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A Laboratory Study of A Gas-Fired Condensing Boiler

George E. Kelly
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Building Equipment Division
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U.S. DEPARTMENT OF COMMERCE, Philip M. Klutznick, *Secretary*
Jordan J. Baruch, *Assistant Secretary for Productivity, Technology, and Innovation*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

ABSTRACT

As a part of the Department of Energy's energy conservation program for consumer products, the National Bureau of Standards (NBS) developed test procedures for conventional gas and oil-fired furnaces and boilers. The Department of Energy (DoE) published their finalized version of these procedures in the Federal Register on May 10, 1978. In an effort to update and refine these test procedures, DoE directed NBS to develop a method of testing condensing furnaces and boilers which could be used to compare the annual performance of condensing and non-condensing residential heating systems. This report summarizes the laboratory tests of a gas-fired pulse-combustion condensing boiler that were carried out as a part of the development effort.

The performance of the pulse-combustion boiler was evaluated under both steady-state and part-load operating conditions. The efficiency of the unit was determined by the input/output method which measured the heat transferred to the circulating water and the energy input during each test period. Steady-state laboratory tests of the unit were conducted at constant return water temperatures of 100, 110, 120, 130 and 140°F (37.8, 43.3, 48.9, 54.4 and 60.0°C). Part-load performance tests were carried out at a number of these return water temperatures at on-times of approximately 5, 15, 22.5 and 42 percent.

A modified version of the heat loss procedure for estimating the seasonal performance of a residential central furnace or boiler was also used to evaluate the boiler's steady-state efficiency and part-load efficiency at a 22.5 percent on-time. A cool-down test and heat-up test were performed to obtain dynamic information which was used to calculate the unit's cyclic performance. The predicted steady-state and part-load efficiencies from the heat loss method were found to be within three percent of the performance determined using the input/output method.

Key Words: Boilers; central heating; condensing boilers; efficiency, part load; fossil-fuel heating systems; gas-fired boilers; hydronic heating; pulse combustion.

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NOMENCLATURE

A_s	area of equal jacket surface temperatures, in ft^2
C_L, C_L'	latent heat loss coefficients
C_p	constant-pressure specific heat for liquid and water vapor in the flue gases
h_c	coefficient of convection for each area of equal surface temperature, in Btu/(h ft^2°F)
h_{rs}	coefficient of radiation heat loss for each surface of equal surface temperature, in Btu/(h ft °F)
HHV	measured higher heating value of test gas, in Btu/lb
H_s	jacket heat loss for each surface in Btu/h
L_c, L_c'	cyclic and steady-state loss which corrects for hot condensate going down the drain instead of up the flue as heated vapor, in percent
$L_{L,A}$	latent heat loss, in percent
$L_{S,OFF}$	off-cycle sensible heat loss, in percent
$L_{S,ON}$	on-cycle sensible heat loss, in percent
$L_{S,SS,A}$	sensible heat loss at steady-state operation, in percent
$\dot{m}$	average water mass flow rate in lbm/h
p_v	partial pressure of water vapor in the flue gases at atmospheric pressure of p.
p	atmospheric pressure of 14.7 psia
p_v^s, $p_v^{s'}$	saturated vapor pressure of water at average on-period flue gas temperature and at steady-state flue gas temperatures respectively in psia
P_o	partial pressure of water vapor at 14.736 psia
P_{sw}	partial pressure of water vapor at 60°F of 0.2563 psia
Q_{IN}	fuel energy input rate during steady-state and cyclic operation test periods, in Btu/h
Q_{OUT}	heat transfered to circulating boiler water during steady-state and cyclic operation test periods, in Btu/h

Δt	length of test period
T_A, T_{LAB}	laboratory air tempeaturing during test periods, in °F
T_S	jacket surface temperature during jacket heat loss rate test, in °F
$\overline{\Delta T}$	average temperature rise across the boiler during steady state and cycle tests, in ΔT°F
$T_{F,ON}$	on-cycle flue gas temperature while the system is in cyclic operation, in °F
$T_{F,SS}$	flue gas temperature at steady-state, in °F
η_{SS}	steady-state efficiency
$\eta_{22.5\%}$	part-load efficiency at 22.5 percent on-time

1. INTRODUCTION

Condensing furnaces and boilers achieve high efficiencies through low flue gas temperatures and by condensing part of the water vapor generated as a result of burning hydrogen in the fuel, to recover its heat of vaporization. Since the formation of condensate requires cooling the flue gas to temperatures below the water vapor's saturation temperature, the heat exchanger's operating temperature must be kept in the condensing region if condensation is to be achieved. In addition, the unit must be equipped with a drain so that the condensed water can be continuously removed.

Manufacturers are in the process of developing condensing furnaces and boilers for the residential home heating market and there is a need for a standard testing and rating procedure to estimate the seasonal performance of these units and to allow their performances to be compared with each other and with non-condensing units. The purpose of the study described in this report was to provide technical data which would assist in the development of such procedures. Test methods for evaluating the performance of non-condensing furnaces and boilers were previously developed in references [1] and [2], but no credit was given in those documents for the recovery of the heat of vaporization of condensed water.

Laboratory tests were conducted on a condensing boiler having a counter flow, fire-tube heat exchanger and a natural gas fired pulse combustor. The unit employed a direct vent system which was designed to bring outdoor air in for combustion and to directly vent the flue products to the outside. A condensate drain was incorporated in the base of this unit to allow condensed water to exit the unit separately from the flue gases.

Steady-state laboratory tests of the unit were conducted at constant return water temperatures of 100, 110, 120, 130, 140°F (37.8, 43.3, 48.9, 54.4 and 60.0°C). At a number of these return water temperatures part-load tests were also performed corresponding to percent on-times of 4.9, 14.7, 22.5 and 42.0 percent.

Each steady-state test was also accompanied by a cool-down test and a heat-up test. The procedure employed required operating the boiler until steady-state conditions were achieved and all necessary steady-state measurements obtained. The unit was then shut down for a 50-minute cool-down period, during which the flue gas temperature was measured at several different times after the burner was shut off. It was then turned on and flue temperatures read after 1.0 and 5.5 minutes of operation.

The efficiency of the unit during the steady-state and part-load operation was determined by measuring the total heat transferred to the water during each test and dividing it by the measured energy input to the boiler during the test period. In addition to using this input/output method, the steady-state efficiency and part-load efficiency at a percent on-time of 22.5% were also calculated using a slightly modified version of the heat-loss procedure given in reference [3].

The heat-loss procedure substracts from the complete input energy identified losses, such as latent and sensible percent flue gas and condensate losses, attributable to operation of a heating systems.

Descriptions of the condensing boilers tested and the instrumentation employed are presented in sections 2 and 3 of this report, respectively. Section 4 discusses the test and calculation procedures, while sections 5 and 6 contain, respectively, a discussion of the experimental results and brief summary. A brief error analysis is presented in Appendix A.

2. DESCRIPTION OF TEST BOILER

The condensing boiler used in this study was a gas-fired 100 000 Btu/h (29307 W) input unit. It used a pulse combustion principle in conjunction with a counter flow, fire-tube heat exchanger and direct venting of flue gases. The unit was designed to take air for combustion directly from outside the residence. The boiler, which is shown in Figures 1 and 2, had a cylindrical shape and was 44 inches (112 cm) high with a 14-inch (35 cm) diameter. An external control box, mounted on the side, housed a starting timer circuit, a coil used to fire the spark plug, and pressure and temperature safety switches.

The pulse combustor operated in a cyclic manner. A thermostat "need-for-heat" signal to the controls began operation of the blower, which purged the combustion chamber and forced air into it at the same time the gas valves opened to allow gas to enter. The spark plug ignited the first few air/gas charges to cause combustion. As the charges exploded, the expanding gases closed the air/gas inlet valves and the rapid build up in pressure forced the flue gases to exit through the heat exchanger tubes and venting system. The drop in pressure in the combustion chamber, as a result of the flue products discharging through the heat exchanger, helped to draw in a new charge of air/gas which was ignited by the residual heat from the preceding charge. After a few combustion cycles, the process became self sustaining, and the spark plug and blower automatically shut off. The boiler continued to operate until the thermostat was satisfied and the gas valve was de-energized.

A counter flow heat exchanger allowed a low return water temperature to reduce the temperatures of flue gases exiting the unit and thus maximize the heat transfer to the circulating water. With moderate return water temperatures (below approximately 130°F (54.4°C)), the counter flow heat exchanger was capable of cooling the flue gases below the saturation temperature of the water vapor present in the combustion products. This resulted in part of the water vapor condensing and giving up its heat of vaporization. The unit was equipped with a condensate drain line (see Figure 2) for removal of this condensed water.

The high temperature safety shut-down control was supplied pre-wired with the unit. The settings of operating control devices were not changed from their manufacturer's settings during the tests described herein.

A flue pipe and a combustion air-inlet pipe were installed using 1.5 inch (3.8 cm) I.D. polyethylene pipe as specified in the installation notes supplied by the maufacturer.

Electrical power required by the boiler was supplied from a 120-volt service panel in the laboratory. Automatic switch contacts for the circulating pump, included as a part of the controls to cycle the pump on and off with the burner, were not used. Instead, the boiler and the water circulating pump were manually switched on-and-off.

3. TEST INSTRUMENTATION DESCRIPTION

Testing was conducted using the boiler test apparatus schematically shown in Figure 3. An insulated 125 gallon (0.47 m^3) water tank, circulating pump, hot water dump, auxiliary boiler, and city water makeup comprised the major support equipment. In order to achieve repeatable water flow between tests, pairs of manually operated valves were employed - one valve for flow control and the other for open/shut (flow/no flow) operation.

The auxiliary boiler, a gas-fired 85 kBtu/h input unit, was used to raise the temperature of the water in the storage tank between tests and to maintain uniform return water temperatures to the test boiler during cycling tests involving long off-periods. Pump and boiler operation was controlled by manual operation.

