$$

This results in an efficiency (η_c), based on circulating air, of

$$\eta_c = 85{,}350/110{,}455 = 77.3\% .$$

The last calculation in the steady-state test is that of stack loss.

$$\dot{m}_{as} = \dot{V}_{as} \times A \times 60/SV \quad (4)$$

$$= 142 \times 0.301 \times 60/13.77 = 186.24 \text{ lb}_m/\text{hr} ,$$

$$C_p = 0.24 + 0.45 \times W \quad (1)$$

$$= 0.24 + 0.45(0.0093) = 0.244 \text{ Btu/lb}_m \cdot {}^\circ\text{F} ,$$

$$Q_{os} = (\dot{m}_{as} \times C_p \times \Delta T) + [(\dot{V}_f \times 2.25 \times 1185)/SVG] \quad (8)$$

$$= (186.24 \times 0.244 \times 291) + (106.1 \times 2.25 \times 1185/23.565)$$

$$= 25{,}230 \text{ Btu/hr} .$$

This results in an efficiency loss of $\eta_s = 25{,}230/110{,}455 = 22.8\%$.

This results in an overall heat balance of

$$\text{BALANCE} = \eta_c + \eta_s = 77.3 + 22.8 = 100\% .$$

The stack loss as read from the ASHRAE combustion nomograph is 22.5%, which also results in about a 100% balance.

CYCLIC

In order to calculate cyclic efficiencies, the same quantities must be measured as in the steady-state calculations. The difficulties arise from the fact that all these quantities must be determined under transient conditions.

In order to discuss cyclic tests in a more concise manner, the term *duty cycle* is used to indicate the time the fire is on, divided by the total time, and multiplied by 100. All calculated duty cycles are based on a total time of 30 min.

The raw data for this sample case are:

duty cycle	= 60%
inlet dry-bulb temperature	t_{db} = 67°F
inlet wet-bulb temperature	t_{wb} = 49°F
gas temperature	T_g = 67°F
gas pressure	P = 7 in. H_2O
barometric pressure	P_B = 399.64 in. H_2O
circulating air flow	CFM_C = 918 ft^3/min
fan watts	FW = 103 W
measured fuel flow rate	V_{fm} = 30.8 ft^3/hr

The heat input to the furnace is calculated in the same manner as in the steady-state calculation.

$$V_f = V_{fm} \times (P/407.52)(520/T) \tag{5}$$

$$= 30.8 \times [(399.64 + 7)/407.52](520/527) = 30.33 \text{ ft}^3/\text{hr} ,$$

$$Q_{in} = V_f \times 1030 + FW \times 3.412$$

$$= (30.33 \times 1030) + (102.9 \times 3.412) = 31{,}590 \text{ Btu/hr} .$$

In attempting to analyze the heat output through the circulating air, transients are incurred. In order to analyze this transient condition, it was necessary to use a strip chart recorder and analyze the area under the curves. The strip chart for this cycle is shown in Fig. C.1. The area under the curve is analyzed only during the period when the fan is running. This strip chart has millivolts recorded on the y axis and time running right to left on the x axis. This results in an area under the curve of minutes-millivolts. The heat output of the circulating air is defined by:

$$C_p = 0.24 + 0.45 \text{ W} \tag{1}$$

$$= 0.24 + 0.45(0.0033) = 0.2415 ,$$

$$\dot{m}_{ac} = CFM_c/SV \tag{4}$$

$$= 918/13.35 = 68.76 \text{ lb}_m/\text{min} ,$$

$$45°\text{F} = 1 \text{ millivolt} ,$$

$$Q_{oc} = 45 \times \dot{m}_{ac} \times C_p \times \Sigma \text{ AREA}$$

$$= 45 \times 69.67 \times 0.2415 \times \Sigma (31.44 + 0.3 + 0.2 + 0.2 + 0.1)$$