The direct vent exhaust/air intake system shown in Figure 4 was used during all tests. It was designed and installed in accordance with the recommendations of the manufacturer. Vent lengths were selected to be typical of what would be expected if the boiler had been installed in a single-story dwelling with the exhaust vent passing through the roof.

Condensate was collected in a stainless steel beaker placed under the drain line attached to the boiler and weighed immediately after each collection period on a scale having an error of less than $\pm$ 10 milligrams. A glass flask was also connected to the bottom of the flue pipe outlet (see detail A of Figure 4) to catch any condensate which formed within the flue pipe. The latter was not included in the reported condensate quantities since the heat of vaporization given up when it condensed would be transferred within the flue pipe and not within the boiler.

A turbine meter in the return pipe to the boiler was used to measure the water flow through the boiler. A manually read totalizer which counted pulses from the turbine meter was used to determine the total water flow through the boiler during each test. The accuracy of the turbine meter was verified using a weigh tank and a timer at return water temperatures from 100 to 140°F (37.8 to 60.0°C). Occasional calibration checks were made throughout the period of time that tests were being conducted.

A 40-junction thermopile made from 30 AWG type T (coppper/constantan) thermocouple wire was used in stainless steel wells inserted in the supply and return water lines to measure the temperature differential across the boiler. The stainless steel wells had an outside diameter of 3/8 inches (0.95 cm) and a 6-inch (15.2 cm) immersion length, as recommended in ASHRAE Std. 41.1-74 (4). All other temperature measurements were made using type T thermocouples.

A multipoint strip-chart recorder was used, in addition to the digital data logger, to provide a continuous recording of the temperature differential across the boiler. A planimeter with a minimum scale division of 0.01 square inches (0.065 cm^2) was used to find the area under this curve in order to determine the average temperature differential across the boiler during the cyclic tests. The actual temperatures of the supply and return water and the flue gases were also continuously recorded for the purpose of monitoring the test results. All other temperatures were recorded on the digital data logger at ten-minute intervals.

The wet-bulb and dry-bulb temperatures of the entering combustion air were measured using a pair of thermocouples and a blower as described in reference 4. The wet bulb thermocouple was equipped with a wick, which was supplied with distilled water from a reservoir. This psychrometer was located at the same elevation as the combustion air intake, as illustrated in Figure 4.

The flue gas temperature was measured using bead-type thermocouples manufactured in an inert gas thermocouple welder. They were installed as a 5-in-1 averaging thermocouple placed in the flue pipe 15 inches (38 cm) from the boiler exit. Radiation shields were not necessary because the design of the test boiler prevented the thermocouple from "seeing" the flame. The first 18 inches (46 cm) of the flue pipe downstream from boiler were insulated with foil-covered insulation.

Flue gas analyses were performed to determine the concentration of carbon dioxide, oxygen and carbon monoxide present in the flue gases. Samples were drawn from the center of the flue pipe at the temperature measurement plane. The carbon dioxide concentration was determined using an infrared absorption analyzer. The sample train included a vapor trap, particle filters, and a flow meter. The instrument was checked each test day using a zero (0.0%), a midrange (5.13%), and a full-scale (15.2%) span gas consisting of CO_2 in nitrogen. The percentage by volume of oxygen present in the flue gases was measured using an electrochemical sensor with oxygen-sensitive electrolyte. The concentration of oxygen provided a check on the CO_2 measurement and was not used in the calculation procedure. The carbon monoxide concentration in the flue gases was determined using a length of stain detector sample tube. The sample was passed through a vapor trap and the sample tube by a manually operated pump. The concentration of CO_2 in parts-per-million (ppm) was based on the length of color change from yellow to brown of potassium palladosulfite impregnated silica gel in a glass tube. This gave a reading within $\pm$ 10 ppm in the 0 to 175 ppm range. The carbon monoxide concentration readings were taken as a check of the boiler's combustion performance and did not enter into the efficiency calculations.

The higher heating value of the natural gas used was continuously recorded on a gas calorimeter located at a nearby NBS building. The amount of gas consumed during a test was measured with a dry-type positive displacement meter with a one cubic foot per revolution register. The gas temperature at the meter was measured using a type T thermocouple and the gas pressure at the meter was determined using an open tube manometer and a barometer. Electrical energy consumption of the boiler was measured by a 2 wire, 115-VAC, watt-hour meter having a resolution of .01 kWh. The line voltage was monitored with a digital voltmeter.

4. TEST AND CALCULATION PROCEDURES

Three types of tests were performed on the boiler: steady state, cool-down/heat-up, and cyclic. During each steady state test a constant return water temperature was maintained and the water flow rate was adjusted to give a constant temperature rise across the test boiler.

A steady-state test always preceded a cool-down/heat-up test. The latter consisted of a 50-minute off-period, during which there was no water flow through the boiler, followed by a brief on-period at the same return water flow rate and return water temperature as the preceding steady-state test.

The cyclic tests consisted of several on/off cycles at a constant return water temperature. During both the cool-down and heat-up portion of the tests, the water flow rate was maintained at a value which resulted in a 20°F (11.1°C) temperature rise across the boiler under steady-state operation at the same return water temperature. Maintaining water flow through the boiler during the off-period of the cyclic tests was necessary in order to accurately determine the heat transferred to the circulating water using the input/output method of testing.*

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* This is because the time constant of the thermocouples and wells tended to yield results which underestimated the amount of heat transferred to the water during the first few minutes of boiler operation. Operation with continuous water flow minimized this problem, since the measurement errors immediately after start-up and after shut-down tended to cancel each other.

4.1 STEADY-STATE TEST PROCEDURE

Prior to beginning the actual steady-state tests, the boiler was operated at the desired return water temperature until changes in the temperature differential across the boiler were less than $\pm$ 0.5°F (0.28°C) per hour.

At least two hours of boiler operation were necessary to achieve this condition, since the firing-rate of the test boiler tended to slowly decrease with time after the initial start-up. In order to assure a constant return water temperature to the test boiler, the temperature of the water in the storage tank was maintained at a constant value by adjusting the rate at which heated supply water was dumped and replaced by 60°F (15.6°C) city water. The flow rate through the boiler was adjusted to give a constant 20°F (11.1°C) temperature rise through the boiler under steady-state operation.

The actual steady-state test consisted of one hour of continuous operation at the desired return water temperature, during which the gas input, gas pressure, barometric pressure, water flow rate, electric consumption, and voltage were monitored and recorded at ten-minute intervals. The concentrations of CO_2, CO, and O_2 in the flue gases were measured once during each steady-state test. The condensate was collected over a period of time which was long enough to give an accurate estimate of the rate of condensate formation. The temperature differential across the boiler, the return water temperature, the supply water temperature, and the flue gas tempertures were recorded on a multipoint strip-chart recorder and, at ten minute intervals, on a digital data logger.

4.2 COOL-DOWN AND HEAT-UP TEST PROCEDURES

The cool-down and heat-up test was always preceded by a steady-state test. They were conducted as described in Sections 3.2 and 3.3 of Reference 1.

To assure that no water flow occurred during the cool-down period (as required in [1]) a valve was closed on the supply side of the boiler immediately after the burner was turned off. During the off-period, flow was maintained through a by-pass loop in order to keep the "return water" at a uniform temperature prior to the start of the heat-up test.

The water flow rate at which the preceding steady-state test was conducted was maintained during the heat-up test.

Five flue gas temperature measurements were made during the cool-down and heat-up tests. Flue temperatures were recorded at 3.75, 22.5, and 45.0 minutes after the unit was turned off, and 1.0 and 5.5 minutes after burner start-up. These flue temperatures were used to estimate the part-load efficiency of the condensing boiler at a cycling rate having an on-period of 22.5%, as described in section 4.5. No attempt was made to measure the heat transferred to the circulating water during the heat-up tests.

4.3 CYCLIC TEST PROCEDURES

The boiler was manually cycled on-and-off based on observed clock test times. The lengths of the on and off-periods for the four different percent on-times studied are given in Table 1. Cyclic tests actually performed are indicated by the "X's" in this table.

Prior to each cyclic test, either a previous cycle test was performed or a brief period of continuous operation was carried out in order to establish the return water temperature and other test conditions. The boiler was operated at the desired cycle rate and return water temperature for at least one cycle preceding any test cycle used for calculating the efficiency.

The flow rate was set to maintain a return water temperature which would result in a 20°F (11.1°C) temperature differential across the boiler if the unit were operated in a steady-state manner at the same return water temperature. This flow rate through the boiler was maintained during both the on and off-periods.

The temperature differential across the boiler was measured using both a multipoint strip-chart recorder and a digital data-logger. A smooth curve was drawn through the individual temperature differential points on the strip chart and a planimeter was used to find the area under the curve. From this information, the average ΔT across the boiler during the entire on/off test cycle could be determined and the heat transferred to the circulating water per cycle could be calculated as described in section 4.5. The amount of gas consumed (corrected to standard conditions) by the boiler during a test cycle and the higher heating value of the fuel were measured with the instrumentation discussed in section 3.

Other temperature and pressure measurements were also performed and recorded throughout the test to assure that the unit was operating properly. Most of these measurements were identical to those made during steady state operation and are discussed in sections 3. and 4.1.

4.4 JACKET LOSS TEST

Jacket loss tests were conducted during steady-state operation at approximately 100° and 130°F (37.8°C and 54.4°C) return water temperatures. The boiler jacket was subdivided into areas of 5-inch squares (32.3 cm^2) and the surface temperature measured at the center of each with a thermocouple. This thermocouple was moved from point to point on the jacket surface. The temperatures were read on a digital temperature meter and were manually recorded. The radiation and convection losses were calculated by the procedures given in Appendix F of ANSI Z21.47-1973.

4.5 CALCULATION PROCEDURE FOR DETERMINING STEADY STATE AND PART-LOAD EFFICIENCIES

Using the input/output method, the efficiency of the test boiler was defined for both steady state and cyclic operation as:

$$\eta = \frac{(Q_{out})}{(Q_{in})},$$

where Q_{out} is the heat transferred to the circulating boiler water during the test period and Q_{in} is the energy input to the unit during the same period of time. For a steady state, this test period consisted of one hour of steady state operation, while for a cyclic test it consisted of one or more complete on/off cycles.

For all tests, the energy input, Q_{in}, was calculated by correcting the measured volume of gas consumption during the test period to standard conditions* and multiplying the result by the higher heating value (in Btu/ft^3) of the fuel. To account for the fact that the gas consumed by the test boiler was very dry (i.e., contained very little water vapor), the latter was adjusted by using the equation:

$$HHV|_{dry} = HHV|_{saturated} \left(\frac{P_o}{P_o - P_{sw}}\right),$$

where P_o is atmospheric pressure of 14.736 psi and P_{sw} is the saturated water vapor pressure at 60°F of 0.2563 psia. The heat output of the boiler, Q_{out}, was calculated using the formula:

$$Q_{out} = \dot{m} \ \bar{c}_p \ \overline{\Delta T} \ \Delta t ,$$

where the average water mass flow rate, $\dot{m}$, was determined by dividing the measured value of the total mass of water passing through the boiler during the test period by the length of the test period, Δt. The average specific heat of the water, $\bar{c}_p$, was assumed to be equal to the specific heat of water at a temperature equal to one-half the sum of the return water temperature and the average supply water temperature. For all steady state tests, the average temperature rise across the boiler, $\overline{\Delta T}$, was determined by calculating the difference between the supply and return water temperature at 10 minute intervals and then averaging these temperature differences for the test period. For all cyclic tests, the instantaneous temperature differential across the boiler, ΔT, was directly measured on a multi-point strip chart recorder and the average water temperature rise, $\overline{\Delta T}$, was calculated by finding the area under this ΔT-time curve (for the test period) and then dividing it by the length of the test period.

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* Standard conditions are defined as 60°F (15.6°C) and 14.736 psia (101.6 kPa).

Using the heat-loss method, the steady-state efficiency and part-load efficiency at a percentage on-time of 22.5 percent were determined by the calculation procedure specified in reference 3. The only exceptions to this were: (1) the steady-state efficiency was found for the actual return water temperature measured (not 180°F (82°C)), (2) credit was given in the steady-state efficiency calculation for the heat gain resulting from condensate formation, and (3) since the primary interest was a comparison of laboratory results, the measured room air temperature was used in place of the recommended 42°F average outdoor winter temperature in calculating the part-load efficiency.

The basic equations employed in the heat loss method of calculating performance were:

$$\eta_{SS} = 100 - C_L' L_{L,A} - L_{S,SS,A} - L_c', \text{ and}$$

$$\eta_{22.5\%} = 100 - C_L L_{L,A} - L_{S,ON} - L_c,$$

where the last equation follows from the fact that the test boiler was assumed (for the purpose of this analysis) to use indoor air for combustion (i.e. $C_S = 1$), it did not employ a pilot light (i.e. PF = 0), and its design prevented air flow through the heat exchanger during the off-period (i.e. D_F and, therefore, $L_{S,off}$ equal 0).

The quantities $C_L' L_{L,A}$ and $C_L L_{L,A}$ accounted for the latent heat loss due to that part of the water vapor (generated by the burning of hydrogen in the fuel) going up the flue during steady-state and cyclic operation, respectively. For a natural gas, the assigned value of 9.55% was used for $L_{L,A}$ [1,2]. The heat loss coefficients C_L' and C_L were calculated using the equations:

$$C_L' = \left(\frac{p_v^{s'}}{p_v}\right) \left(\frac{p - p_v}{p - p_v^{s'}}\right), \text{ and}$$

$$C_L = \left(\frac{p_v^{s}}{p_v}\right) \left(\frac{p - p_v}{p - p_v^{s}}\right),$$

where p_v is the partial pressure of water vapor in the flue gases if there were no condensation and the atmospheric pressure, P, was assumed to be 14.7 psia. The equation for calculating p_v is given in [3]. The quantities $P_v^{s'}$ and P_v^{s} correspond, respectively, to the saturated vapor pressure of water at the steady-state flue temperature, $T_{F,SS}$, and the average on-period flue gas temperature, $T_{F,ON}$, defined in [3].

The quantities L_C' and L_C account for the loss due to hot condensate going down the drain and for the fact that this condensate did not go up the flue as heated vapor [3]. They were calculated using:

$$L'_C = \frac{L_{L,A}\,(1 - C'_L)}{1053.3}\,((C_{P_{liquid}} - C_{P_{vapor}})\,(T_{F,SS} - T_{LAB})) \quad \text{,and}$$

$$L_c = \frac{L_{L,A}\,(1 - C_L)}{1053.3}\,((C_{P_{liquid}} - C_{P_{vapor}})\,(T_{F,SS} - T_{LAB})),$$

where T_{LAB} was the laboratory air temperature during the test period.

The terms $L_{S,SS,A}$ and $L_{S,ON}$ represent the sensible heat loss during the steady state and cyclic tests, respectively. They were calculated using the procedures prescribed in [1,2,3], except that C_s was set equal to 1.0, which meant that the temperature of the combustion air was not corrected to 42°F (5.6°C). This was done in order to allow the efficiencies calculated by the heat loss method to be properly compared with those obtained using the input/output method.

Since the input/output method accounts for the boiler jacket losses while the heat loss method does not, it was necessary to directly measure the jacket losses in order to fairly compare the results obtained with the two methods. Using the surface and air temperatures as measured in section 4.4, steady state percent jacket losses were calculated for the test unit operating at approximately 100 and 130°F (37.8 and 54.4°C) return water temperatures.

The heat loss rate from the jacket was calculated using the sum of the radiant and convective losses given for each surface area. The equation employed was:

$$H_S = (h_{rs} + h_c)\,A_s\,(T_s - T_a)$$

where H_s is jacket heat loss rate for each surface,

A_s is area of equal surface temperatures,

h_{rs} and h_c are coefficients of radiation and convection for each area of equal surface temperatures, and

T_s and T_a are the surface and room temperatures respectively.

Specific values of the h_{rs} and h_c coefficients were determined for each area of similar surface temperatures. The loss rate for all surface areas was totaled and divided by the measured input fuel rate, Q_{in}, to give the percent jacket losses.

5. DISCUSSION OF RESULTS

The close relationship between the flue gas temperatures and the return water temperature for this condensing boiler is clearly illustrated in Figure 5 for a number of steady state tests conducted at different return water temperatures. The difference between these two temperatures varied from about 14°F (7.8°C) to approximately 4°F (2.2°C) as the return water temperature was increased from 100°F (37.8°C) to 140°F (60.0°C). This resulted in a rapid drop off in the rate at which condensate was generated as the return water temperature was increased, as shown in Figure 6. As a consequence, the steady-state efficiency measured using the input/output method dropped from approximately 97.8% at a return water temperature of 100°F (37.8°C) to approximately 90% at a return water temperature of 140°F (60.0°C).

The exact steady-state efficiencies, calculated using both the input/output and the heat-loss method, are presented, respectively, in Tables 2 and 3, along with information on the various test conditions. Table 2 also contains the results of two steady-state jacket loss tests conducted at return water temperatures of 100°F (37.8°C) and 130°F (54.4°C). This information is plotted in Figure 7, where the dash-dot-dash line through the heat loss data was "estimated" in order to give an inflection point at 125°F (51.7°C)* and the straight solid line through the input/output data represents a linear least-squares fit.

Since the heat loss efficiency data do not include jacket losses while the input/output data do, the latter were adjusted by adding the jacket losses measured at return water temperatures of 100 and 130°F (37.8 and 54.4°C) and plotting the dashed line shown in Figure 7 through the results. Comparing this dashed line with the heat-loss results (the dash-dot-dash line) indicated that the input/output method yielded steady state efficiencies that were slightly greater (between 1 to 3 percentage points) than the corresponding heat loss efficiencies. The above differences tended, however, to be either within or nearly within the range of experimental error at most of the return water temperatures studied.

The results of the cyclic tests conducted at different return water temperatures and different percent on-times are presented in Table 4. The cyclic efficiencies contained in this table, which were calculated using the input/output method, are plotted in Figure 8 along with linear lines representing a least-squares-fit to each set of data. It can be seen from this figure that the efficiency of the boiler in the range of con-

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* A return water temperature of 125°F (51.7°C) corresponds to a flue gas temperature (see Fig. 6) which approximately equals the flue gas vapor's calculated saturation temperature. Flue temperatures below this value should result in condensation, while those above it should not.

ditions studied, is much more strongly dependent on the return water temperature than upon the percent on-time. For a percent on-time of 22.5% (the rating point used in both the non-condensing and condensing furnace/boiler test procedures [1, 2, 3]), the efficiency of the test boiler was found to drop 11 percentage points as the return water temperature was increased from 100°F to 140°F (37.8°C to 60.0°C).

Table 5 contains the results of the cool-down and heat-up tests and the part-load efficiencies predicted for a percent on-time of 22.5% using the procedure described in section 4.5. This calculation procedure is the same as the method proposed for condensing furnaces and boilers in [3], except that a slight adjustment was made for the use of indoor laboratory air for combustion (instead of outdoor air). The predicted part-load efficiency decreased from approximately 94% to 89% with increasing water temperatures from 100 to 140°F (37.8 to 60.0°C).