$$= 24{,}090 \text{ Btu} .$$

This results in an efficiency of

$$\eta_c = 24{,}090/31{,}590 = 76.3\% \ .$$

The final calculation is that of stack loss. This is accomplished in a similar manner as the heat output through the circulating air flow, the only difference being that the air flow is not a constant and must be included in the summation. The heat loss is determined by:

$$Q_{oss} = C_p \times 45 \times 0.301 \times [\Sigma(V_{as} \times A)]/SV \qquad (12)$$

$$= 0.2415 \times 45 \times 0.301 \times 17{,}660/13.35 = 4330 \text{ Btu} \ ,$$

$$Q_{osl} = V_f \times 2.25 \times 1185 \qquad (11)$$

$$= 30.33 + 2.25 \times 1185/23/565 = 3430 \text{ Btu} \ ,$$

$$Q_{os} = Q_{oss} + Q_{osl} \qquad (13)$$

$$= 4330 + 3430 = 7760 \text{ Btu} \ .$$

This results in a stack loss of

$$\eta_{sl} = 7760/31{,}590 = 24.6\% \ .$$

This gives a balance of

$$\text{BALANCE} = \eta_c + \eta_{sl}$$

$$= 76.3 + 24.6 \sim 100\% \ .$$

OPTIMUM FAN SETTING

In order to determine the fan setting that would minimize energy consumption, it was necessary to perform an energy balance around the furnace. The two quantities that had to be balanced were the energy consumed by the fan and the energy delivered to the conditioned space. The equation used for calculating the energy consumed by the fan was:

$$Q = \frac{FW + 3.412}{\eta_{net}} \ ,$$

where η_{net} is the network efficiency for the electricity (0.295).

The equation used for calculating the energy delivered to the conditioned space was:

$$Q = \dot{m}C_p \ \Delta T \ .$$

Figure C.1 shows a plot of these two equations vs ΔT. When the fan is operating, its energy consumption is less than the energy delivered to the conditioned space for all ΔT's greater than 5. From this figure, it can be shown that the optimum fan setting is 5 F° above room temperature.

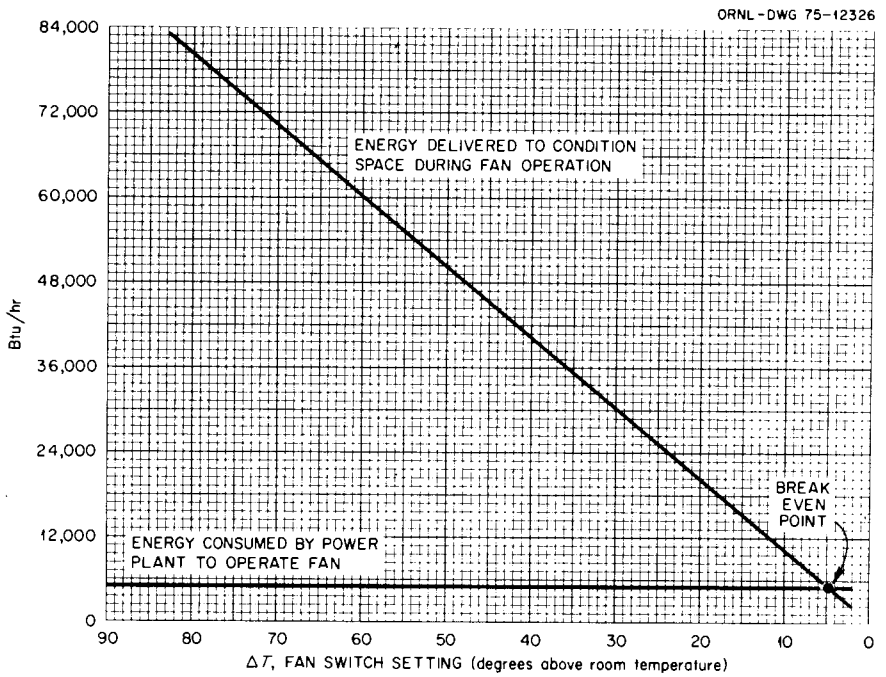

Fig. C.1. Fan energy consumption vs delivery of energy to the conditioned space.

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