The efficiencies in Table 4, corresponding to values measured using the input/output method at a percent on-time of 22.5%, are plotted in Figure 9 along with the predicted efficiencies (for the same percent on-time) from Table 5. Lines were "estimated" to both sets of data, and made to have inflection points between return water temperatures of 120 to 125°F (48.9 to 51.7°C). If the line through the measured cyclic test data is adjusted for the effect of jacket loss*, the dashed line in Figure 9 is obtained. Comparing this dashed line with the line through the predicted efficiencies in this figure (the dash-dot-dash line) shows that the input/output method tended to give efficiencies which were two to five percentage points higher than the values obtained using the heat-loss method over the range of return water temperatures studied. However, a large part of this discrepancy can be attributed to the expected experimental error calculated for such tests (see Appendix A).

Another feasible explanation for the input/output method yielding both steady state and part load efficiency results which were slightly higher than those obtained with the heat loss method is the fact that the heat loss method recommended in [3] uses the existing flue gas temperature to estimate the amount of water vapor which was not condensed. If a substantial part of the heat exchanger was actually at temperatures below the measured flue gas temperature, this procedure would tend to overestimate the amount of uncondensed water vapor and thus the latent

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* To adjust for the effect of jacket losses under cyclic operation, it was assumed that the cyclic jacket losses at a given return water temperature could be approximated by the steady state jacket loss (as shown in figure 7) at the same return water temperature divided by 0.225. This approximation, which assumes that the rate of heat loss from the jacket during the off-period is the same as during steady state operation, is likely to overestimate the cyclic jacket losses.

heat losses. An examination of the amount of condensate actually collected tended to indicate that this was indeed the case, but that the overestimated latent heat loss resulted in efficiencies that were on the average only about one percentage point low. Consequently, the use of the existing flue gas temperature to estimate the latent heat losses contributes to, but does not fully explain, the discrepancy between the two methods.

6. SUMMARY

The performance of a gas fired condensing boiler was evaluated in the laboratory for the purpose of providing technical data and background information that would assist NBS in the development of a testing procedure for condensing boilers and furnaces [3].

Steady state tests were performed on the unit at several return water temperatures and the efficiency of the boiler was calculated using both the input/output and heat-loss methods. The steady state efficiency was found by the input/output method to vary between 98% and approximately 90% as the return water temperature was increased from 100 to 140°F (37.8 to 60.0°C). These efficiencies tended to be between one and three percentage points higher than the corresponding steady state efficiencies estimated using the heat-loss method.

Cyclic tests at several different return water temperatures and percent on-times were also conducted. For a percent on-time of 22.5%, the input/output method resulted in efficiencies which varied between 98% and approximately 88% with increasing return water temperatures from 100 to 140°F (37.8 to 60.0°C). Cool-down and heat-up tests were also conducted and the results were used to predict the part load efficiency at different return water temperatures and a fixed percent on-time of 22.5%. The calculation procedure employed was the same as the one proposed for rating condensing furnaces and boilers in [3], except that a correction was made for using indoor laboratory combustion air. A comparison of the efficiencies obtained for the cyclic tests employing a 22.5% on-time with the predicted efficiencies when credit was given for jacket losses, showed that the efficiencies measured using the input/output method were between two and five percentage points greater than the corresponding heat loss method results.

A part of the discrepancy between the results obtained using the input/output method and those obtained using the heat loss method can be explained by the fact that the heat loss calculation procedure [3] uses the exiting flue gas temperature to estimate the amount of uncondensed water vapor. This tends to overestimate the latent heat loss and results in efficiencies which were on the average about one percentage point low. The remaining portion of the difference observed between the two methods is probably due to the large experimental error associated with the input/output method (see Appendix A).

7. REFERENCES

1. Kelly, G.E., Chi, J., Kuklewicz, M.E., "Recommended Testing and Calculation Procedures for Determining the Seasonal Performance of Residential Central Furnaces and Boilers," NBSIR 78-1543 (Washington, DC: NBS, 1978) Available from NTIS,PB289484.

2. "Test Procedures for Furnaces and Vented Home Heating Equipment", Federal Register, Volume 43, No. 91, Wednesday, May 10, 1978, pp. 20147-20205.

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4. ASHRAE Standard Measurement Guide: Section on Temperature Measurements, 41.1-74, American Society of Heating, Refrigerating and Air-Conditioning Engineers, New York, N.Y., 1974.

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6. Kelly, G.E., Didion, D.A., "Energy Conservation Potential of Modular Gas-Fired Boiler Systems;" NBS BSS 79, (Washington, D.C.: NBS 1978). Available from Sup. of Doc., SD Cat. No. C13: 29/2:79.

7. ASHRAE Standard Measurement Guide: Engineering Analysis of Experimental Data, 41.5-75, pp. 4-5, American Society of Heating, Refrigerating and Air-Conditioning Engineers, New York, N.Y., 1976.

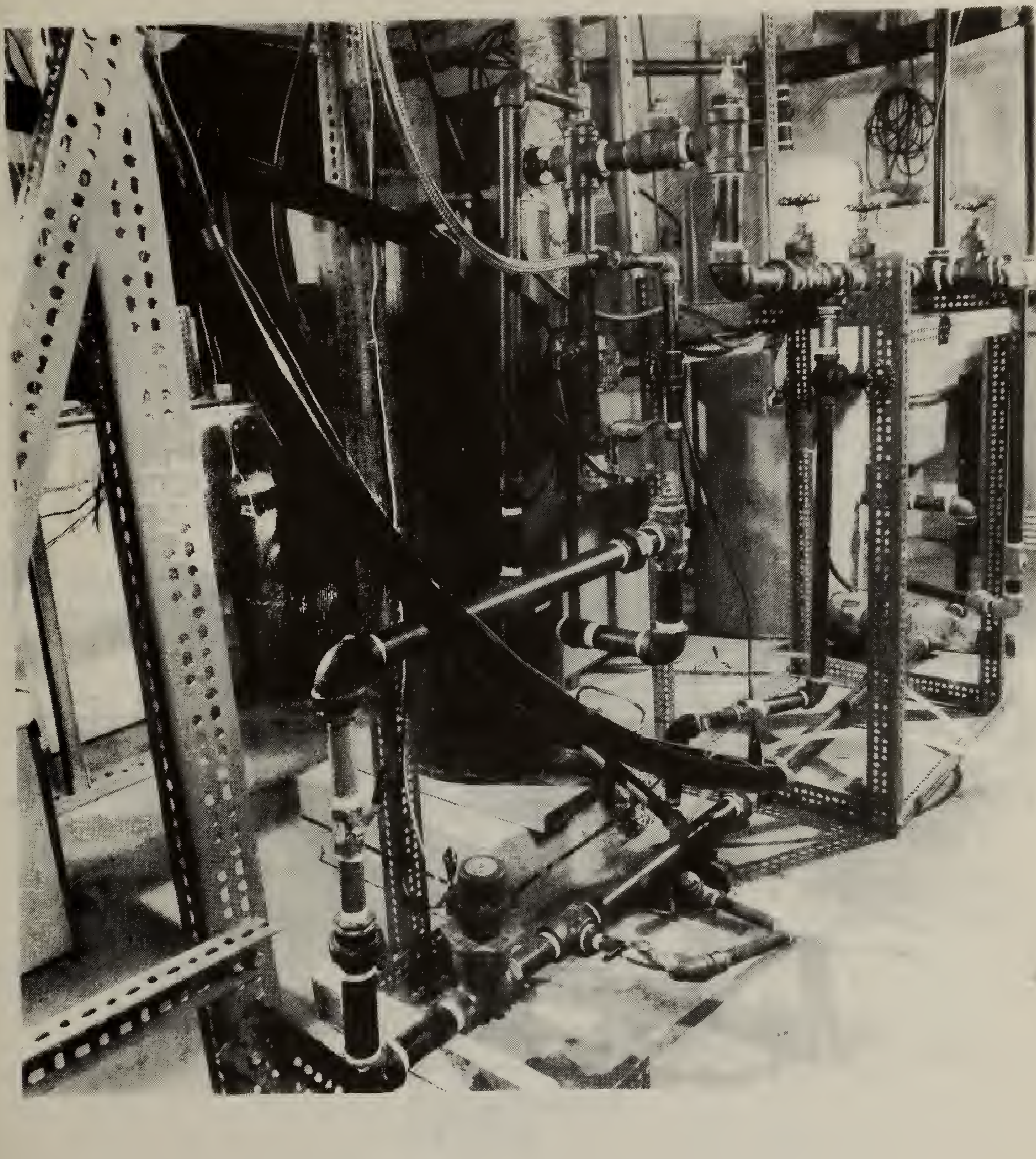

Figure 1. Condensing boiler test arrangement.

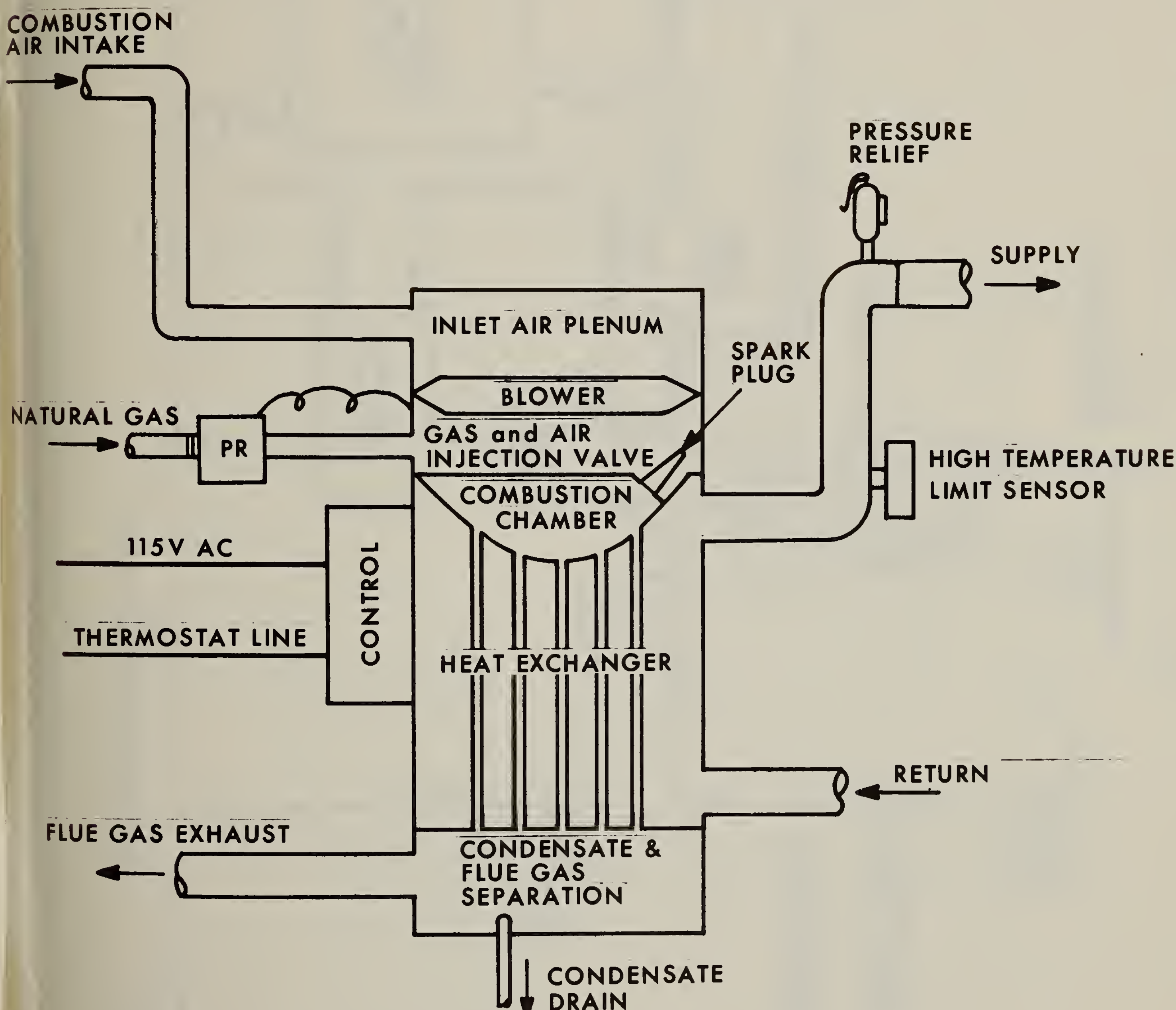

Figure 2. Pulse-combustion condensing boiler schematic.

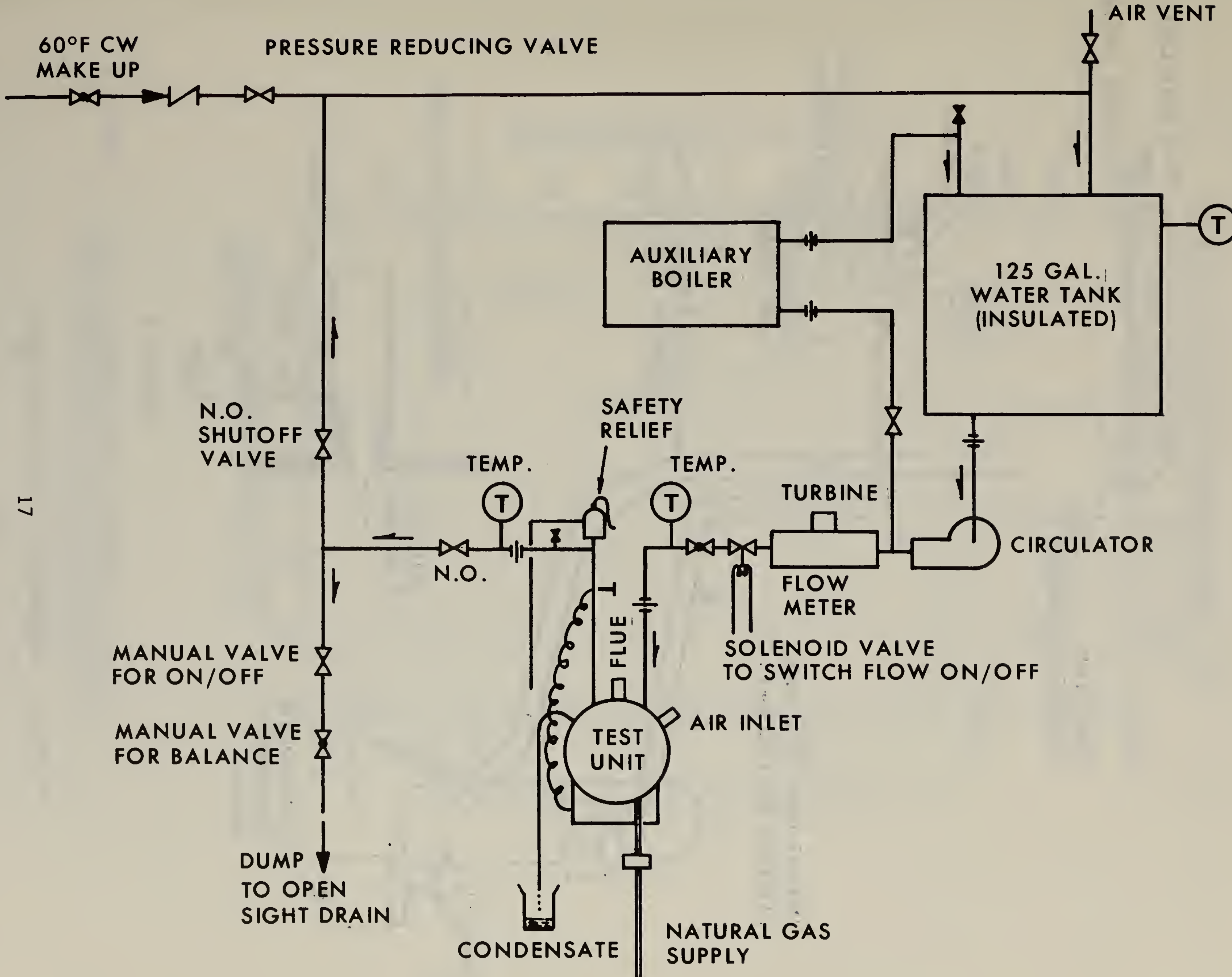

Figure 3. Test apparatus schematic.

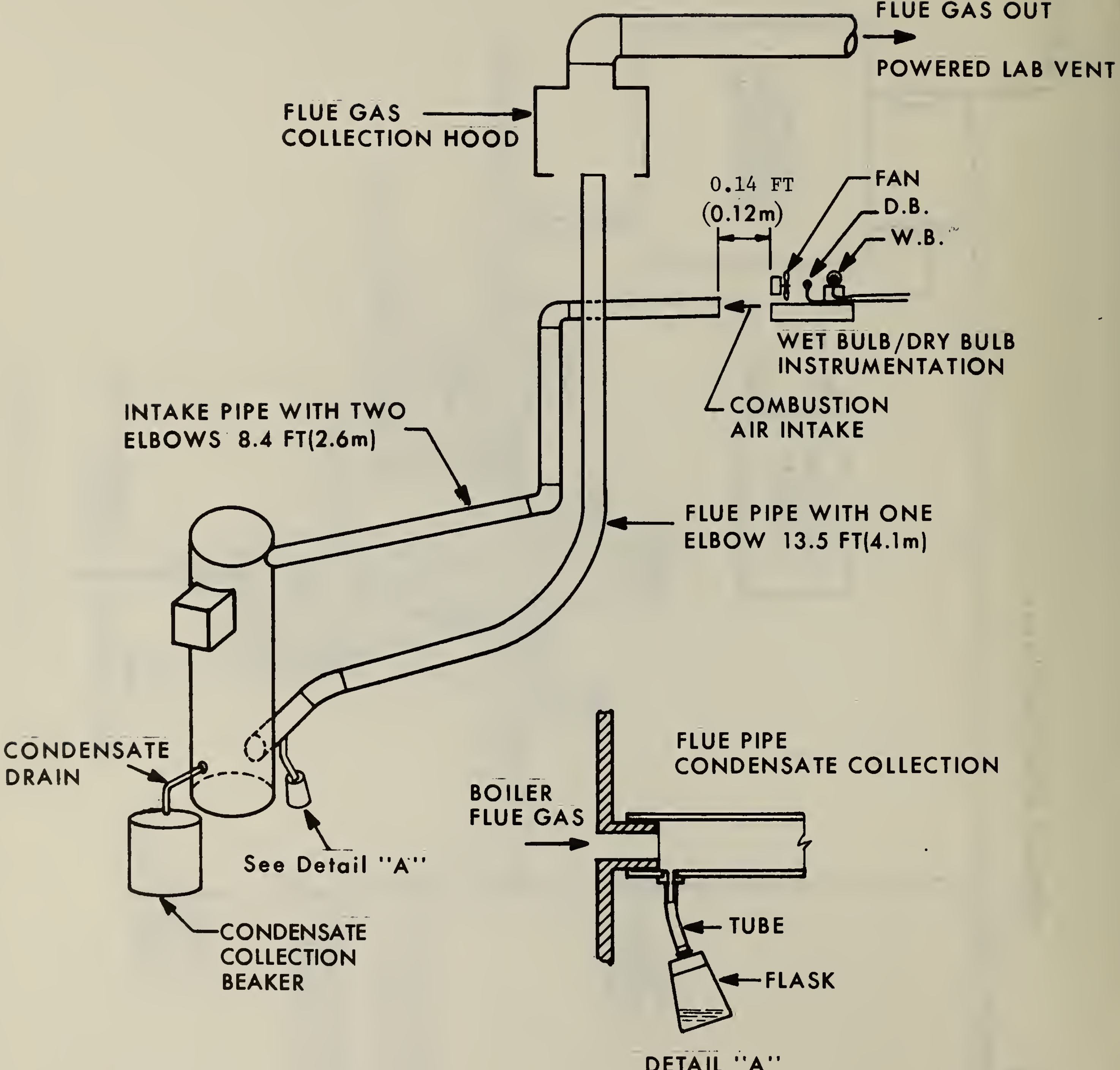

Figure 4. Exhaust/air intake installation.

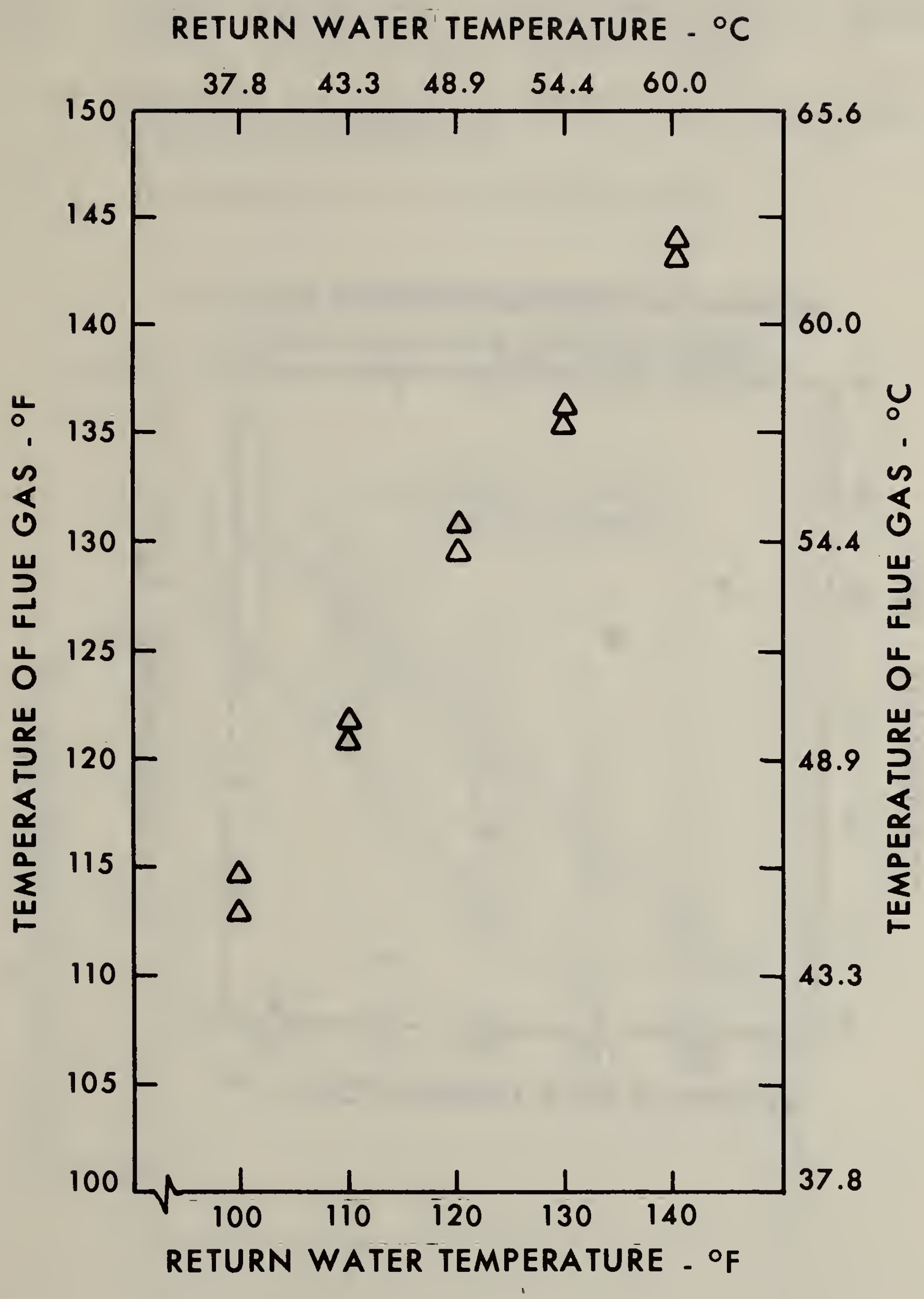

Figure 5. Flue gas temperature dependence on return water temperature from steady-state tests.

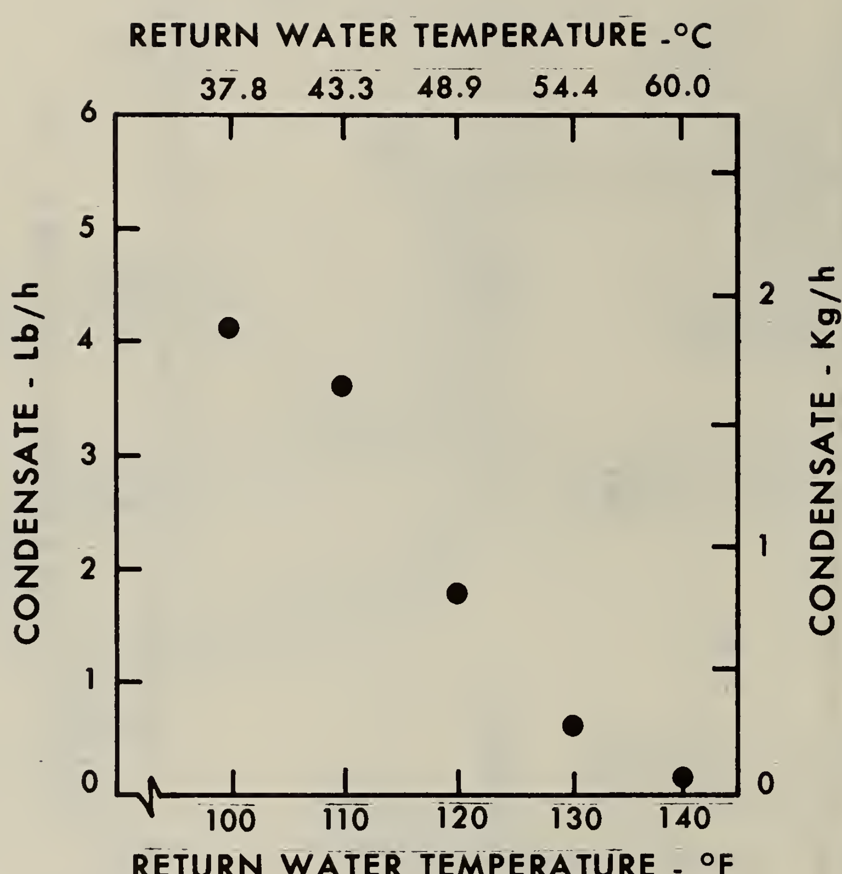

Figure 6. Condensate vs. return water temperature during steady-state tests

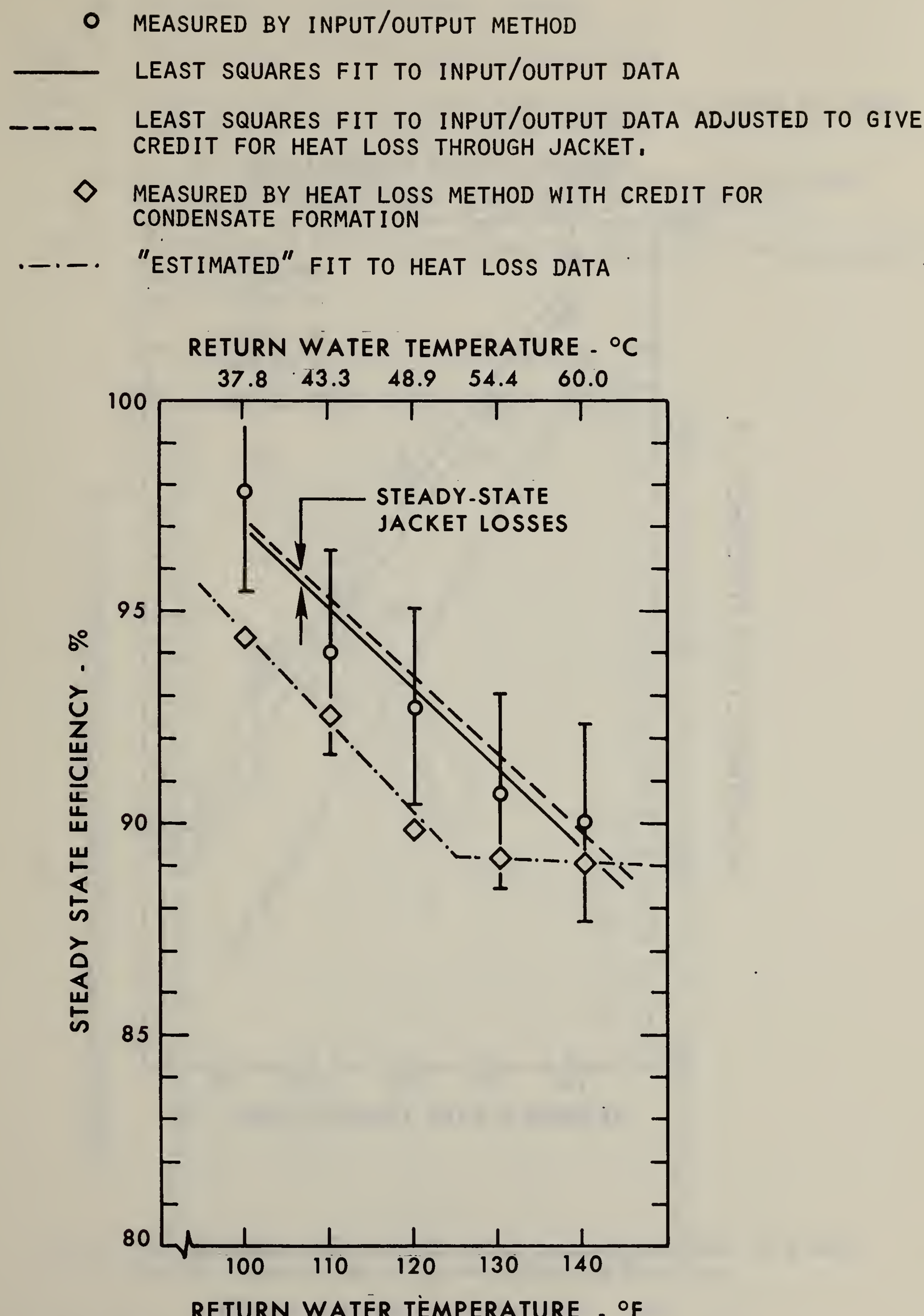

Figure 7. Steady-state efficiency vs. return water temperature.

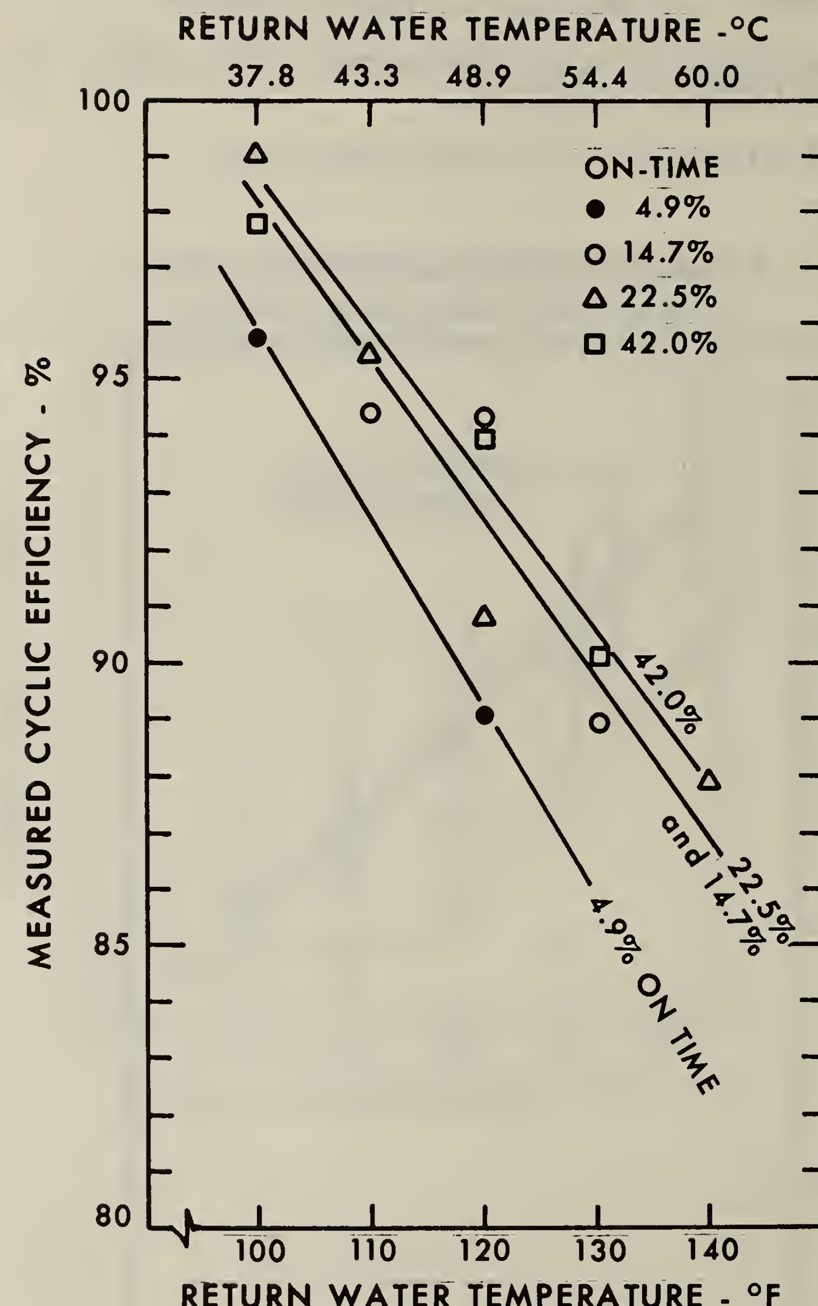

Figure 8. Cyclic efficiency at percent on-times of 4.9, 14.7, 22.5, and 42.0% as determined using input/output method.

△ MEASURED BY INPUT/OUTPUT METHOD

——— "ESTIMATED" FIT TO INPUT/OUTPUT DATA

- - - - "ESTIMATED" FIT TO INPUT/OUTPUT DATA ADJUSTED TO GIVE CREDIT FOR HEAT LOSS THROUGH JACKET

□ EFFICIENCIES BASED UPON COOL-DOWN/HEAT-UP TESTS AND CALCULATION PROCEDURE DESCRIBED IN SECTION 4.5

·—·—· "ESTIMATED" FIT TO EFFICIENCIES DERIVED FROM COOL-DOWN/HEAT-UP DATA

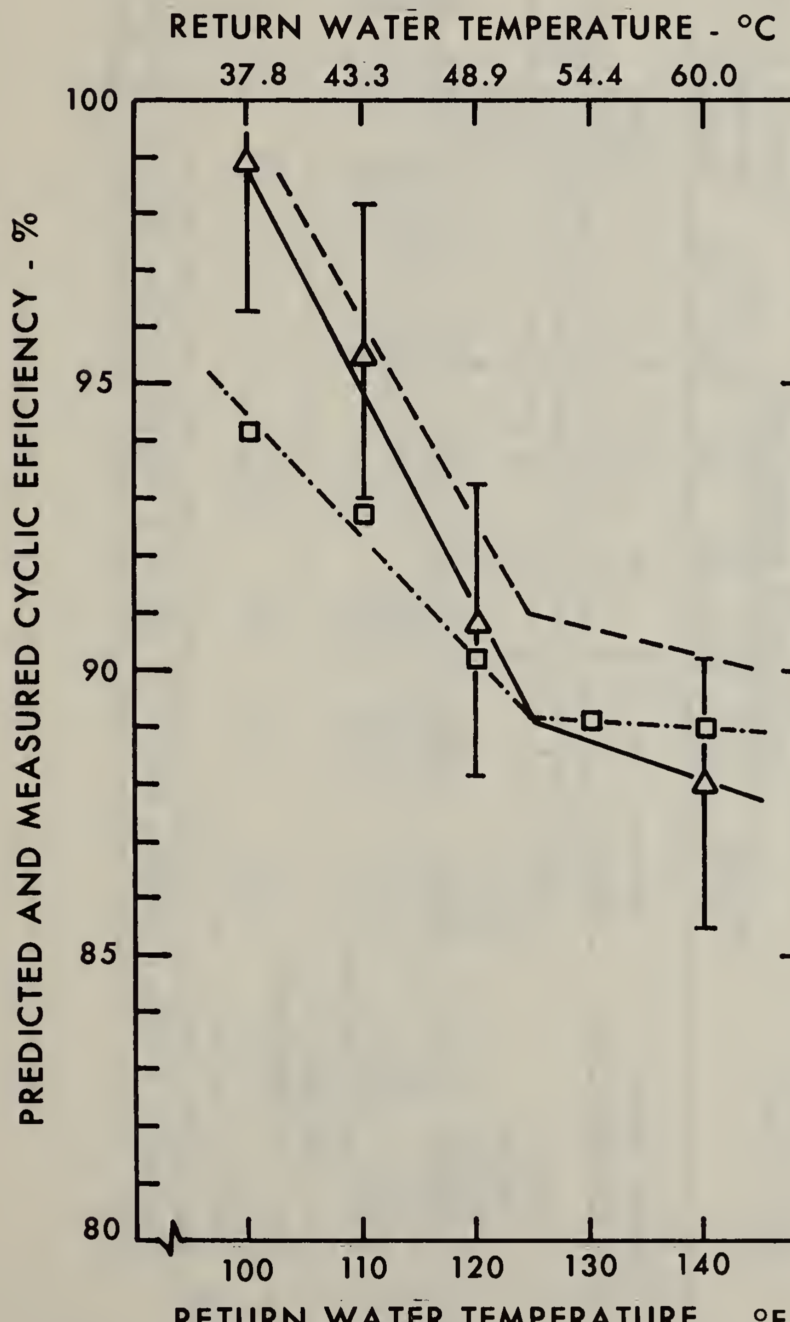

Figure 9. Cyclic efficiency at 22.5% on-time vs. return water temperature.

TABLE 1. GENERAL INFORMATION ON TESTS PERFORMED

TYPE OF TEST	ON-TIME %	RETURN WATER TEMPERATURE 37.8 (°C) 100 (°F)	43.3 (°C) 110 (°F)	48.9 (°C) 120 (°F)	54.4 (°C) 130 (°F)	60.0 (°C) 140 (°F)	DURATION OF OFF/ON TIMES OFF-PERIOD minutes[a]	ON-PERIOD minutes[a]
Steady State	N.A.	X	X	X	X	X	N.A.	N.A.
Cool Down/Heat Up	N.A.	X	X	X	X	X	50.0	5.5
Cyclic	4.9	X	-	X	-	-	153.0	7.89
Cyclic	14.7	-	X	X	X	-	51.0	8.79
Cyclic	22.5	X	X	X	-	X	33.3	9.68
Cyclic	42.0	X	-	X	X	-	17.8	12.90

X - Test Completed
\- No Test
N.A. Not Applicable

[a] The on/off times for the cyclic tests were based upon the performance of a typical thermostat giving a cycling rate of 2 cph at a percent on-time of 50%. The on/off times for the cool-down/heat-up test were the values recommended in [1, 2, 3].

TABLE 2. STEADY-STATE TEST RESULTS OBTAINED USING INPUT/OUTPUT METHOD

Average[a] Return Water Temp. °F	Average[a] ΔT°F	Measured S.S. Efficiency %	Uncertainty Actual Percent %	Q_{IN} Btu/h	Q_{OUT} Btu/h	Condensate[a] Lb_m/h	CO_2 %	O_2 %	CO ppm	D.B °F	W.B. °F	Air rh %	Flue Temperature[a] °F	Jacket Loss % Points
100.3	20.1	97.8	± 2.4	87700	85800	4.18	10.4	3.5	100	77.9	61.5	39	114.8	0.20
110.0	20.3	94.0	± 2.4	87900	82700	—	10.7	3.1	120	77.8	64.0	49	122.0	--
120.2	19.0	92.7	± 2.3	87200	80800	1.84	10.4	3.2	100	77.3	61.9	41	129.4	—
130.2	18.9	90.7	± 2.3	84800	77000	--	10.7	2.8	130	77.4	64.8	52	135.6	0.45
140.2	18.0	90.0	± 2.3	85400	76800	0.18	10.8	3.1	100	77.4	62.0	42	143.2	--

[a] The International System of Units of Measurement (SI) values are included with the text and figures of this document, and within this table the following conversions should be applied to the U.S. conventional units.

To convert from Btu/h to watt (W), multiply by 2.930 711 E-01.

To convert temperatures from degree Fahrenheit to degree Celsius, subtract 32 and divide by 1.8.

To convert lbm/h to kg/s, multiply by 1.259 979 E-04.

To convert differential temperature ΔT degree Fahrenheit to ΔT degree Celsius, divide by 1.8.

Table 3. STEADY STATE TEST RESULTS USING HEAT-LOSS METHOD

AVERAGE RETURN WATER TEMP.[b] °F	AVERAGE TEMP. RISE[b] $\Delta T°F$	CALCULATED STEADY-STATE EFFICIENCY[a] %	BOILER CONDENSATE[b] lb_m/h	O_2 %	CO ppm	rh of COMBUSTION AIR %
99.8	20.5	94.2	4.19	2.8	160	50
109.7	20.8	92.5	3.62	2.9	140	60
120.4	20.5	89.8	---	3.2	100	78
129.6	20.5	89.1	0.64	3.0	120	60
140.0	20.1	89.0	0.23	2.5	150	60

[a] Efficiency calculations adjusted for use of laboratory temperature air for combustion instead of outdoor air.

[b] The Internal System of Units of Measurement (SI) values are included within the text and figures of this document, and within this table the following conversions should be applied to the U.S. conventional units.

To convert temperatures from degree Fahrenheit to degree Celsius, subtract 32 and divide by 1.8.

To convert lbm/h to kg/s, multiply by 1.259 979 E-04.

To convert differential temperature ΔT degree Fahrenheit to ΔT degree Celsius, divide by 1.8.

TABLE 4. CONDENSING BOILER CYCLIC TEST RESULTS

Percent On-Time %	Avg. Cycle[b] Return Water Temp. °F	Cyclic[a] Effy. %	Q_{IN} Btu/h	Q_{OUT} Btu/h	$\dot{m}$ lbm/h	Air rh %	Boiler Condensate lbm/h
	99.2	95.8	4900	4700	4432	50	0.301
4.9	120.3	89.1	4900	4400	4064	62	0.278
	110.0	94.4	14600	13800	4359	56	0.586
14.7	118.6	94.3	14600	13700	4292	52	--
	130.2	88.9	14300	12700	4147	56	--
	100.9	99.0	23400	23200	5402	65	--
22.5	108.8	95.6	22300	21300	4359	64	--
	118.6	90.8	22200	21200	4279	52	--
	138.8	88.0	21400	18800	3922	58	0.190
	102.3	97.8	44300	43300	5107	43	2.293
42.0	118.5	93.9	41300	38800	4286	53	0.951
	130.6	90.1	40300	36300	4151	55	0.355

[a] Measured using input/output method.

[b] The International System of Units of Measurement (SI) values are included within the text and figures of this document, and within this table the following conversions should be applied to the U.S. conventional units.

To convert from Btu/h to watt (W), multiply by 2.930 711 E-01.

To convert temperature from degree Fahrenheit to degree Celsius, subtract 32 and divide by 1.8.

To convert lbm/h to kg/s, multiply by 1.259 979 E-04.

TABLE 5. COOL-DOWN AND HEAT-UP TEST RESULTS AND PREDICTED EFFICIENCY AT A PERCENT ON-TIME OF 22.5%

Return Water Temp.[b] °F	Predicted Efficiency at 22.5% On-Time[a] %	Concentration CO_2 %	TFSS °F	TFOFF3 °F	TFOFF4 °F	TFOFF5 °F	TFON1 °F	TFON2 °F	TRA °F	Q_{IN} Steady-State Btu/h
99.8	94.2	10.9	113.1	108.7	100.8	94.4	111.5	112.1	85.8	82940
109.7	92.7	10.9	121.6	116.2	101.4	85.6	117.8	120.6	77.4	85559
120.4	90.2	10.6	130.8	124.0	109.1	85.8	124.0	129.8	76.5	88594
129.6	89.1	10.9	135.9	128.9	110.7	75.5	131.2	134.8	75.5	84683
140.0	89.0	10.9	144.0	134.8	113.3	78.0	135.0	139.8	78.0	84281

[a] Efficiency calculations adjusted for use of laboratory temperature air for combustion instead of outdoor air.

[b] The International System of Units of measurement (SI) values are included within the text and figures of this document, and within this table the following conversions should be applied to the U.S. conventional units.

To convert from Btu/h to watt (W), multiply by 2.930 711 E-01.

To convert temperatures from degree Fahrenheit to degree Celsius, subtract 32 and divide by 1.8.

APPENDIX

Summary of Errors

The uncertainty for the cyclic efficiency tests can be estimated by reviewing the possible sources of errors in the measured variables used to determine the input and output energy.

The sources of measurement error in calculating the cyclic heat output are inaccuracies in water flowrate, the strip chart recorder and the error associated with determining the area under the temperature differential (across the boiler) vs. time curve. The turbine meter and output totalizers error is estimated at equal to or less than ±1.0%. The strip chart recorder manufacturer's stated error was ±0.5%. The area under the temperature differential vs. time curve for each cycle is believed to be determined within ±0.6%, based upon repeated measurement of a prepared known sample area. Possible error from variation in specific heat of water is ignored as the value used in the calculations was based upon the average temperature of the water within the heat exchanger and should have little effect on the overall uncertainty.

Other possible sources of measurement inaccuracies are the total gas input (±0.5%), the gas higher heating value (±1%), the gas pressure (±2%), and gas temperature (±0.1%). These inaccuracies, when combined with those of the output measurements, give a total uncertainty of ±2.4% to ± 2.7% in the measured part load efficiencies which were applied to the measured cycle data in Figure 9.

The technique used to calculate the above total uncertainty was the recommended ASHRAE Standard Procedure for determining the propagation of uncertainties in single sample experiments [7]. Briefly, this technique assumes that the total percentage uncertainty is equal to the square root of the average of the sum of the squares of the individual percent uncertainties.

The measurement of the temperature rise across the boiler involved the use of a thermopile. The number of thermocouple junctions in this thermopile was selected to result in an error of less than ±0.02°F or 0.11% in a 20 degree Fahrenheit (11.1 degree Celsius) temperature differential across the boiler using the automatic data logger. This error combined with the uncertainties cited above to give an uncertainity in the steady state efficiencies of ±2.4 when determined by the input/output method.

No attempt was made to estimate the accuracy of the heat loss method of determining boiler efficiency because it involves a fairly complex calculation procedure based on numerous assumptions. Instead, an estimate of the <u>precision</u> of the method was obtained by running a computerized version of the procedure to determine the effect of uncertainties in the input variables (e.g. flue temperature & CO_2 reading). The uncertainties used to adjust the measured flue gas temperature and CO_{2F} concentrations were, respectivley, $\pm$ 0.76% and $\pm$ 0.1%. The result of this sensitivity study was that a variation of less than $\pm$0.3 percentage points was observed in the steady-state and part-load efficiencies over the range of return water temperature studied.

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15. SUPPLEMENTARY NOTES

16. ABSTRACT - As a part of the Department of Energy's energy conservation program for consumer products, the National Bureau of Standards (NBS) developed test procedures for conventional gas- and oil-fired furnaces and boilers. The Department of Energy (DoE) published their finalized version of these procedures in the Federal Register on May 10, 1978. In an effort to update and refine these test procedures, DoE directed NBS to develop a method of testing condensing furnaces and boilers which could be used to compare the annual performance of condensing and non-condensing residential heating systems. This report summarizes the laboratory tests of a gas-fired pulse-combustion condensing boiler that were carried out as a part of the development effort.

The performance of the pulse-combustion boiler was evaluated under both steady-state and part-load operating conditions. The efficiency of the unit was determined by the input/output method which measured the heat transferred to the circulating water and the energy input during each test period. Steady-state laboratory tests of the unit were conducted at constant return water temperatures of 100, 110, 120, 130 and 140°F (37.8, 43.3, 48.9, 54.4 and 60.0°C). Part-load performance tests were carried out at a number of these return water temperatures at on-times of approximately 5, 15, 22.5 and 42 percent.

A modified version of the heat loss procedure for estimating the seasonal performance of a residential central furnace or boiler was also used to evaluate the boiler's steady-state efficiency and part-load efficiency at a 22.5 percent on-time. A cool-down test and heat-up test were performed to obtain dynamic information which was used to calculate the unit's cyclic performance. The predicted steady-state and part-load efficiencies from the heat loss method were found to be within three percent of the performance determined using the input/output method.

17. KEY WORDS - Boilers; central heating; condensing boilers; efficiency, part load; fossil-fuel heating systems; gas-fired boilers; hydronic heating; pulse combustion.

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